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M1078 Hybrid Hydraulic Vehicle Fuel Economy Evaluation

INTERIM REPORT
TFLRF No. 430

by
Edwin A. Frame, Joe Redfield, Glenn Wendel,
Vikram Iyengar, Jack Harris,
& Walter Olson

U.S. Army TARDEC Fuels and Lubricants Research Facility
Southwest Research Institute® (SwRI®)
San Antonio, TX

for
U.S. Army TARDEC
Force Projection Technologies
Warren, Michigan

Contract No. W56HZV-09-C-0100 (WD06)

September 2012

Disclaimers

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SwRI® Project No. 08.14734.06**

Approved for public release: distribution unlimited

September 2012

Approved by:

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U.S. Army TARDEC Fuels and Lubricants
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REPORT DOCUMENTATION PAGE			*Form Approved OMB No. 0704-0188*
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1. REPORT DATE *(DD-MM-YYYY)* 28-09-2012	**2. REPORT TYPE** Interim Report		**3. DATES COVERED** *(From - To)* June 2010 – September 2012
4. TITLE AND SUBTITLE M1078 Hybrid Hydraulic Vehicle Fuel Economy Evaluation			**5a. CONTRACT NUMBER** W56HZV-09-C-0100
			5b. GRANT NUMBER
			5c. PROGRAM ELEMENT NUMBER
6. AUTHOR(S) Frame, Edward; Redfield, Joe; Wendel, Glenn; Iyengar, Vikram; Harris, Jack; Olson, Walter			**5d. PROJECT NUMBER** SwRI 08.14734.06
			5e. TASK NUMBER WD 06
			5f. WORK UNIT NUMBER
7. PERFORMING ORGANIZATION NAME(S) AND ADDRESS(ES) U.S. Army TARDEC Fuels and Lubricants Research Facility (SwRI®) Southwest Research Institute® P.O. Drawer 28510 San Antonio, TX 78228-0510			**8. PERFORMING ORGANIZATION REPORT NUMBER** TFLRF No. 430
9. SPONSORING / MONITORING AGENCY NAME(S) AND ADDRESS(ES) U.S. Army RDECOM U.S. Army TARDEC Force Projection Technologies Warren, MI 48397-5000			**10. SPONSOR/MONITOR'S ACRONYM(S)**
			11. SPONSOR/MONITOR'S REPORT NUMBER(S)
12. DISTRIBUTION / AVAILABILITY STATEMENT Approved for public release; distribution unlimited			
13. SUPPLEMENTARY NOTES			
14. ABSTRACT The primary objective of this work is to demonstrate improved fuel economy of a M1078-A1R vehicle through the application of hybrid hydraulic technology. A secondary objective is to evaluate ways that hybrid hydraulic technology may be applied to benefit ancillary vehicle functions and enhance the mission usefulness of the vehicle. A hybrid hydraulic drivetrain was designed, developed, and integrated into a M1079-A1R vehicle.			
15. SUBJECT TERMS			

16. SECURITY CLASSIFICATION OF:			**17. LIMITATION OF ABSTRACT**	**18. NUMBER OF PAGES**	**19a. NAME OF RESPONSIBLE PERSON**
a. REPORT Unclassified	**b. ABSTRACT** Unclassified	**c. THIS PAGE** Unclassified	Unclassified	115	19b. TELEPHONE NUMBER *(include area code)*

Standard Form 298 (Rev. 8-98)
Prescribed by ANSI Std. Z39.18

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EXECUTIVE SUMMARY

The primary objective of this work was to improve fuel economy of a military vehicle or family of vehicles by applying hybrid hydraulic technology. A secondary objective was to investigate ways that hybrid hydraulic technology may be applied to benefit ancillary vehicle functions and to enhance the mission usefulness of the vehicle. The effort was a continuation of a previous program where a preliminary design of a hybrid hydraulic M1078-A1R was developed. The scope of this program was to complete the hybrid hydraulic drivetrain design, fabricate the system, integrate it into a M1078-A1R vehicle, and perform testing that validates and characterizes the fuel economy of the hybrid hydraulic vehicle.

SwRI completed the detailed design of a heavy duty hybrid hydraulic powertrain for a M1078-A1R platform. As part of the vehicle detailed design, a modification package was created for the Allison 7-speed 3700 transmission to allow it to be driven by a hydraulic motor. The existing 3700 transmission was then, characterized, modified, and then tested in a hardware-in-the-loop environment to validate shifting in the new HHV configuration. Using 3-D modeling tools, the designed hybrid hydraulic drivetrain components were placed in the vehicle chassis. The existing drivetrain components were removed and the vehicle HHV drivetrain was installed. A dSPACE microautobox control system was developed and integrated into the HHV system to control the vehicle.

SwRI completed the design of a Hydraulic Tool Power System to allow for the HHV hydraulic system to power a range of off-board hydraulic tools. Using 3-D modeling tools, the system was placed in the HHV vehicle. The system was partially fabricated but not integrated in the vehicle based on the tool power system developmental hydraulic motor system not meeting speed control performance requirements.

The University of Toledo completed two tasks in support of the program. The first task was the development and validation of optimal shift strategy for an Allison 3700 transmission driven by an Eaton hydraulic motor. The second task was the development of speed control systems for HHV auxiliary pumps and the Hydraulic Tool Power System drive hydraulic motor.

Integration of the HHV system into the M1078-A1R was completed and preliminary on-road testing and vehicle shakedown was successful. The HHV was moved to the SwRI heavy duty dynamometer for continued system shakedown and developmental testing. During developmental testing, a hydraulic motor failure occurred. Although not catastrophic, the motor could not be repaired to allow for final fuel economy testing to be completed.

FOREWORD/ACKNOWLEDGMENTS

The U.S. Army TARDEC Fuel and Lubricants Research Facility (TFLRF) located at Southwest Research Institute (SwRI), San Antonio, Texas, performed this work during the period July 30, 2009 through September 30, 2012 under Contract No. W56HZV-09-C-0100. The U.S. Army Tank Automotive RD&E Center, Force Projection Technologies, Warren, Michigan administered the project. Mr. Eric Sattler and Mr. Allen Comfort served as the TARDEC contracting officer's technical representatives. Phat Truong of TARDEC served as project technical monitor.

The authors would like to acknowledge the contribution of the TFLRF technical support staff along with the administrative and report-processing support provided by Ms. Rita Sanchez and Ms. Dianna Barrera.

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ACRONYMS AND ABBREVIATIONS

cc	cubic centimeter
cc/rev	cubic centimeters per revolution
mpg	miles per gallon
rpm	revolutions per minute
mph	miles per hour
gpm	gallons per minute
GCW	Gross Combined Weight
GVW	Gross Vehicle Weight
GVWR	Gross Vehicle Weight Rating
PSI	pounds per square inch
Hp	horse power
kW	kilowatt
mm	millimeter
ACV	Accumulator control valve (HiP)
AMT	Automated Manual Transmissions
COTR	Contract Officer’s Technical Representative
Eaton	Eaton Corporation
ECM	Engine Control Module
FMTV	Family of Medium Tactical Vehicles
FUDS	Federal Urban Driving Cycle
HDPE	High Density Polyethylene
HiP	High Pressure
HHV	Hybrid Hydraulic Vehicle
HUDDS	Highway Urban Dynamometer Driving Schedule
HyToPS	Hydraulic Tool Power System
LoP	Low Pressure
M&S	Modeling & Simulation
MTR	Hydraulic Motor
OEM	Original Equipment Manufacturer
ORD	Operational Requirements Document
PI	Proportional Integral
PMP	Hydraulic Pump
PTO	Power Take Off
PMs	Program Managers
SwRI	Southwest Research Institute
TCM	Transmission Control Module
TFLRF	TARDEC Fuel and Lubricants Research Facility
UT	University of Toledo

1.0 OBJECTIVE AND BACKGROUND

1.1 OBJECTIVE

The primary objective of this work was to demonstrate improved fuel economy of a M1078-A1R vehicle through the application of hybrid hydraulic technology. A secondary objective was to investigate ways that hybrid hydraulic technology may be applied to benefit ancillary vehicle functions and enhance the mission usefulness of the vehicle. The end result of the effort was a functional hybrid hydraulic M1078-A1R and testing that validates and characterizes the fuel economy of the hybrid hydraulic vehicle. A further benefit of this program at the end was to have a hybrid hydraulic vehicle that can be used as a test bed for the inclusion of advanced hybrid hydraulic components and systems as they are developed for TARDEC.

Commercial and governmental entities have evolved hybrid electric and hybrid hydraulic technology with the goal of increasing fuel economy and reducing vehicle emissions. Some applications achieve higher fuel economy when implementing hybrid electric technology and others achieve higher fuel economy with hybrid hydraulic technology. There is not a clear threshold which determines whether hydraulic or electric hybrid technology is the best choice. Much depends on the weight of the vehicle and its typical drive (duty) cycle. For instance, hybrid hydraulic technologies have been shown to be more efficient than hybrid electric technologies in heavily loaded vehicles that have a great deal of stop and go driving in their duty cycles. Recent work by industry and Government entities has improved the efficiency, functionality and reliability of hydraulic components.

1.2 BACKGROUND

This work is the continuation of a previous program that developed a design for a hybrid hydraulic M1078-A1R. The previous work was conducted under contract DAAE07-99-C-L053, Work Directive 50 in a program entitled, “Military Hybrid Hydraulic Vehicle Analysis and Design”.

The focus of this work is to complete the design of the hybrid hydraulic M1078-A1R, implement the design on the M1078-A1R target vehicle, and explore what fuel economy benefits could be realized by collecting fuel economy data for comparison of the hybrid hydraulic and baseline vehicle configurations.

2.0 APPROACH

The approach taken in this program to realizing the project objectives is delineated in the following sections. Each section specifically targets a particular aspect to the overall program. Special emphasis was placed on describing activities or design information that will assist existing and future programs exploring the benefits of hybrid hydraulic technology on heavy duty military applications.

2.1 POWERTRAIN DESIGN AND DEVELOPMENT

The first element of this work was to expand the detailed design developed under the previous effort into a design package that could be fabricated and installed on the M1078-A1R test platform. The approach for developing a design package for vehicle involved dividing the overall design into two separate sections, the mechanical drivetrain design and the hydraulic system design. Each are discussed separately below.

2.1.1 Drivetrain Design

The drivetrain design that was developed under the previous program was the result of extensive modeling and simulation (M&S) activities. Numerous drivetrain configurations were considered. For many of the configurations initial optimization of hardware and associated control strategies were required for comparison. In addition, the three major hydraulic component suppliers were engaged to assist in the specification of off-the-shelf or near term available hydraulic components. The outcome of the previous program design activities was to finalize on the Hybrid Hydraulic Vehicle (HHV) design that would meet all the vehicle acceleration and grade performance requirements as well as increase the vehicles fuel economy.

The key component of the HHV drivetrain design was the transmission concept. Initially, concepts were developed that were similar to comparable HHV's having two or three speed Automated Manual Transmissions (AMT). The concepts contained one high numeric ratio to assist 60% grade climb and one cruising ratio close to 1:1. With further analysis, however, it became apparent that additional ratios were necessary to provide similar advantages for performance and economy optimization (including regeneration) as they do with standard powertrains. With this realization, the final design was based on adapting the current Allison 3700 (7 forward speeds) to the HHV.

The approach for incorporating the Allison 3700 transmission into the HHV includes the following transmission changes for the 3700's adaptation to the HHV are:

1. Elimination of the torque converter
2. Elimination of the transmission fixed displacement pump
3. Elimination of the mechanical Power Take Off (PTO)
4. Elimination of the transfer case scavenge pump that was driven by the PTO
5. Addition of a remote auxiliary pump system to replace the eliminated pumps

The program approach for implementing these changes is described in Section 2.5, Allison Transmission Characterization.

The transmission plays a crucial role in the approach for enabling the hybrid hydraulic powertrain to meet FMTV requirements. Table 2.1.1-1, FMTV Performance Requirements, illustrates the transmission design features that contribute to meeting the performance metrics.

Table 2.1.1-1. FMTV Performance Requirements

Performance Metric	FMTV Requirement	Trans. Design Contribution
Speed:		
0% Slope	Governed Max Speed of 65MPH	7 Speeds provided
2% Slope	55MPH @ GVW, 40 @ GCW	7 Speeds provided
3% Slope	44MPH @ GVW, 30 @ GCW	7 Speeds provided
60% Grade Operation	Climb and descend at GVW with intermediate starts and stops on a dry concrete surface free from any lose material without stalling, slipping, overheating, or upsetting.	6.93 First Gear Ratio. No torque converter – minimal cooling requirements
Vertical Step	Negotiate a 24” vertical step in forward and reverse	With fully inflated tires, the HHV drivetrain’s ground clearance will exceed this requirement by providing 90mm clearance.
Fuel Economy	Capable of 300 miles minimum on highway with integral fuel capacity at any speed greater or equal to 25MPH	7-Speeds provided enabling optimum ratio choice for pump motor operation during acceleration and regeneration

The following functional HHV requirements are specific to the transmission. Table 2.1.1-2, Transmission Functional Requirements, illustrates the approach to meet program objectives.

Table 2.1.1-2. Transmission Functional Requirements

Functional Requirement	Design Solution
Automatic Shifting	The modified Allison 3700SP design will function just as the OEM design does in the current FMTV. It will shift without operator involvement and provide the same choice of gear ranges which the operator selects manually. Allison Transmission has agreed to provide technical information on the production Transmission Control Module (TCM) to SwRI. SwRI engineers will develop logic in the main hydraulic hybrid system control module to request the TCM to shift the transmission.
Clutch Operation with engine-off (hydraulic-only power) or at low RPM	The auxiliary pump system will provide clutch pressure at low input shaft rpm and even with engine off. The pump system will be driven by hydraulic energy stored in the accumulators.
Park Mechanism	Not Required – Vehicle air brake system used to immobilize vehicle when parked – Same as current FMTV
Limp Home Provision - electric system failures	Same as current Allison 3700SP
Durable / Reliable	The current Allison 3700SP was designed for applications that have up to 1450 ft-lbs of input shaft torque. The 160CC pump motor will not deliver more input shaft torque than 812 ft-lbs. The transmission capacity can be used to evaluate larger hydraulic motors as part of the planned HHV technology demonstrator.

The baseline FMTV powertrain consists of a 205 kW Caterpillar C7 engine driving an Allison 3700SP transmission. The seven speed Allison drives a single speed, lockable transfer case, which transmits power via driveshafts to the front and rear ring and pinion sets, differentials, and axles. Wheel hub units mounted on the ends of the axles provide additional ratio reduction and drive the front and rear wheels. This arrangement is illustrated in Figure 2.1.1-1, M1078-A1R Baseline Diagram.

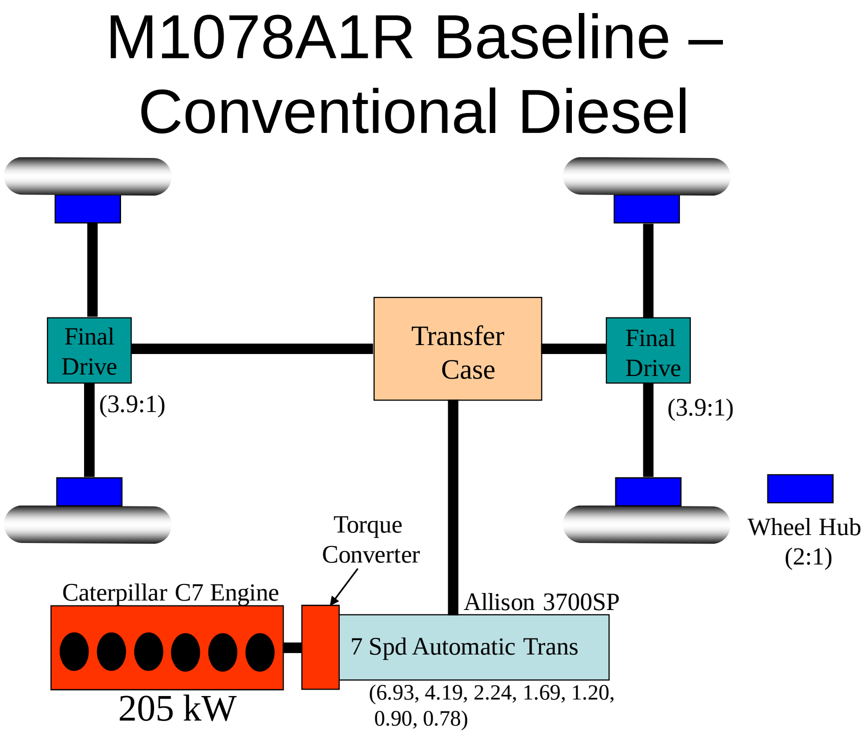

Figure 2.1.1-1. M1078-A1R Baseline Diagram

A block diagram of the series hydraulic hybrid system approach is shown in Figure 2.1.1-2, Hybrid Hydraulic Vehicle Block Diagram.

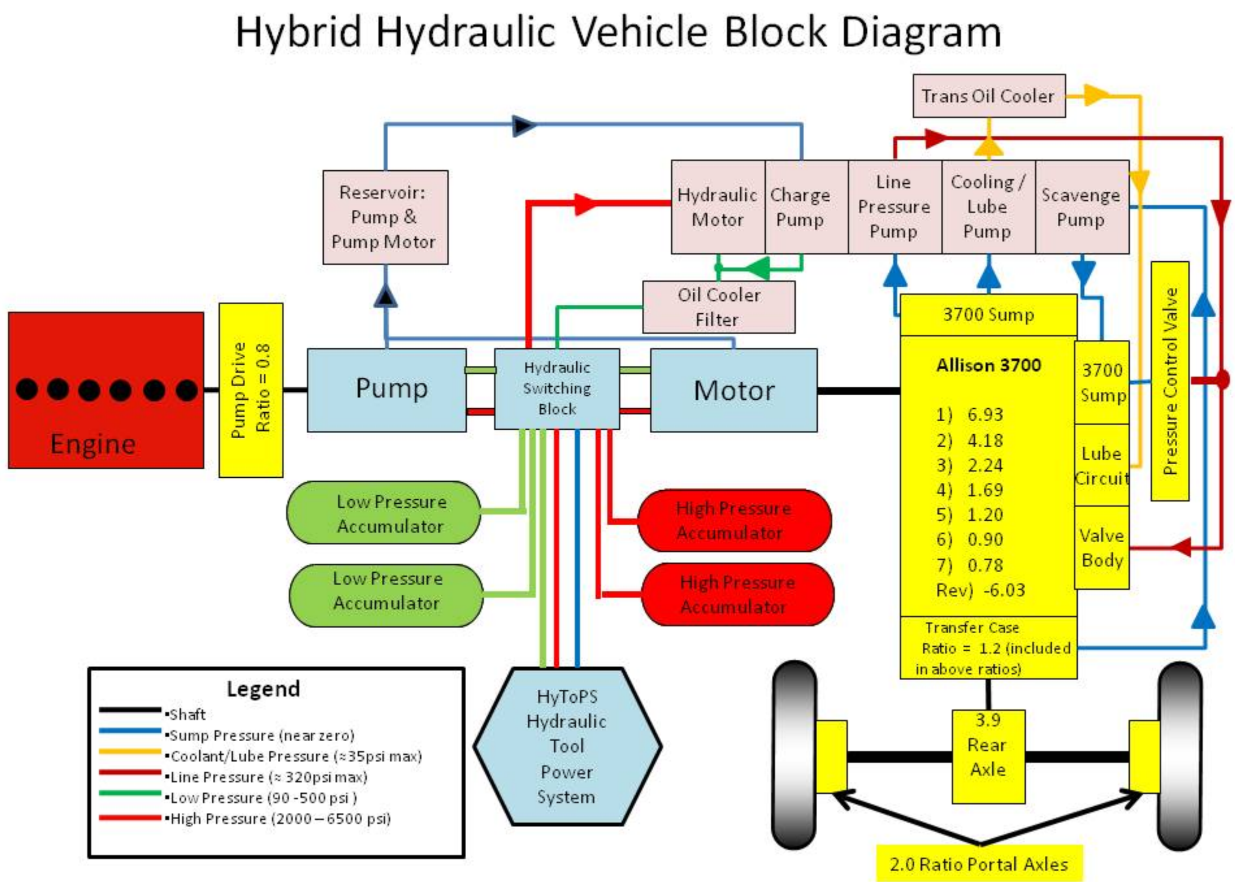

Figure 2.1.1-2. Hybrid Hydraulic Vehicle Block Diagram

Referring to Figure 2.1.1-2, the engine drives the main pump via a fixed ratio gearbox which enables both the engine and the pump to operate at their peak power speeds simultaneously. The engine-driven pump develops hydraulic pressure which drives the hydraulic motor and charges the accumulator system. The hydraulic motor's output shaft is mechanically connected to the transmission input shaft. The HHV will use the conventional FMTV transfer case, axles, and hub units. An auxiliary pump system is required to replace the conventional transmission pump and transfer case scavenge pump. The system also contains a charge pump which returns fluid from the main hydraulic pump and motor to the low-pressure accumulators via the Hydraulic Switching Block. The auxiliary pump system is driven by a hydraulic motor which receives energy from the accumulators. The Hydraulic Tool Power System (HyToPS) also receives energy from the accumulators via the switching block.

Pro/Engineer solid models show the planned location of the hybrid powertrain components. Figure 2.1.1-3, Top/Plan View of HHV Powertrain, illustrates the offset layout of the engine, gearbox and pump assembly and the motor and transmission assembly. The offset engine gearbox allows the pump and motor to be offset which minimizes the total length of the HHV powertrain. This is shown in Figure 2.1.1-4, OEM and HHV Powertrain Length Comparison.

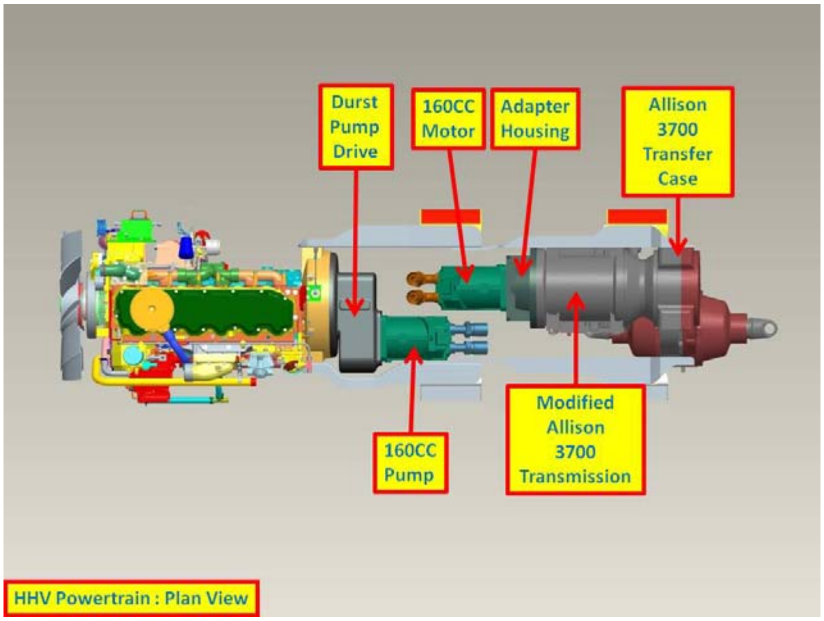

Figure 2.1.1-3. Top/Plan View of HHV Powertrain

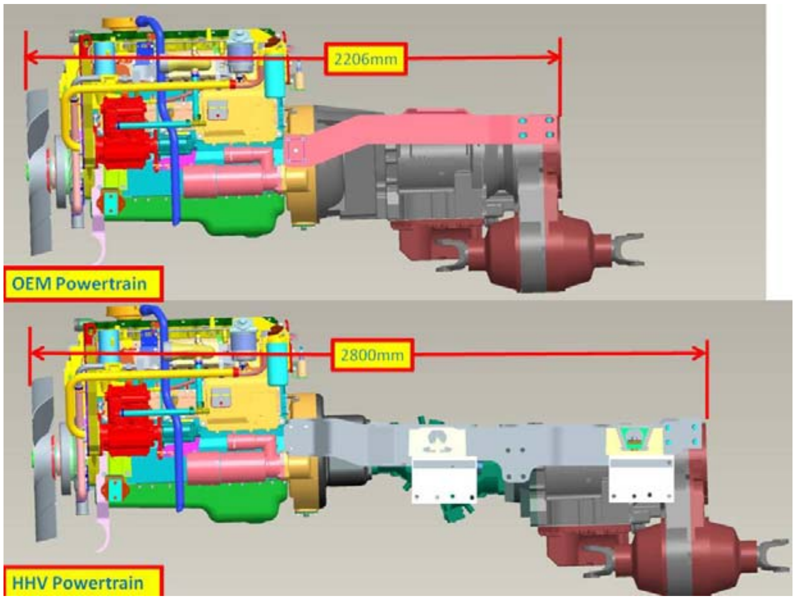

Figure 2.1.1-4. OEM and HHV Powertrain Length Comparison

The OEM engine and transmission assembly is mounted within a common subframe, which is supported by isolation mounts on the vehicle chassis frame. A concept of this same arrangement for the HHV configured vehicle is shown in Figure 2.1.1-5, HHV Powertrain.

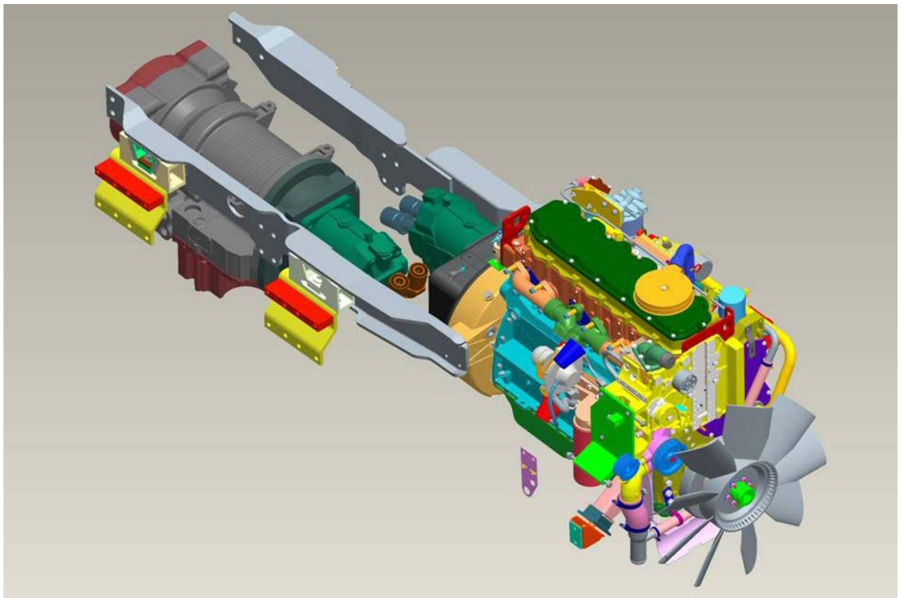

Figure 2.1.1-5. HHV Powertrain

The underside of the FMTV in HHV configuration is shown in Figures 2.1.1-6, Underbody View of the HHV Powertrain and Figure 2.1.1-7, Complete View of HHV Underbody. Hydraulic hoses for the main hydraulic system and the transmission hydraulic system are routed through and around the frame members and powertrain components. Custom mounting brackets are used to support the major and minor HHV components assembled to the chassis. A hydraulic switching valve controls the flow to the vehicle hydraulic components.

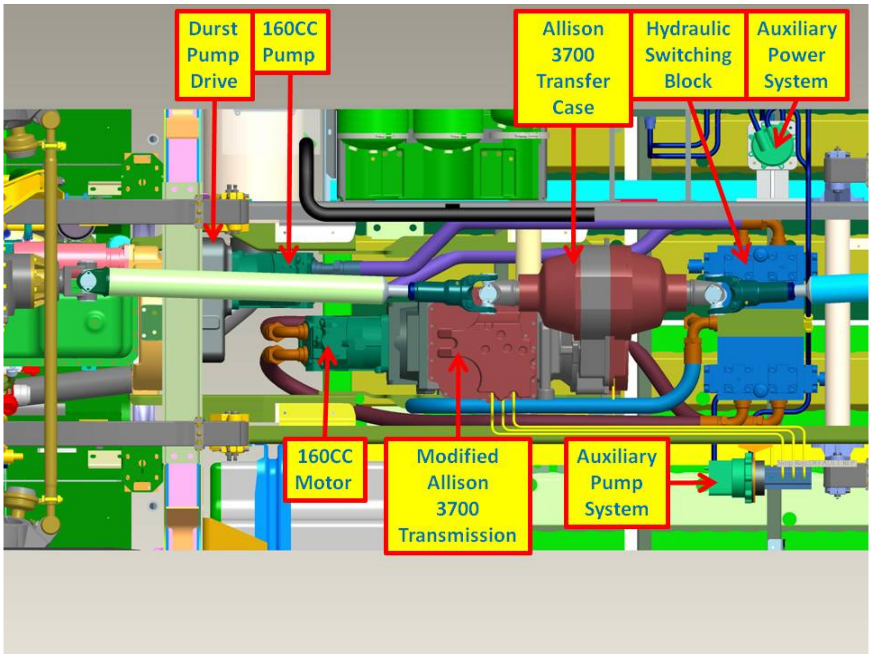

Figure 2.1.1-6. Underbody View of the HHV Powertrain

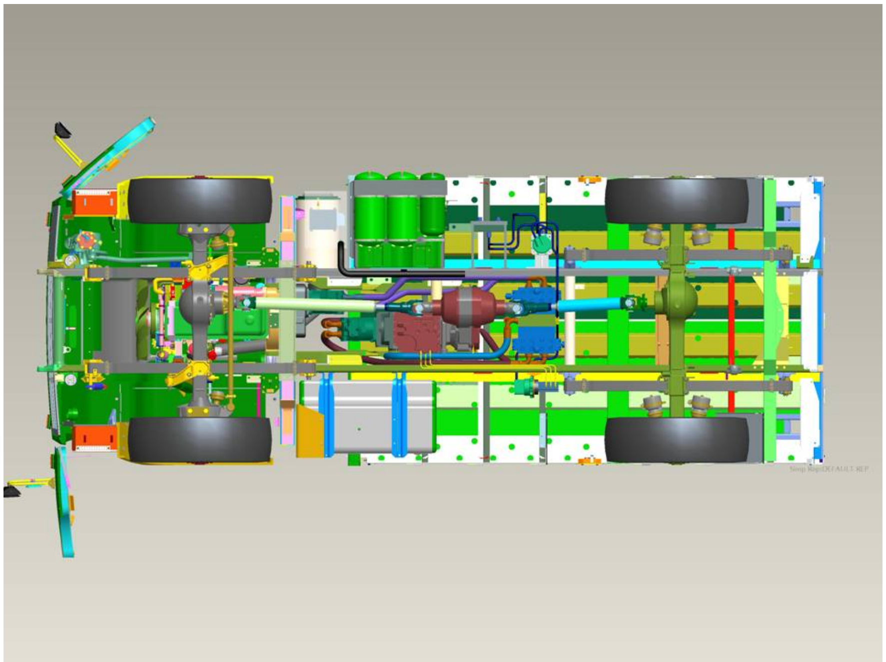

Figure 2.1.1-7. Complete View of HHV Underbody

The accumulators are located in the bed for the demonstration vehicle as shown in Figure 2.1.1-8, Accumulator Bed Mounting, and Figure 2.1.1-9, Complete HHV Vehicle View. This is a cost avoidance measure since non-standard shaped accumulators, which are required for underbody packaging, would be expensive and thus would take project resources away from higher-priority aspects of the project. Further, modifications to the OEM vehicle's frame and chassis assembly would be extensive if standard hardware was installed in the chassis assembly.

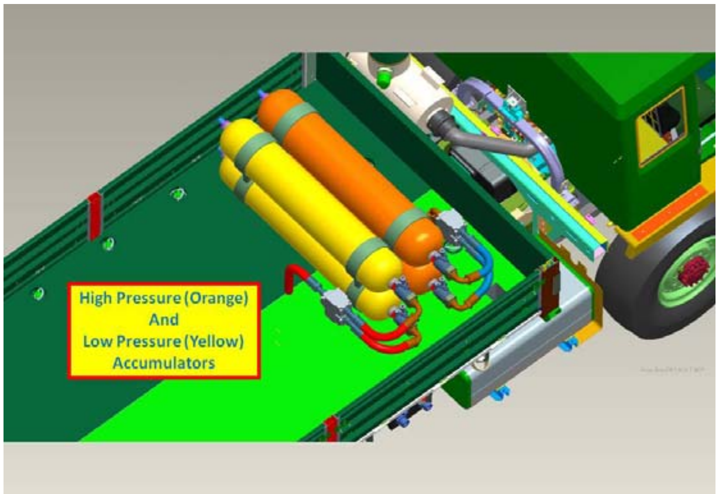

Figure 2.1.1-8. Accumulator Bed Mounting

Figure 2.1.1-9. Complete HHV Vehicle View

2.1.2 Hydraulics Design

The engine-driven hydraulic pump and the pump motor used to drive the transmission were supplied by Eaton. The size of both units is 160 cc/rev, which is the maximum theoretical volumetric displacement. These units are of the variable displacement bent-axis type, which are generally capable of higher efficiencies than other types of hydraulic pump motors such as swashplate-type axial-piston pumps. This type of pump motor is most commonly used commercially as a drive motor for hydrostatic transmissions, in which case it is generally designed with a minimum displacement that is 20 percent of the maximum displacement. For hybrid vehicle applications, the minimum displacement is required to go to zero displacement to provide a smooth transition in applying drive or braking torque. Eaton and one of its partner companies, Brevini, have developed a version of their commercially available 160 cc/rev bent-axis motor that has a zero minimum displacement.

The 160 cc/rev pump motor is capable of delivering peak engine power of 300 hp (224 kW) to the hydraulic drivetrain when the system pressure state is above 5000 psi (345 bar). The drive pump motor is capable of delivering full engine power to the transmission when the proper transmission shift strategy is employed.

The selected bent-axis pump motors are not capable of over-center operation as are swashplate-type axial-piston pumps. A switching valve is required to direct the flow of high and low pressure fluid to the correct side of the pump motors to accommodate either pumping or motoring operation. Flow through the drive pump motor will always be in one direction. When the switching valve is set to motoring mode it will direct high-pressure to flow into the pump motor to provide propulsion torque and power to the transmission to accelerate the vehicle forward and maintain a desired speed. For regenerative braking, the switching valve is set to pumping mode for which the flow coming out of the pump motor is directed to high-pressure to store the braking energy in the high-pressure accumulators. This in turn results in a negative torque into the transmission which decelerates, or brakes the vehicle instead of relying on the normal friction type service brakes. The drive motor is never intended to go into reverse. To obtain reverse vehicle motion the Allison transmission will shift to reverse gear. For the engine-driven pump, the switching valve normally directs high-pressure flow from the pump to the high-pressure accumulators, converting engine power into hydraulic power. When the switching valve is changed to motor mode, high pressure fluid is directed to the inlet of the pump, which is now acting as a motor to provide torque to start the engine, particularly for use when engine-off operation is employed.

The switching valve for both the engine-driven pump and the drive pump motor is located remotely in individual manifolds, mounted directly on the pump/motors. A third manifold, which was called the Megablock, was designed to serve as a common gallery for high-pressure and low-pressure fluid, as well as connection to an atmospheric reservoir. The fluid flow is routed to the accumulator, the pump, the drive pump motor, the Hydraulic Tool Power System, the power steering gear, and the auxiliary pump system which contains the transmission pumps and the charge pump. A schematic of the primary hydraulic circuit is shown in Figure 2.1.2-1, which includes the Megablock, the two pump/motors and the accumulators. The engine driven pump, the drive pump motor, and the Megablock were designed and manufactured by Eaton for this project.

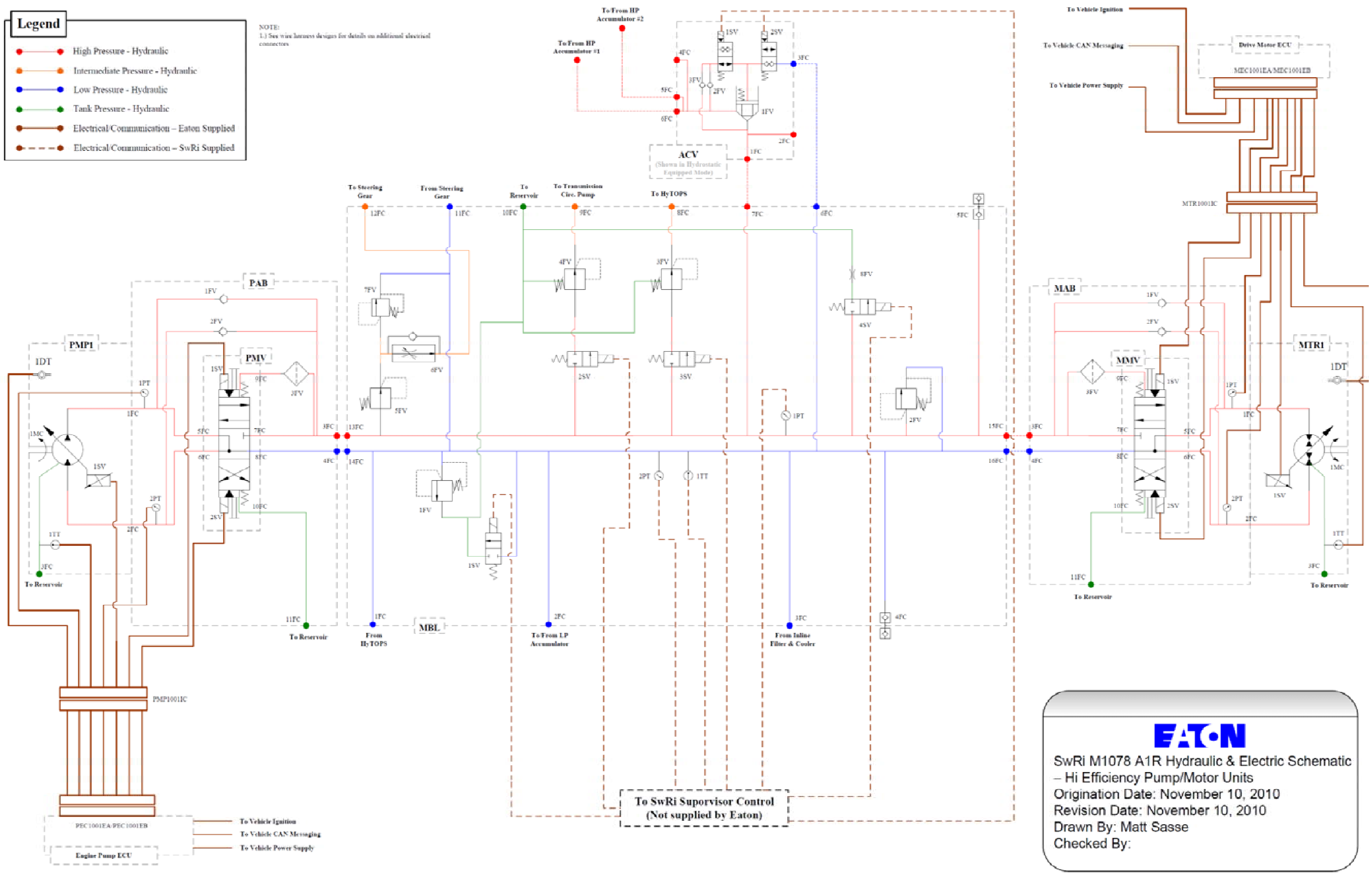

Figure 2.1.2-1. Schematic of Primary Hydraulic System

The switching valves themselves are 4-way, 3-position spool valves. The neutral position blocks high pressure from being exposed to the pump but connects low-pressure to both sides of the pump motor to minimize leakage losses. When activated in one direction or the other, high and low-pressure are connected to one or the other of the two main ports of the pump and pump motor. The switching valve manifold also contains a set of safety check valves each for the pump and the drive pump motor. These check valves protect the pump and pump motor from over-pressurizing or cavitating, in the event that the 4-way valve has excessive restriction.

The accumulators designed for the vehicle are a composite bladder-type design, which were developed by SwRI. The size selected is two 22 gallon (nominal internal volume) tanks, for a total volume of 44 gallons for the high-pressure accumulator system, and the same for the low-pressure system. A cross-sectional view of a bladder-type 22 gallon composite accumulator is shown in Figure 2.1.2-2, 22 gallon 7000 psi Composite Bladder Accumulator. The accumulator vessel design is based on proven technology, similar to that used for fuel storage on vehicles powered by compressed natural gas (CNG). These high pressure accumulator vessels were fabricated by Lincoln Composites in accordance to the ANSI/CSA Standard NGV-2 2000. The high-pressure vessels are of the Type NGV2-4 with a carbon fiber overwrap. The liner consists of a plastic, High Density Polyethylene (HDPE) material, with metal bosses molded into the ends for fluid and gas connections. With this construction, the plastic liner and boss provide a seal for the internally pressurized fluid. The carbon fiber overwrap provides the structural support to contain the internal pressure. This is the best construction for applications with high cycles because carbon fiber has a very good fatigue life, and the plastic liner is not highly stressed in a way that would result in a fatigue failure. The required stress ratio for this type of vessel in compressed natural gas applications is 2.25, however, since HHV applications have many more cycles, a stress ratio of at least 3.0 was used as an added safety margin. The vessels procured include high pressure vessels rated for a 7,000 psi working pressure with a 3.0 stress ratio. These accumulators were constructed in accordance to the NGV-2 standard; however, they were not qualified to that standard, because no standard is currently in place that applies to accumulators used on hydraulic hybrid vehicle applications.

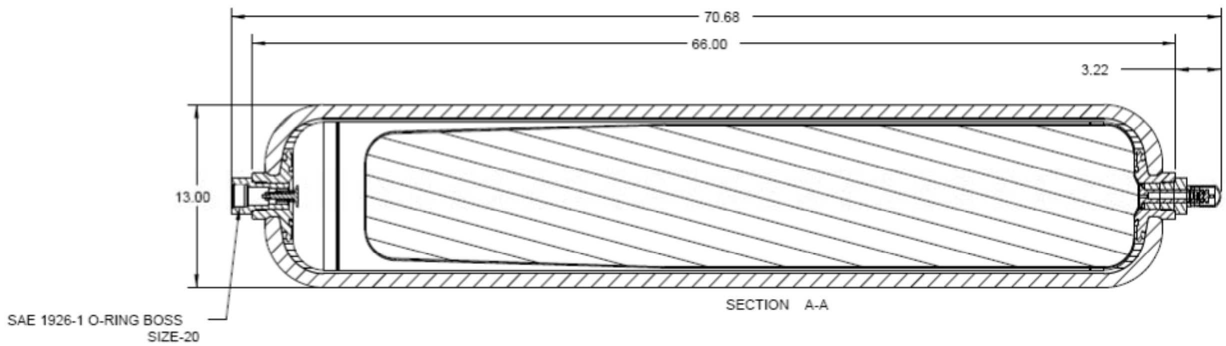

Figure 2.1.2-2. 22 Gallon 7000 psi Composite Bladder Accumulator (Units = Inches)

The low-pressure accumulator vessel is also be of the Type NGV2-4. It has an operating pressure rating of 500 psi and a stress ratio of approximately 5.0. It is overdesigned to contain the pressure, to ensure adequate thickness of the overwrap and to prevent damage during handling when it is unpressurized. The fiber overwrap is a mixture of carbon and glass fiber. The low-pressure accumulator also includes a bleed port, which is an additional small port installed to bleed off any gas that may have collected in the accumulator due to gas permeation across the bladder.

Both high and low-pressure accumulators utilize the latest barrier technology between the gas and the hydraulic fluid. The most common and extensively proven method is a molded nitrile rubber bladder. However, nitrile rubber bladders have a higher than desirable permeation rate, so maintenance procedures were developed during this project to remove permeated gas from the system and to monitor the accumulator gas precharge.

The accumulator bladders have a gas stem that protrudes out of the vessel. A small manifold is be attached to the gas stem. Three components attached to the stem are: a gas fill valve, a pressure transducer, and an overpressure safety valve with a burst disc.

All bladder accumulators require an anti-extrusion device to prevent the bladder from extruding out of the fluid port when the fluid is fully expelled. In addition, an accumulator shut-off valve (ACV) is mounted to the one of the high pressure accumulator shut-off valves and ported to the second high pressure accumulator to open and close accumulator pressure from the rest of the system. This valve is normally in a failsafe closed position and can be commanded to open by activating an electric solenoid. The anti-extrusion valves also act as a safety fuse, also called a velocity fuse. In the unlikely event of a large high-pressure leak that would result in flow rates that exceed normal flow rates, the valve will automatically shut off, minimizing the release of fluid and energy. When the value is in the de-energized normally closed position, there is an option for two different functions. The ACV can act as a check valve and allow flow into the accumulator, but not out. The valve can also be configured to lock the valve closed, preventing flow into the accumulator, enabling hydrostatic operation of the hydraulic drivetrain. The ACV was set up in the latter configuration. The accumulator shut-off valve was designed and manufactured by Eaton for this project.

The energy stored in the high pressure hybrid system is used to provide power to three subsystems. The maximum rated pressure for the hydraulic hybrid drivetrain is 6500 psi. Each of the subsystems, the Hydraulic Tool Power System drive motor, the transmission lubrication pumps, and the steering gear) is rated for lower pressures. High-pressure flow is supplied to each subsystem by activating a solenoid valve, which then passes through a pressure-reducing valve to drop the pressure down to the pressure rating of the components. The drive motor for the transmission pump system and for the hydraulic tool power supply is rated for a maximum of 4600 psi. The steering gear is rated for 2600 psi.

2.2 HYDRALIC TOOL POWER SYSTEM

The hydraulic tool power system, HyToPS, was originally designed to use fluid from the high pressure hybrid hydraulic system to power a variable-displacement motor. The hydraulic motor shaft drives fixed-displacement pumps, which was originally designed to be activated to provide flow to tools in up to three different modes. The first mode is a very high-pressure mode, at 10,000 psi, which can include powering emergency rescue tools such as those produced by Enerpac. The second mode is a medium-pressure, low-flow mode for small tools. The third mode adds the flow of a third pump to the flow from the pump of the second mode for a total maximum flow of 8 gpm at 2000 psi. While in the first mode, the second mode pump flows through the 10,000 psi pump, which is a check valve pump and smaller in displacement than the second mode pump. This serves to provide faster actuation of tools when they are under low load. When the load results in a pressure in excess of 2000 psi, the excess flow of the second mode pump is relieved and the 10,000 psi pump provides lower flow at pressures of up to 10,000 psi. In an attempt to simplify the design, however, and demonstrate basic capability, it was decided to complete the design for only performing one mode, a single 8 gpm 2000 psi load.

The combination of the variable displacement motor and fixed displacement pump uses a speed sensor on the shaft and the control system is intended to regulate the speed at a particular set point by adjusting the motor displacement. This system, in effect becomes a hydraulic transformer. Unfortunately there was very limited selection and availability of variable displacement motors that were capable of high pressure and which had electronic displacement control. The original choice was an Eaton aerospace motor rated for 5000 psi. Other electronically controlled motors were available in the size range, however, the pressure rating was less than 3000 psi. A motor was selected with an 11 cc/rev displacement and rated for 4600 psi manufactured by Rotary Power. However, this motor did not have an electronic displacement control actuator but a mechanical servo-actuator to control displacement. An electronic linear servo actuator was selected to be integrated into the system to control the displacement actuator. Figure 2.2-1 show the design of the mounting of the electronic servoactuator onto the variable displacement motor.

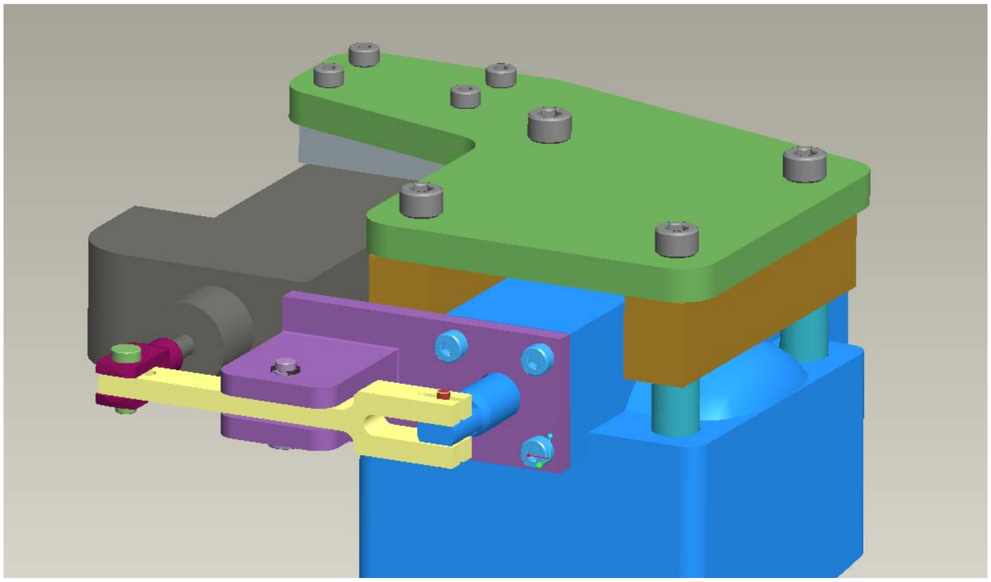

Figure 2.2-1. HyToPS Motor Displacement Control Actuator

2.3 COMPONENT PROCUREMENT

In the previous effort as part of the detailed design, a set of major components were identified that would need to be procured. All of these components were long lead time items. Where the component was available off-the-shelf or a simple modification of an existing product, a vendor procurement was initiated. Where the component was a custom fabrication or required a significant level of design interaction with the SwRI design team, a subcontract procurement was initiated. SwRI then developed a procurement package or specification for each of the components.

Table 2.3-1. Major Component Procurement

Description	Vendor/Subcontractor	Procurement Type
Hydraulic Drive Pump and Motor	Eaton Corporation	Subcontract
Hydraulic Switching Block (Megablock)	Eaton Corporation	Subcontract
Step-up Gearbox	Durst	Vendor Purchase
Hydraulic Accumulators	SwRI/Lincoln Composites	Vendor Purchase
Vehicle Control System (Micoautobox)	dSPACE	Vendor Purchase
Closed Center Steering Gear	RH Sheppard	Vendor Purchase

2.4 BASELINE VEHICLE TESTING

Southwest Research Institute selected a number of drive cycles for the vehicle to run to show the non-hybrid (baseline) and hybrid vehicle performance and fuel economies. Using M&S tools, SwRI analyzed military and commercial drive cycles to quantify baseline, hybrid hydraulic, and hybrid electric drive train performance and fuel economies[1]. The majority of the drive cycles selected were established and published drive cycles (see Table 2.4-1). However, several drive cycles will be created by SwRI to demonstrate the benefits gained from a hydraulic hybrid vehicle conversion.

[1] The M&S software employed in this effort included VPSET, MATLAB and Simulink.

The baseline vehicle will run several different drive cycles to establish reference performance and fuel economy numbers. The same drive cycles will be repeated after the hybrid hydraulic conversion for comparison. The drive cycles being considered are shown below.

Table 2.4-1. Selected Drive Cycles

Drive Cycles	Test Weights	Repeats
Central Business District (CBD)	Empty and Gross Vehicle Weight (GVW)	3
New York Bus	Empty and GVW	3
Highway Dynamometer Driving Schedule (HUDDS)	Empty and GVW	3
Heavy Heavy-Duty Diesel Truck (HHDDT) Creep Mode	Empty and GVW	3
Heavy Heavy-Duty Diesel Truck (HHDDT) Transient Mode	Empty and GVW	3
Orange County Bus	Empty and GVW	3
California Air Resources Board (CARB) Low Speed Cruise	Empty and GVW	3
1-minute Full Pedal Performance	Empty and GVW	3
5-minute Idle	Empty and GVW	1
SwRI Custom 1	Empty and GVW	3
SwRI Custom 2	Empty and GVW	3

Four of the drive cycles were selected or created by SwRI to specifically take advantage of the hybrid hydraulic vehicle conversion. Southwest Research Institute considered multiple drive train geometries including series, parallel and hydromechanical with energy storage. Additionally, SwRI examined operating modes including launch assist, brake energy recovery, dual mode braking (regenerative and service brakes), engine stop/start, silent watch mode, and stationary tool use mode. SwRI looked at ways in which the hybrid hydraulic system could augment ancillary vehicle systems to make the overall vehicle system more efficient, lighter weight, or increase mission usefulness.

As previously mentioned, several drive cycles were specifically chosen or created by SwRI: Full Pedal Performance, 5-Minute Idle, and SwRI Custom 1 and 2. As with all other drive cycles, the non-hybrid conventional vehicle performance will be compared to the hybridized vehicle performance. The anticipated performance and fuel economy gains from each of these drive cycles is listed below:

1. Full Pedal Performance – The full pedal performance drive cycle is a 1-minute run at full pedal. The run will demonstrate the power benefits from starting with a fully charged hydraulic system vs. a conventionally fueled vehicle at full throttle. The hybrid version of the drive cycle includes recharging the hydraulic system after the 1-minute, full throttle run.
2. 5-Minute Idle – As the name implies, the conventional vehicle engine will idle for the five minutes of testing. However, the control system designed for the hydraulic hybrid vehicle should not run the engine while the vehicle is not moving; *i.e.*, zero fuel burned in the hybrid vs. burned fuel for the conventional vehicle.
3. SwRI Custom 1 and 2 – These two drive cycles will be designed to take advantage of the immediate power available from the hybrid hydraulic system, the advantages gained from regenerative braking, and engine start/stop.

The benefits gains from these four drive cycles should be substantial based upon the advantages from using a hydraulic hybrid vehicle. There should also be gains when running the remaining drive cycles based upon vehicle weight savings, launch performance potential, and engine start/stop control strategy.

The benefits gains from these four drive cycles should be substantial based upon the advantages from using a hydraulic hybrid vehicle. There should also be gains when running the remaining drive cycles based upon vehicle weight savings, launch performance potential, and engine start/stop control strategy.

2.5 ALLISON TRANSMISSION MODIFICATION

2.5.1 Allison 3700 Series Transmission

In stock configuration, the FMTV vehicle uses an Allison 3700 series 7-Speed automatic transmission. The military version of the 3700 series Allison is a 246kW power rated unit, consisting of 7 forward gears and 1 reverse gear. The ratio spread of the 7-speeds range from

6.93:1 for first gear and to 0.78:1 for 7th gear. The output of the transmission is equipment with a power take off unit, as shown in Figure 2.5.1-1.

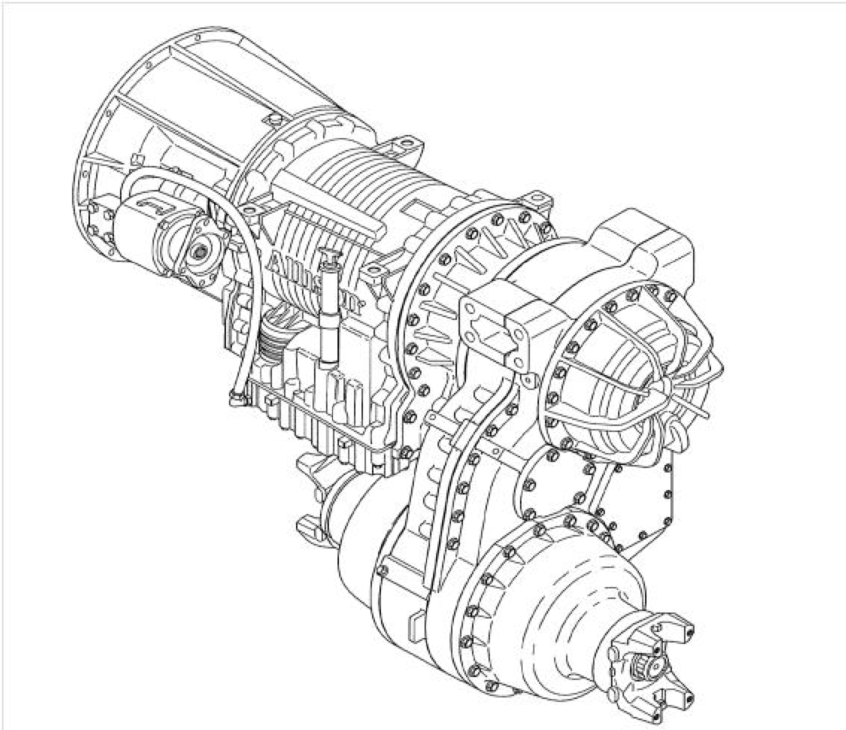

Figure 2.5.1-1. Allison 3700 Series Transmission

The Allison transmission was a key portion of the FMTV hybrid hydraulic integration. The ratio spread, torque capacity, and electronic vehicle integration made it an excellent choice for reuse on the program. The goals of the transmission were to provide an efficient power path for the prime mover hydraulic motors, grant flexible gear selection depending on driver demand, and allow for transmission braking and regenerative energy storage.

The overall objective of the following sections is to present the methodologies and discuss the results around integration of the Allison 3700 series transmission for the hybrid hydraulic program.

In order to use the transmission, several mechanical and electronic adaption's were made to ensure the transmission performed normally. In addition, once the adaptations were made, laboratory testing of the transmission occurred to validate the hardware and characterize the hydraulic system. The following sections discuss the design modification made to the transmission and the testing methodologies used to validate the adaption's prior to vehicle integration.

2.5.2 Design Modifications and Adaptations

In order to adapt the Allison transmission to the HHV, several modifications were made. These modifications were down to allow coupling of the input pump motor as the prime mover, reduce transmission parasitic losses, and allow for use of the production transmission control module (TCM).

The first modification made was removal of the torque converter. In the traditional configuration the torque converter is used as a fluid disconnect between the engine and drive train of the vehicle and allows the vehicle to launch from zero output speed. In the hybrid hydraulic configuration the input pump motors have capability to provide torque down to zero speed. Therefore, the torque converter could be removed which allowed for a reduction of unnecessary parasitic loss and inertia to the transmission. Figure 2.5.2-1 shows a cross section of the adapted transmission with torque converter and main pump removed.

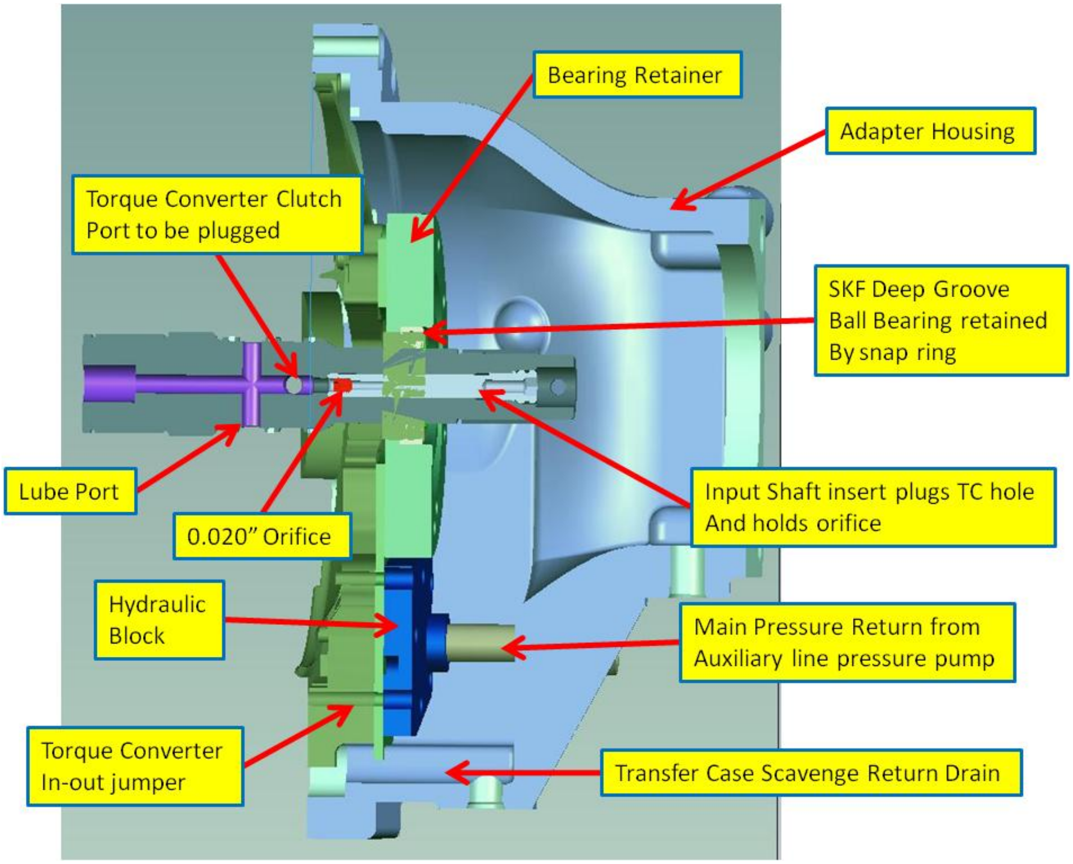

Figure 2.5.2-1. Adaption Cross Section

The second modification made was removal of the transmission main pressure pump. In the traditional configuration the main pressure pump serves two purposes: (1) provide high-pressure oil for clutch engagements and (2) provide high-flow oil for transmission lubrication and cooling. To better optimize these two needs, the single transmission pump was removed and two external auxiliary pumps were specified. This allowed for separate optimization of the pump power consumption to meet both high-pressure (low-flow) and high-flow (low-pressure) requirements.

The third modification made was redesign of transmission housing and turbine shaft to hydraulically orient and seal the transmission and provide attachment to the input pump motor. With the torque converter removed the turbine shaft oil passages which normally apply and release the converter clutch were no longer needed and therefore blocked. In addition, the independent circuits for main pressure and lube pressure were adapted to plumb in the auxiliary pumps. Figure 2.5.2-2 shows a image of the re-designed housing used for the transmission adaption.

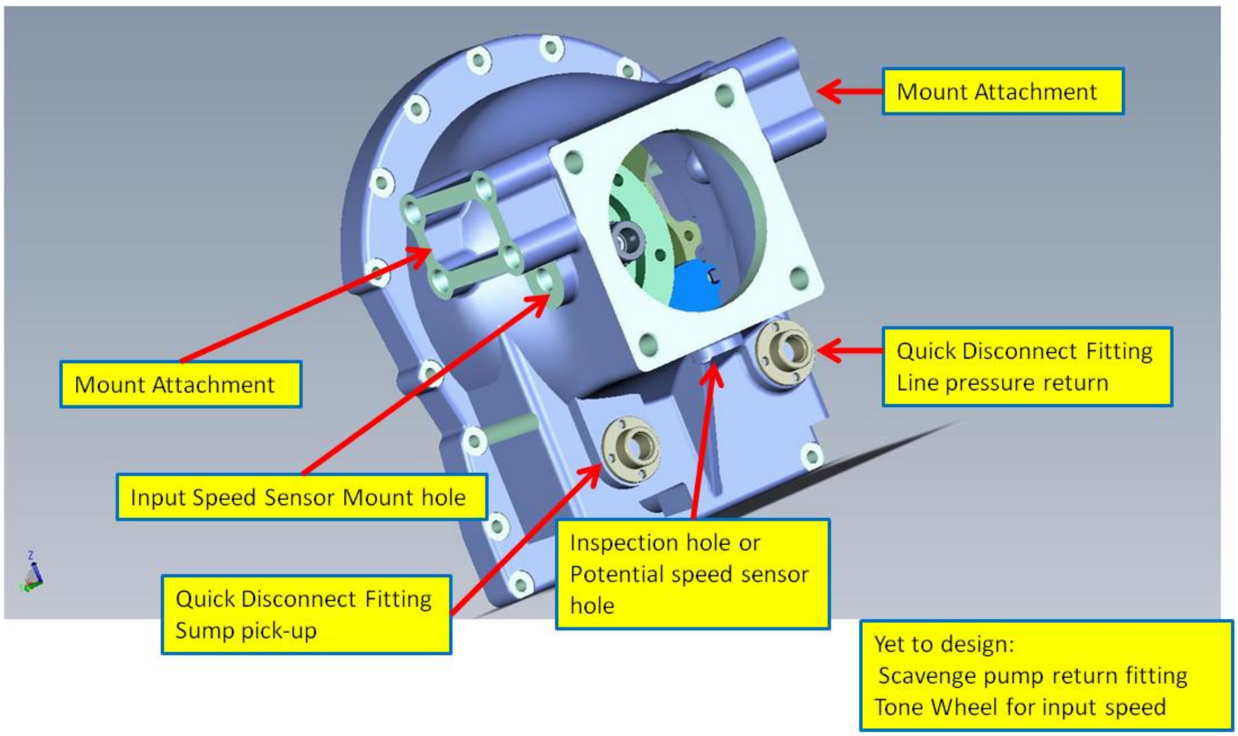

Figure 2.5.2-2. Adapter Housing Design

The last modification was design of a speed sensor tooth wheel. In the traditional configuration, the transmission uses an engine speed sensor which picks up a speed signal from marks on the outer shell of the torque converter. To mimic the speed profile, the tooth wheel was designed and installed to replicate this torque converter pattern. This modification allowed the transmission control module to properly read an engine speed signal and perform without diagnostic concern.

2.5.3 Test and Evaluation

Testing of the Allison 3700 7-speed automatic transmission was conducted on Test Stand #5 in Building 163 at SwRI campus in San Antonio, TX. The test stand was configured with a 330 kW motoring dynamometer. The transmission was mounted to a headstand with a torque meter installed between the hub of the dynamometer and the driveshaft connected to the headstand. The transmission was tested unloaded with the output PTO shafts disconnected. The test stand configuration used for the testing of the transmissions is shown in Figure 2.5.3-1.

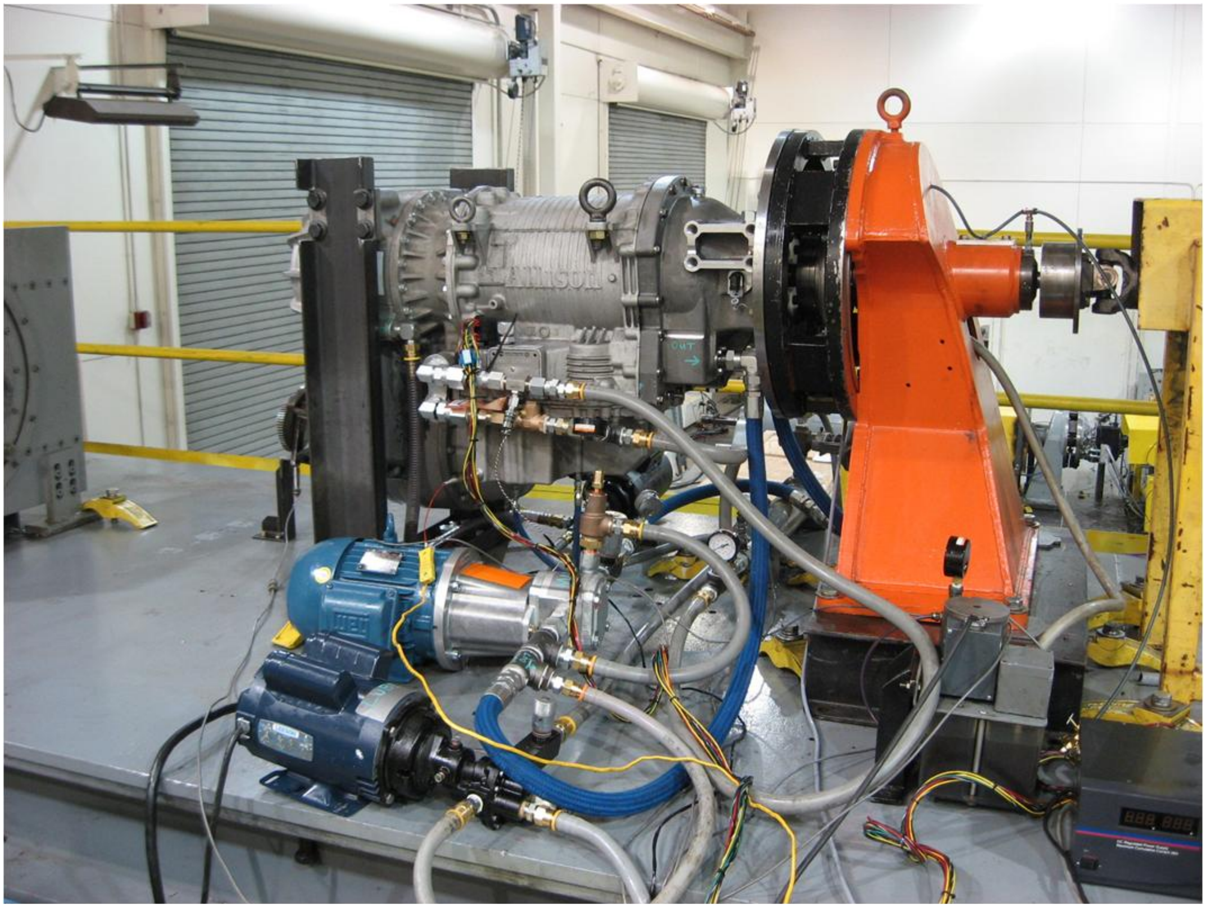

Figure 2.5.3-1. Transmission Testing

A National Instruments data acquisition and control system was used to record data and to control the test stand equipment. An HBM flange type torque meter was used to measure input torque. An optical encoder integral to the torque meter was used to measure input speed and a magnetic pickup speed sensor was used to measure output speed. Temperatures were measured with K-type thermocouples. During the testing, the following parameters were recorded:

Table 2.5.3-1. Instrumentation List

Sensor	Accuracy
Input Speed	+/- 1 RPM
Input Torque	+/- 3 NM
Output Speed	+/- 1 RPM
Sump Temperature	+/- 1 C
Lube Circuit Flow Rate	+/- 1 LPM
Main Circuit Flow Rate	+/- 1 LPM
Main Pressure	+/- 10 KPA
Lube Pressure	+/- 10 KPA

Testing of the Allison 3700 7-Speed transmission was divided up in to three phases to validate and characterize the mechanical adaptations. Phase I testing consisted of characterization and benchmarking of the transmission in its OEM production configuration (pre-modifications). This included running unloaded spin loss of the transmission at multiple speeds and temperatures, and measuring the parasitic drag torque losses, line pressure, lubrication pressure, and flow rates. Phase II testing included initial checkout and benchmarking of the updated transmission design prototype parts. This testing consisted of two main objectives. The first objective was to run the same testing completed Phase I to validate comparable functionality. The second objective was to optimized the main and lube pump power requirements through characterization of the hydraulics. Phase III testing purpose was to run the prototype transmission in a Hardware in the Loop (HIL) environment to validate shifting and pressure control functionality. Running in the HIL environment allowed for dynamic transmission shifting and evaluation of the transmission shift maps.

2.5.3.1 Phase I Testing Results

The first part of phase I testing involved parasitic torque loss measurement for all seven forward gears. Testing was done at 1 temperatures and 6 input speeds. In addition, testing was completed with torque converter unlocked and locked.

Below is a summary of the completed spin loss test conditions.

- 7 gears
- 6 input speeds: 700, 1000, 1500, 2000, 2500, 3000 rpm
- 1 transmission sump temperatures: 80 °C
- 2 torque converter states: unlocked, locked

Figure 2.5.3.1-1 shows the test results for the parasitic torque loss measurements. As typical for automatic transmissions, the higher gears have increased spin losses versus lower gears due to larger windage losses as output speed increases. Also, testing with an open torque converter (O) has increased losses due to slip through the fluid coupling versus locked (L).

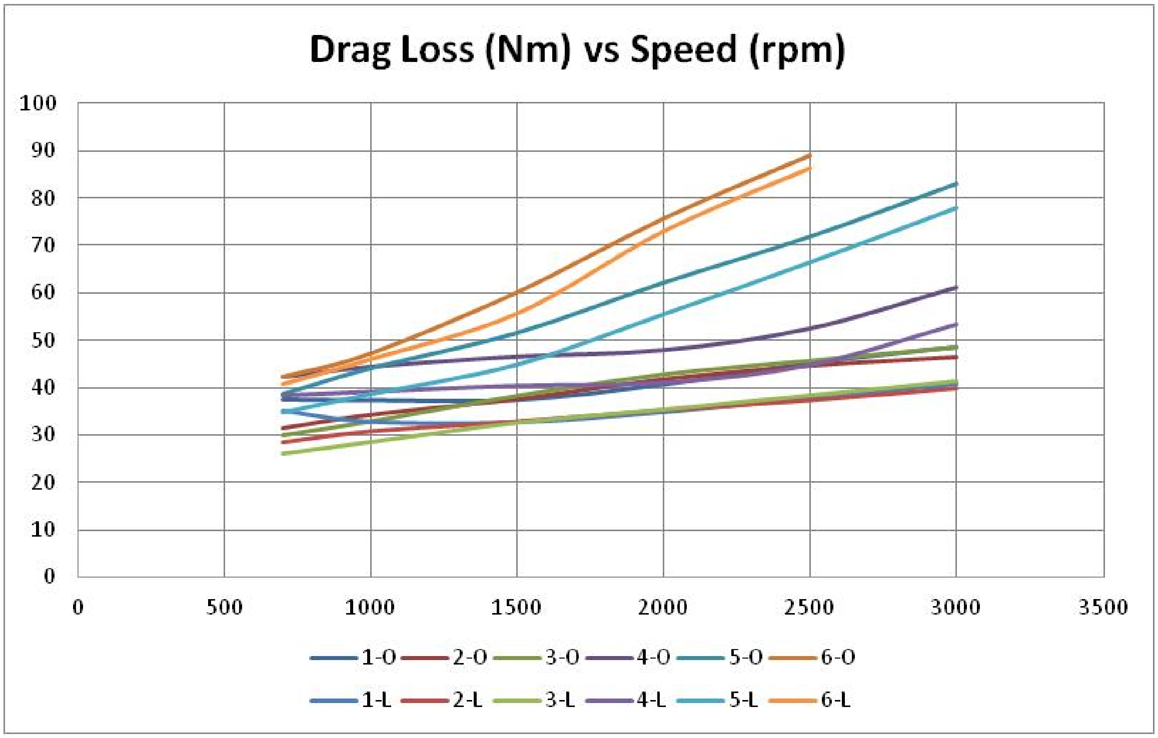

Figure 2.5.3-1. Transmission Testing

Figures 2.5.3.1-1 and 2.5.3.1-2 show the test results for the main (line) pressure and lube pressure measurements for all gears with both locked and open torque converter.

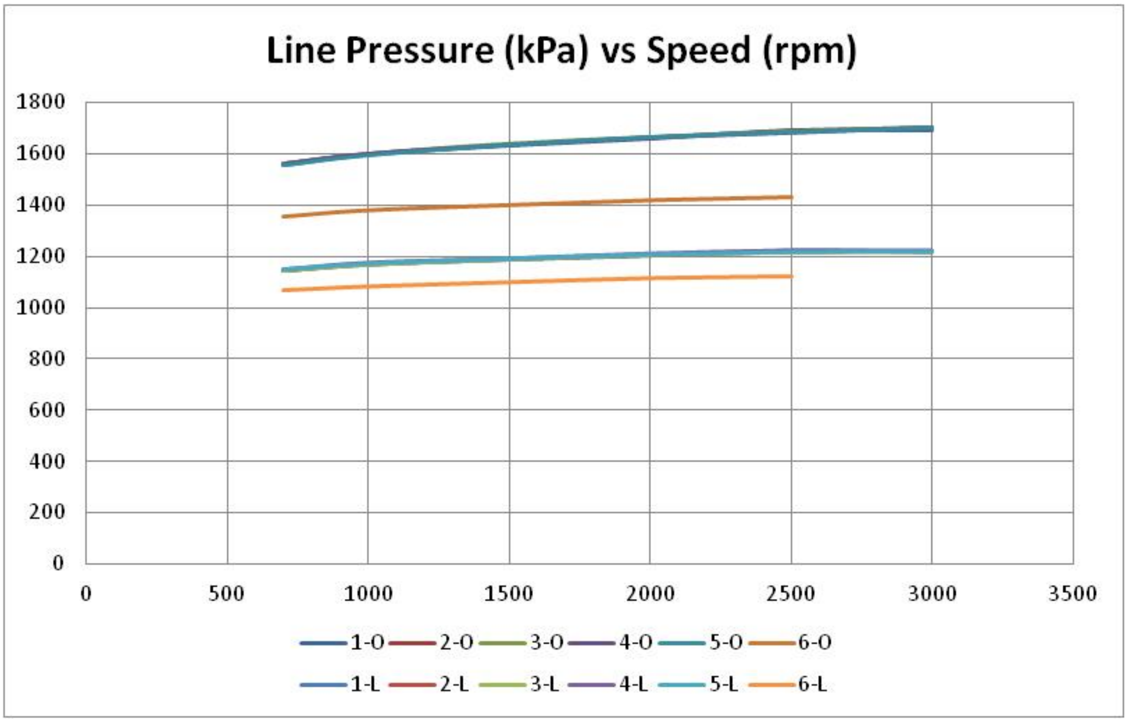

Figure 2.5.3.1-2. Transmission Testing

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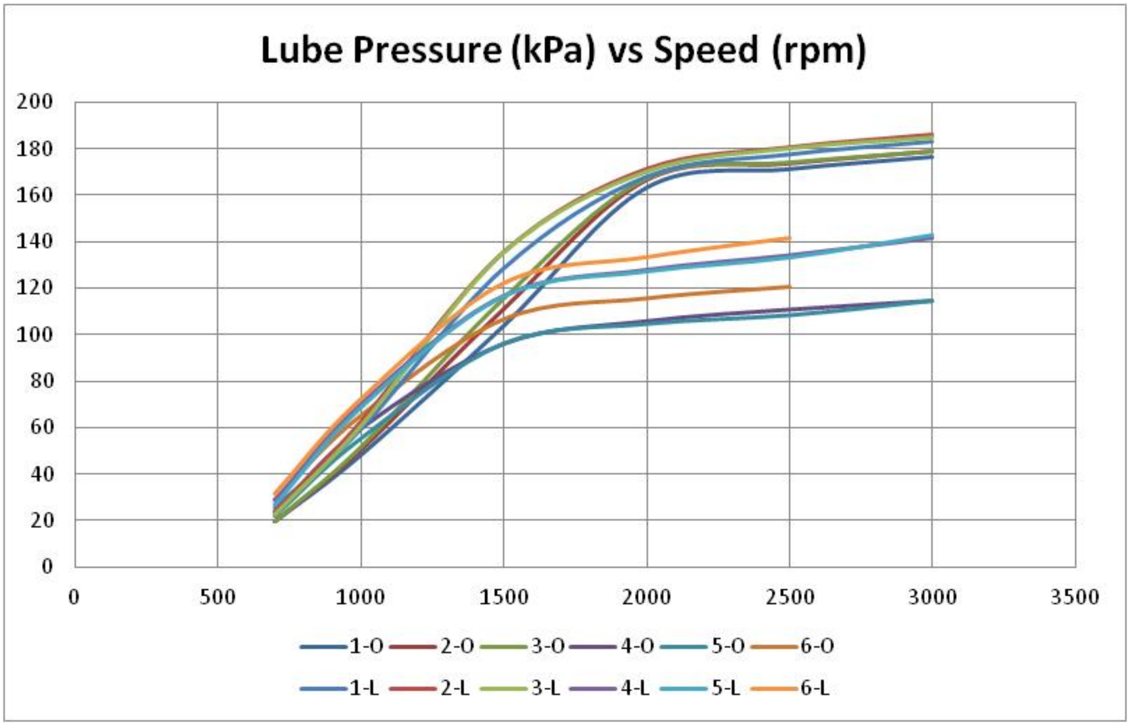

Figure 2.5.3-3. Transmission Testing

2.5.3.2 Phase II Testing Results

The first part of phase II testing involved general checkout of the design hardware modifications including monitoring of pressures, torques, gear states, flows, and visual inspection of the input shaft bearing for lubrication. Additionally, characterization testing was completed similar to Phase I testing to compare the two Allison configurations.

The second objective of Phase II was to determine minimum power consumption points for the lubrication and main pressure circuits. The minimum power point for the lubrication circuit was set just above the minimum flow to reach lube pressure regulation. The minimum power point for the main circuit was set just above the minimum pressure to hold clutch capacity, including during shifting events when clutches must be filled.

The results of the parasitic loss testing for the modified Allison design is shown in Figure 2.5.3.2-1. As shown, the overall drag losses were improved with the new design. This is primarily a result of the removal of the transmission pump and torque converter.

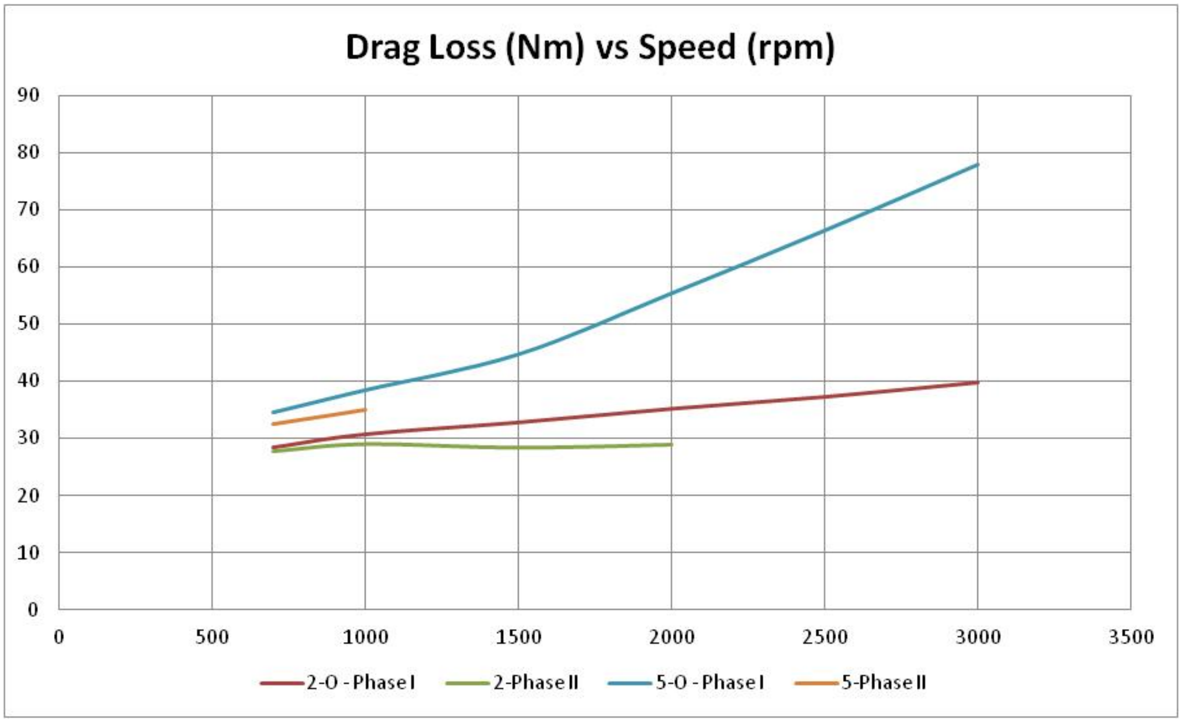

Figure 2.5.3.2-1. Transmission Testing

Figures 2.5.3.2-2 and 2.5.3.2-3 show the results of the main (line) pressure and lube pressure testing for the modified Allison unit. In comparison with the OEM production unit, the main (line) and lube pressure circuits were validated to yield comparable pressures and flows.

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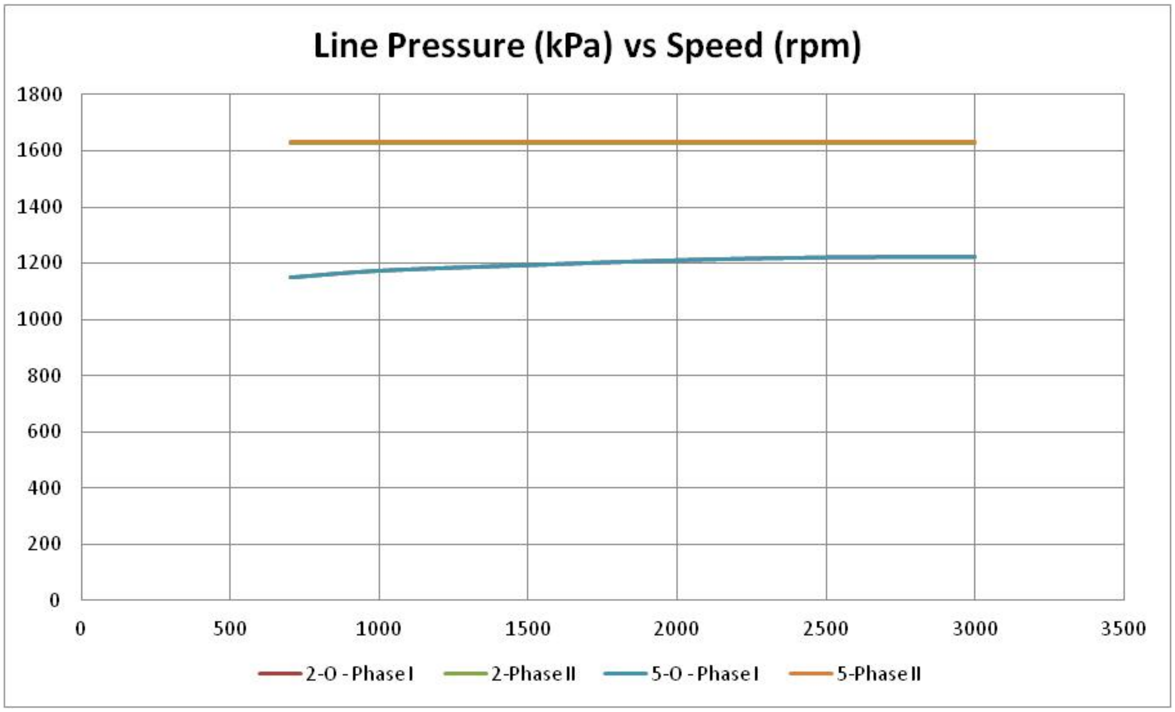

Figure 2.5.3.2-3. Transmission Testing

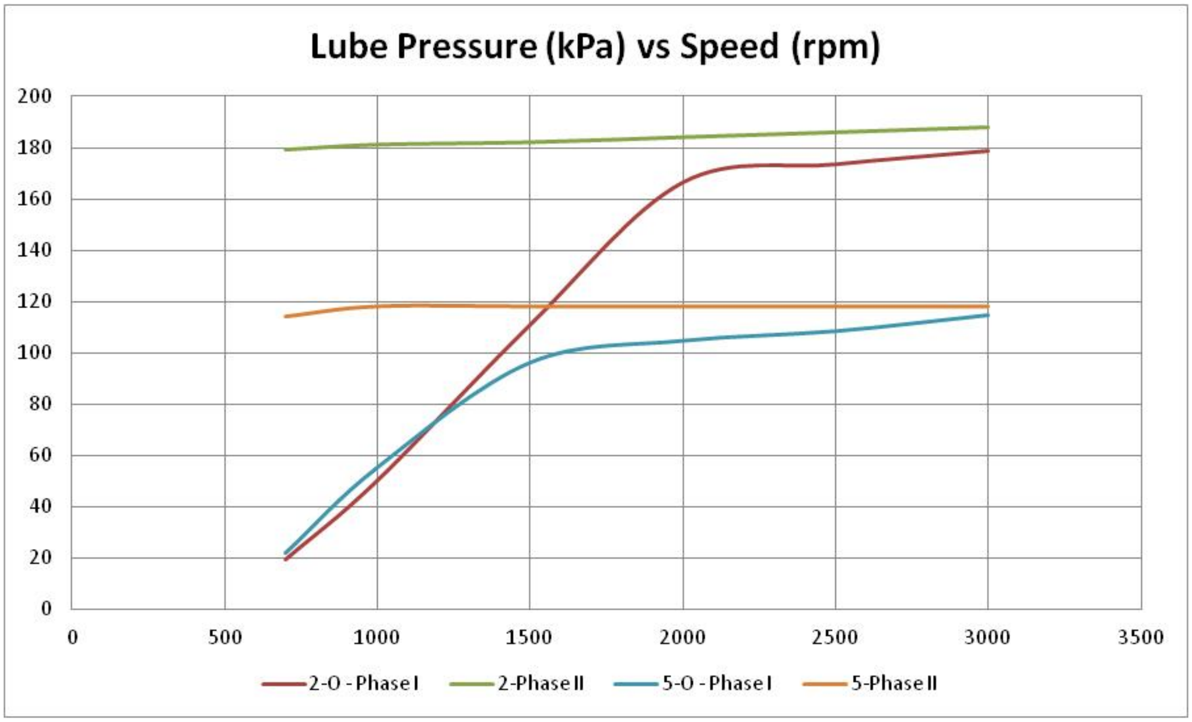

Figure 2.5.3.2-4. Transmission Testing

2.5.3.3 Phase III Testing Results

Phase III testing consisted of fully integrated HIL testing to validate the complete system was functional before vehicle installation. This testing consisted of using the vehicle electronic shift panel, the production transmission control unit (TCM), and software to simulate the vehicle CAN traffic. Figure 2.5.3.3-1 shows a series of full throttle upshots completed on the test stand to validate the hydraulic functionality as a complete system.

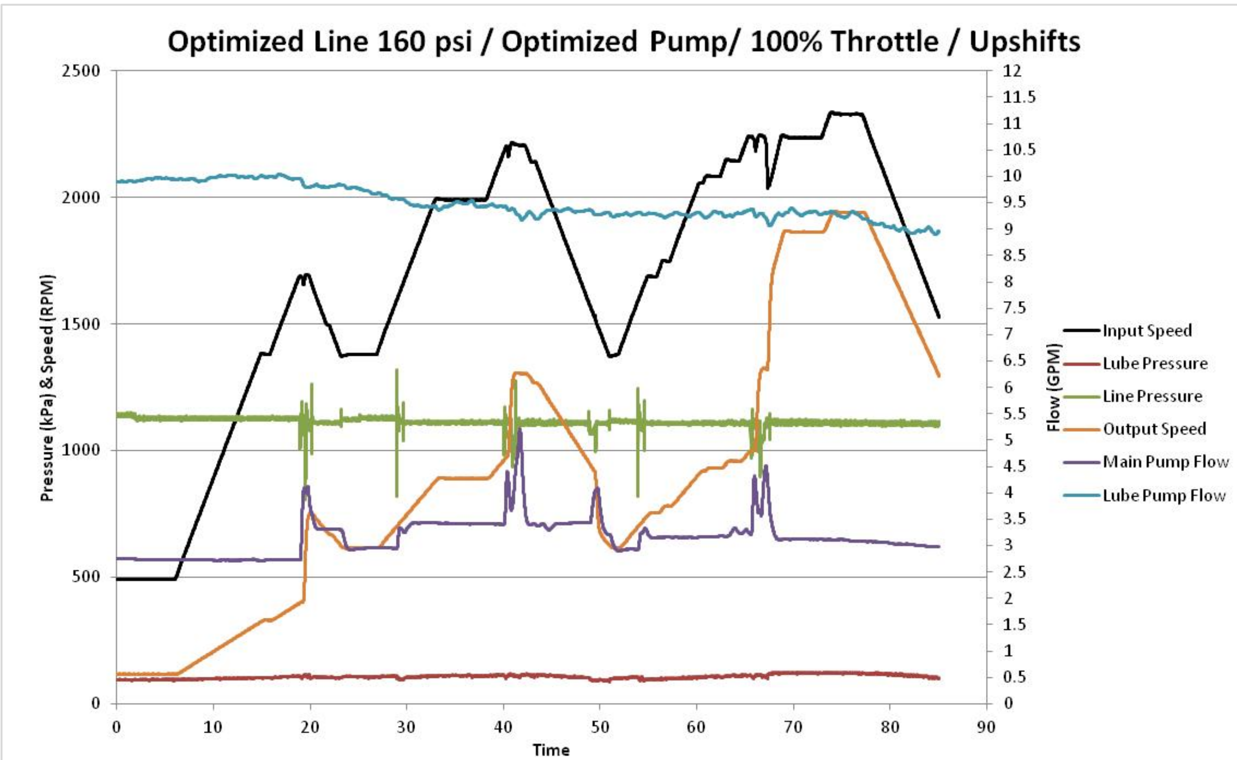

Figure 2.5.3.3-1. Transmission Testing

2.6 CONTROL SYSTEM DEVELOPMENT

The control development was conducted in the following stages:

1. Measurements and parameters from HHV components (e.g. engine curves, pump efficiency, etc) were used to create a Matlab-Simulink model of the stock and HHV-FMTV using SwRI’s internal vehicle simulator tool (VPSET/ RAPTOR)
2. Simulations were run to identify improvements and the capability of the HHV to reduce fuel consumption over the cycles we ran the stock FMTV on the chassis dynamometer

3. The various controller-controller interactions of the stock FMTV were studied to be able to design the supervisory controller to control them on the HHV-FMTV
4. The supervisory controller was implemented in Matlab-Simulink using dSPACE microAutobox rapid-prototyping platform. The control algorithms developed and tested in simulation were the backbone for the supervisory control logic.
5. A physical model based approach makes it easy to adapt to modifications in the hardware or configuration of the HHV.

2.6.1 Supervisory Control Implementation

This section defines the various modes of operation of the hydraulic series hybrid . It also covers the implementation of the supervisory control algorithm, its calibration and real world implementation issues and fixes.

The supervisory controller comprises of 9 modes of operation and during operation, the mode transition logic made sure it is in the appropriate mode to perform various functions as a HHV. This section discusses the modes and the sub-modes of operation which gives a high-level architecture for the controls. This is followed by description of the modular control logic embedded within the modes which is responsible for sub-system level control of the HHV-FMTV.

2.6.2 Mode Selection

This section describes the mode transition implemented within the supervisory controller of the series hydraulic hybrid FMTV.

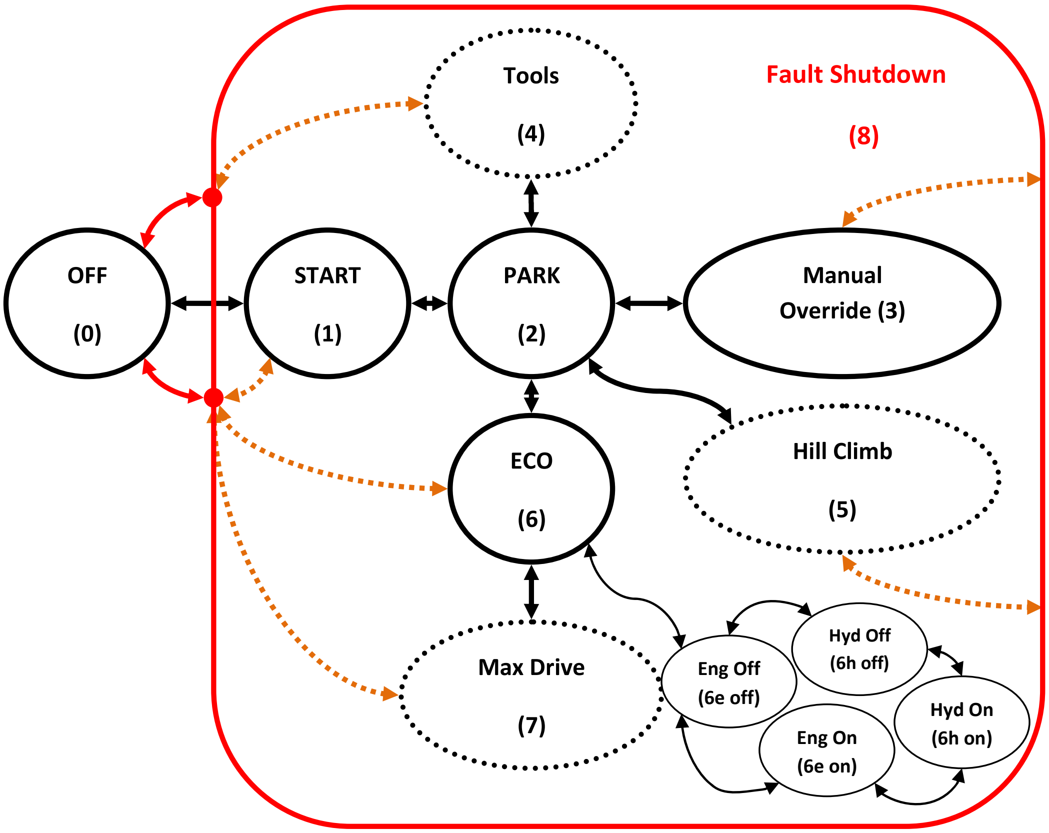

Figure 2.6.2-1. Mode transition schematic

If automatic mode is selected, the following mode switching logic is used. If manual mode selection is active, the user can fix the desired mode. The Mode transition in Figure 2.6.2-1 gives an overall flow between modes of operation and is further detailed in Table 2.6.2-1.

Table 2.6.2-1. Operating Mode Description

Current Mode		Description	Next Mode		Mode switching triggers	Transition
0	**OFF**	System is in Off state waiting for IGN to be turned On	1	Startup	If IGN key is On	
			0	OFF	None of above triggers	Reset Integrator
1	**Startup**	IGN is On and the user still hasn’t started the vehicle electrically. This happens only from IGN off when electric start is needed	8	Fault Shutdown	If active faults are triggered	Reset Integrator
			3	Manual Ovrd	If manual override is active	Reset Integrator
			2	Park	If IGN is On & Engine speed > threshold RPM	Reset Integrator
			0	Off	If IGN is turned Off	Reset Integrator
			1	Startup	None of above triggers	

Current Mode		Description	Next Mode		Mode switching triggers	Transition
2	**Park**	This state exists if Park gear exists in the transmission. For this FMTV, the gear goes into N on startup	8	Fault Shutdown	If active faults are triggered	Reset Integrator
			4	Tools	If Tools switch is selected	Reset Integrator
			5	Hill Climb	If (1) Lo gear is selected	Reset Integrator
			6	Efficiency	If (R)reverse or (2-7)higher gear is selected	Reset Integrator
			2	Park	If (N) Neutral gear is selected	
3	**Manual Override**	In each of the operating modes (4,5,6,7) we can individually select a manual override to take control of a specific sub-module (Eg. Charge pump control)	8	Fault Shutdown	If active faults are triggered	Reset Integrator
			1	Startup	If manual override is not active	Reset Integrator
			3	Manual Override	If None of above triggers	Reset Integrator
4	**Tools**	This mode is used to control HYTOPS system for operating hydraulic tools. It has not been implemented in this project	8	Fault Shutdown	If active faults are triggered	Reset Integrator
			2	Park	If Tools switch is not active	Reset Integrator
			4	Tools	If none of above triggers	
5	**Hill Climb**	This mode is used to operate the FMTV during hill climbing. It has not been implemented in this project	8	Fault Shutdown	If active faults are triggered	Reset Integrator
			6	Efficiency	If N, R or (2+) gear is selected	Reset Integrator
			2	Park	If Park is selected & If vehicle has Stopped	Reset Integrator
			5	Hill Climb	If none of above triggers	
6	**Efficiency**	This is the normal operating mode for the hydraulic hybrid FMTV	8	Fault Shutdown	If active faults are triggered	Reset Integrator
			2	Park	If Park is selected & If vehicle has Stopped	Reset Integrator
			5	Hill Climb	If (1) Lo gear is selected	Reset Integrator
			7	Max Drive	If high acceleration is demanded by the user	
			6	Efficiency	If none of above triggers	
7	**Max Drive**	This mode is used to operate the FMTV during high power operation like high accelerations. It has not been implemented in this project	8	Fault Shutdown	If active faults are triggered	Reset Integrator
			2	Park	If Park is selected & If vehicle has Stopped	Reset Integrator
			5	Hill Climb	If (1) Lo gear is selected	Reset Integrator
			6	Efficiency	If acceleration demanded by the user falls below a certain threshold	
			7	Max Drive	If none of above triggers	

Current Mode		Description	Next Mode		Mode switching triggers	Transition
8	**Fault Shutdown**	The hydraulic sub-system is shut down in this mode of operation. Supervisory controller continues to monitor the system and data logging is kept ON	8	Fault Shutdown	Stay in Fault shutdown mode until manual Mode reset is initiated by the user. The hydraulic sub-system is shut down in this mode of operation. microAutoBox continues to monitor the system and data logging is kept ON	Reset Integrator

The Figure 2.6.2-2 shows the Simulink implementation of the mode selection described above.

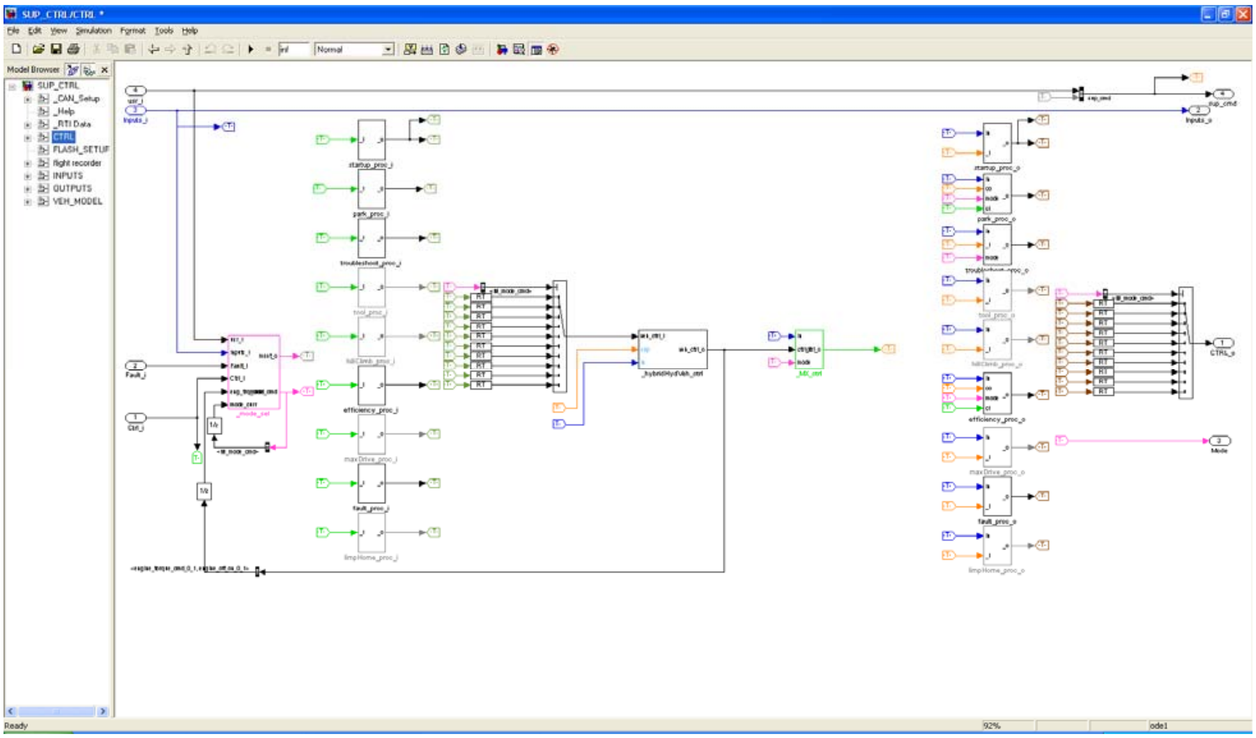

Figure 2.6.2-2. Mode transition schematic

2.6.3 Standby Engine and Hydraulic Stop-Start

Standby engine Stop (Start)

A crucial feature of the series hydraulic hybrid configuration is the auto-Stop-Start of the engine. The engine does not provide the motive force directly and it's only function in this configuration is to fill up the hiP accumulator and pneumatic accumulator (for service brake). If the two accumulators have sufficient energy (pressure) to let the user operate the vehicle (steering, driving, braking, transmission, etc) the engine can be safely turned off.

Table 2.6.3-1. Standby Stop-Start Description

Next Mode		Description	Current Mode		Entry Trigger from Mode 6	Transition
6e Off	**Eng Stop**	Engine is stopped by turning OFF the FMTV ECM. All other vehicle controllers are kept On for hydraulic only vehicle operation	6	**Efficiency**	When the pressure in the hi P accumulator goes beyond the target Pressure at the operating speed and mode, the engine is turned Off	Reset Integrator
6h Off	**hydSys Stop**	This turns off the charge pump too to conserve hi P accumulator pressure	6	**Efficiency**	If hi P falls below certain safe threshold & If vehicle is stopped If the reservoir level is between safe levels & If user selects N gear(to avoid transmission errors) If engine is turned Off by ‘engStop’	Reset Integrator
6h On	**hydSys Start**	This turns the charge pump On	6	**Efficiency**	If hi P falls below certain safe threshold OR Any of the above conditions cease to exist	Reset Integrator
6e On	**hydEng Start**	Engine is hydraulically started by the Pump in Motor mode	6	**Efficiency**	When the pressure in the hi P accumulator goes below the target Pressure at the operating speed and mode or the vehicle speed is above a certain threshold	Reset Integrator

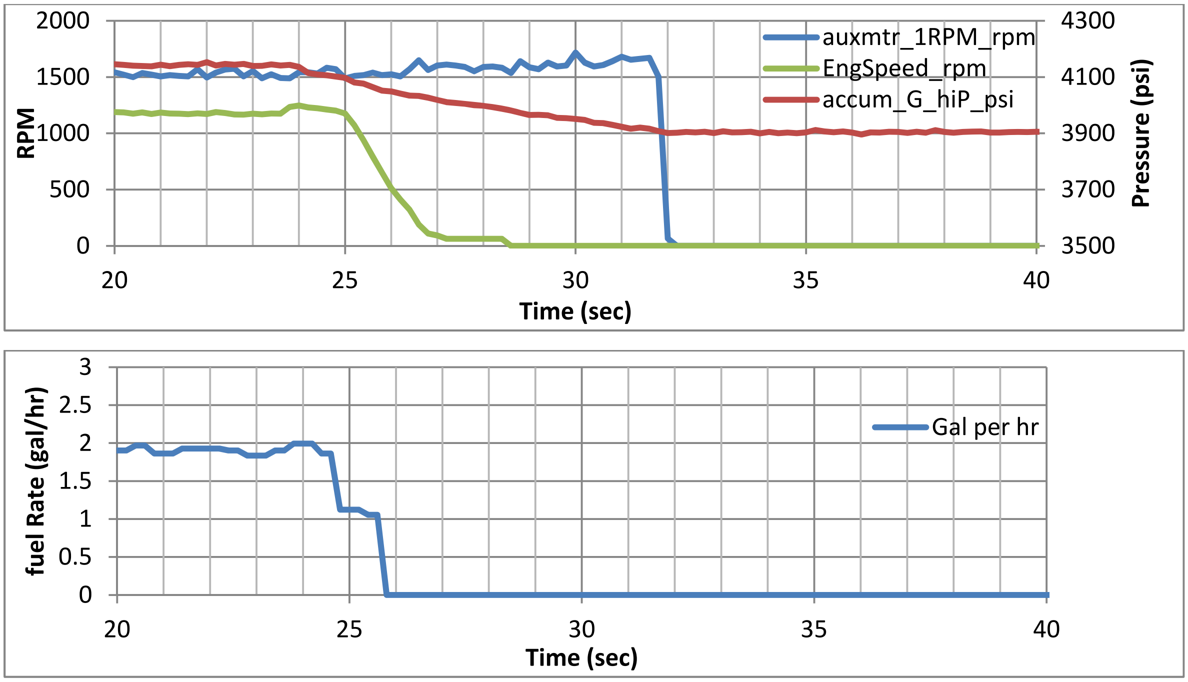

Figure 2.6.3-1. Standby Fuel Consumption

As shown in Figure 2.6.3-1, we see that when the vehicle is stationary, to maintain a pressure of 4200 psi, the engine consumes around 1.9 gallons/ hr. The reason is evident from the reduction in hiP accumulator pressure once the engine is turned off. The reason for this is the leakages (which increase with pressure) and draw from auxiliary motor used to provide circulation of hydraulic fluid between the LoP accumulator and the reservoir, to provide cooling, lubrication and line pressure and scavenging for the transmission.

The engine can be stopped to reduce fuel consumed during vehicle stoppages or if the pressure is sufficiently high. If the auxiliary motor is not turned off, the engine will have to be restarted often. The penalty of starting the engine hydraulically is also to be considered while stopping the engine. For the existing system, the fuel-equivalent of pressure consumed to start the engine hydraulically is the same as 3 seconds of engine running to maintain a nominal pressure of 4200 psi.

If automatic mode is selected, the following mode switching logic is used. If manual mode selection is active, the user can fix the desired mode. The mode transition in Figure 2.6.2-1 gives an overall flow between modes of operation and Table 2.6.3-1 describes the trigger conditions for the Engine Stop-Start modes of operation.

Standby Hydraulic Stop (Start)

In addition to the engine stoppage, the auxiliary motor also has to be stopped to reduce the standby energy loss when not being used (For e.g. vehicle stops). This can be allowed only if the system is at a relatively lower pressure (to avoid system startup at a higher pressure).

Standby Stop-Start Sequence

Referring to the Figure 2.6.3-2(Standby Stop-Start sequence), the standby sequence of events can be summarized as follows:

1. Standby engine Stop: The engine is turned off after the vehicle stops. This is indicated by EngSpeed_rpm going to 0 around 25 seconds
2. Standby hydraulic Stop: The user selects Neutral gear at 30 seconds. This triggers the hydraulic Stop mode which turns off the auxiliary motor. This is indicated by auxmtr_1RPM_rpm going to 0 following the gear change to Neutral (TransCurrentGear goes to 0 around 32 seconds)
3. Standby hydraulic Start: The user indicates change from Neutral just before 110 seconds. This triggers the hydraulic. This starts the auxiliary motor (indicated by auxmtr_1RPM_rpm reaching around 1500 rpm at 110 seconds). The gear change is inhibited until the user presses the gear selector again to make sure the line pressure is applied before shifting from Neutral
4. The gear shifts into 2nd gear (TransCurrentGear goes from 0 to 2 around 118 seconds)
5. Standby engine Start: The falling hiP accumulator pressure triggers the engine to start hydraulically which happens around 125 seconds (indicated by EPopmode going briefly to ‘Motor mode’ and applying a starting torque to the engine to start it hydraulically followed by the engine rpm going to Idle speed of 700 around 127 seconds)
6. The pressure control then kicks in just after 130 seconds to reach a higher target pressure

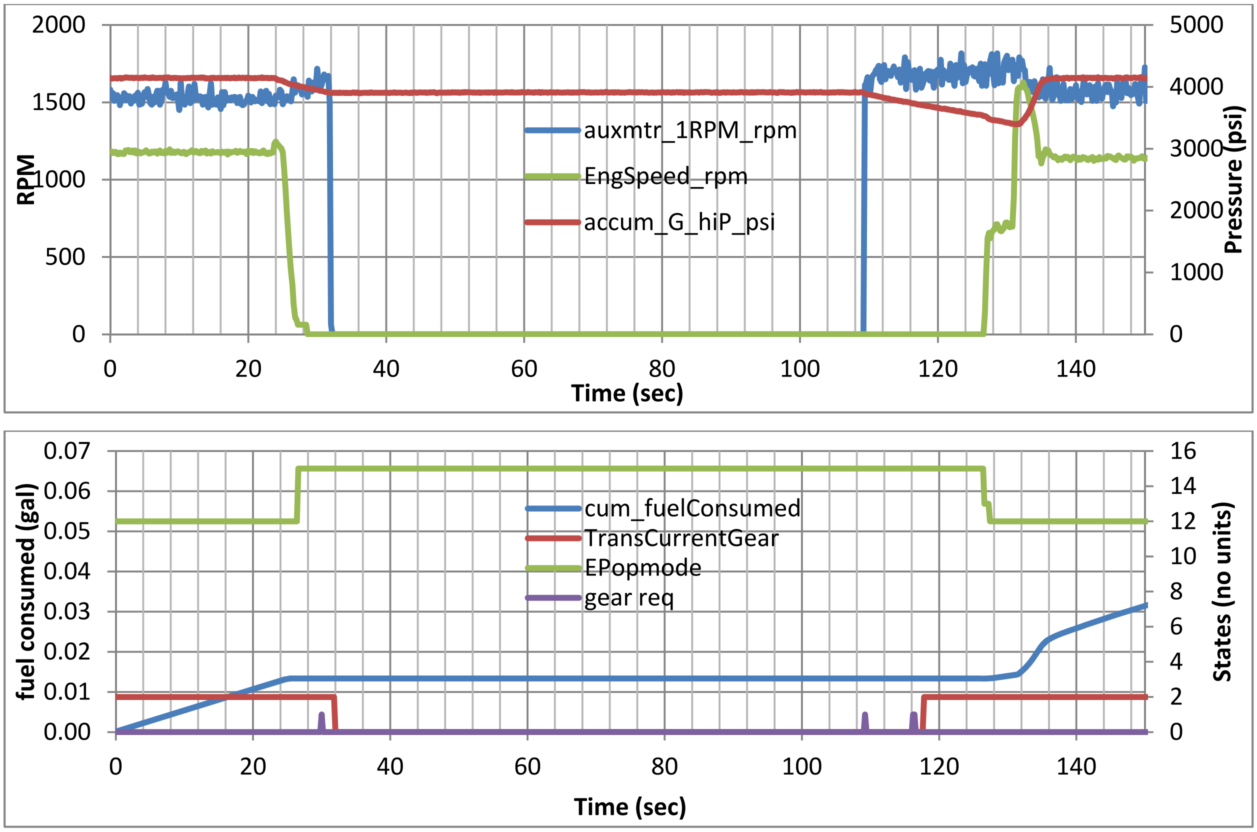

Figure 2.6.3-2. Standby Stop-Start Sequence

2.6.4 Control Layout and Real-World Implementation Issues

The supervisory controller has a modular architecture with each having its own manual override modes:

1. Hydraulic start
2. Engine control
3. Hydraulic pump control
4. ACV control
5. Auxiliary hydraulic motor and reservoir level control:
6. Hydraulic traction Motor control
7. Transmission control
8. Power distribution

Hydraulic Engine Start:

Function

This function is triggered within Efficiency mode and it takes control over the engine and pump to start the engine hydraulically.

Calibration and Implementation Issues

Calibrating hydraulic engine start involves studying the starting torque to overcome engine start inertia, the torque profile to efficiently and successfully start the engine each time and the lowest HiP pressure at which we can provide the starting torque.

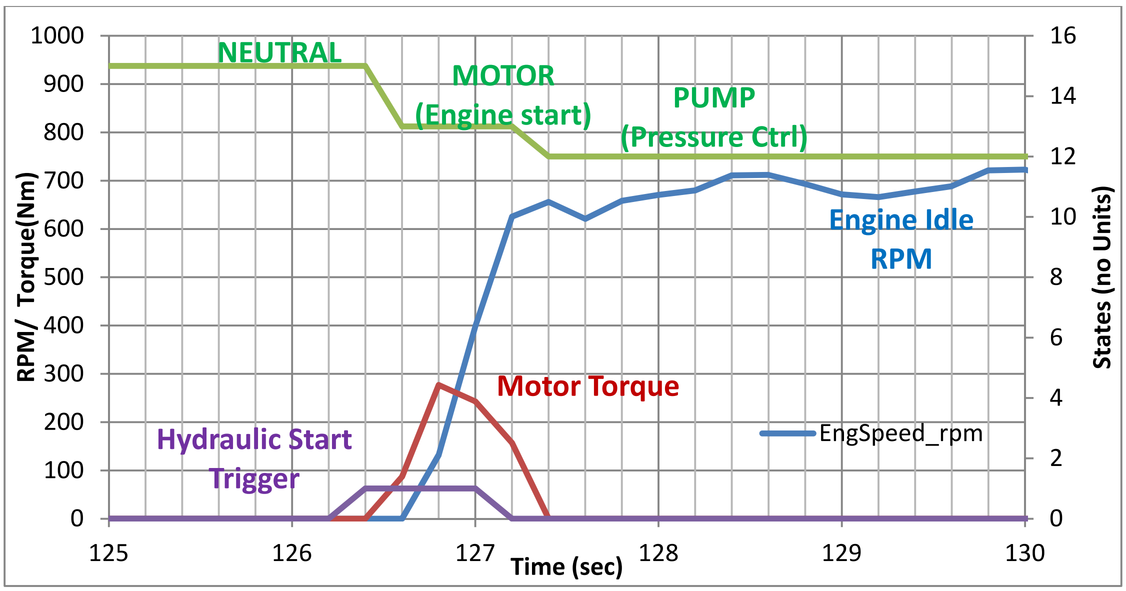

Figure 2.6.4-1. Hydraulic Engine Start

The starting torque is tapered down with increasing Engine speed such that no motoring torque is applied beyond the idle speed as shown in Figure 2.6.4-1.

Engine controller:

Function

The engine in this FMTV is a CAT C7 engine with the ECM in engine speed control mode. The signal from the accelerator pedal is used by the engine controller (ECM) to set the speed of the engine. In the case of the FMTV in a series hybrid configuration, the user's accelerator pedal command is not tied to the engine as the engine is not directly driving the wheels. Instead, the engine provides the torque to drive the hydraulic pump to charge the HiP accumulator. The pressure control provides a pedal signal based on the optimum engine speed needed to provide torque to the pump.

Calibration and Implementation Issues

Fake pedal to ECM: The pedal signal in the case of the FMTV is a signal which maps the position of the accelerator pedal into a PWM duty cycle at a fixed frequency. This fake pedal signal is then given by the supervisory controller to the engine to achieve pressure control.

Noise and vibration harshness (NVH): The torque applied to the engine is a function of engine speed and it diminishes to zero below 1000 rpm. This ensures that the engine is always operated for pressure control in its most efficient window and reduces NVH.

Safety: The low speed zero pump-torque also ensures that the engine doesn’t stall or is spun backwards.

Hydraulic pump controller:

Function

The pressure control is responsible to maintain the accumulator pressure at a target based on the mode of operation (hill climb, efficiency, etc) ,vehicle speed and road demand and other predictive algorithms beyond the scope of this project. For this proof of concept FMTV, the desired pressure set point is based on vehicle speed. A PI controller tries to keep the pressure error to a minimum. The output of this PI controller is a desired engine power that should be applied to the hydraulic pump which pumps fluid into the HiP accumulator until the error diminishes to zero. Using the engine’s optimum speed-power curve, an optimum speed is selected and commanded from the engine controller.

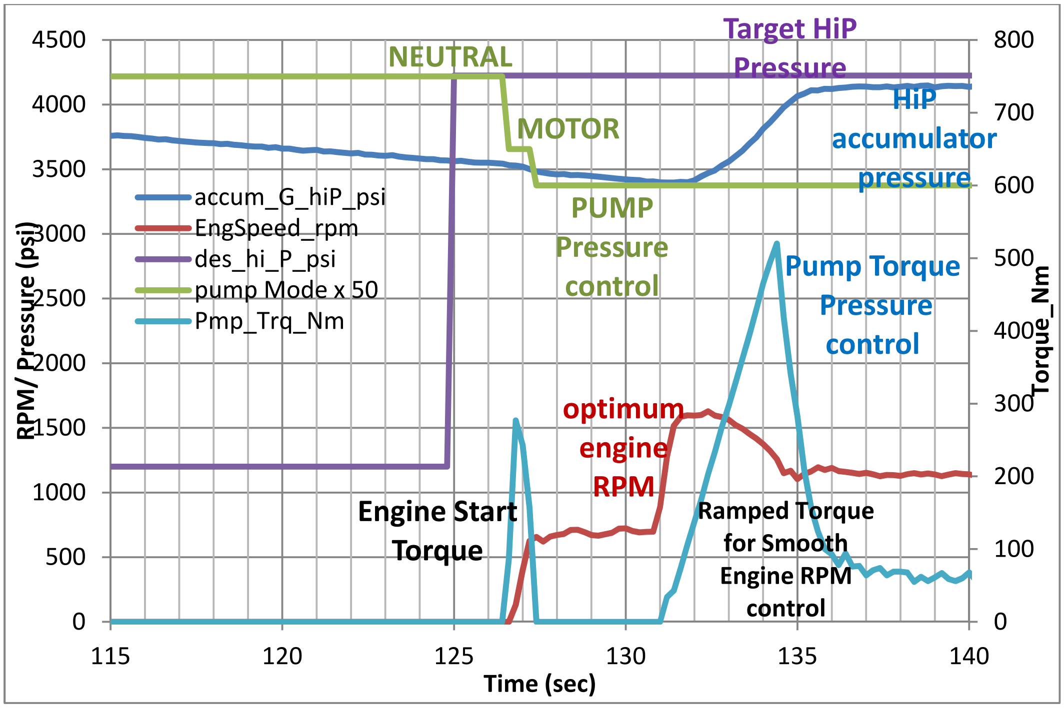

Figure 2.6.4-2. Pressure Control

Calibration and Implementation Issues

Energy balance: To prevent pressure drain during high power operations (E.g. high speed accelerations), the pump power (accumulator fill-up) command is kept higher than the motor power (accumulator drain) taking into account other losses in the system.

Speed control: The engine, being in speed control mode, sees abrupt torques applied to it as control disturbances. The resulting oscillatory response can be damped by ramping up the pump torque applied to the engine (as shown in Figure 2.6.4-2). The pump torque up-ramp is calibrated such that the oscillations are minimal and yet not too slow to keep with the demands of the test cycles.

Pump direction: One important issue is to verify the direction of the hydraulic pump. Most hydraulic pump-motors are reversible with minimal consequences. But rotating the transmission or the engine opposed to its design can be detrimental. In the case of the series hydraulic hybrid FMTV, the pump direction was to be reversed (pump starts the engine motor and motor mode pumps fluid into the accumulator)

ACV controller

Function

This controls the ACV valve connecting the HiP accumulator with the hydraulic circuit. Closing this completely seals the HiP accumulator from the hydraulic circuit. When open, the motor and pump can use the accumulator pressure as a buffer. The ACV is kept on as long as the system is operational. It is turned off only during faults (mode 8), Off state (mode 0) or standby hydraulic off (mode 6h-off).

Auxiliary hydraulic motor and reservoir level controller

Function

This controls the valve that runs the auxiliary motor which then drives the charge pump, transmission line and lube and scavenging pump. This also controls the bypass valve which then controls the flow from the reservoir to the loP accumulator through the filter and cooling loop. The auxiliary motor ideally should be separated into multiple motor-pump loops to isolate the charge, transmission line and lubrication and scavenging circuit. The motors should also be having variable speed control to be able to optimize the energy lost in carrying out these essential functions. In the current implementation, auxiliary motor control is on during vehicle operation. It is turned off during modes 6h-off to conserve energy.

Calibration and Implementation Issues

Auxiliary motor RPM: The calibration is a manual flow control valve which adjusts the flow rate (and speed) of the fixed motor to be able to respond to the normal operational needs of components connected to it. It shouldn't be too high so as to cause excessive losses.

Reservoir charge-discharge levels: The hydraulic fluid is shared between the accumulators, components and mainly the reservoir. The level is set such that fluid between the HiP and LoP is always around a threshold. Higher fluid mass between HiP and LoP accumulators will mean the accumulators will operate at higher pressures which could lead to safety disk rupturing. On the other hand, lower fluid mass will result in pressures lower than the charge pressures of the Nitrogen bladders within the accumulators running the risk of bladders closing off the accumulator stem-valve.

Hydraulic traction Motor controller

Function

The supervisory controller converts the driver's accelerator and brake command into a road demand torque. This along with the gear ratio of the transmission results in a torque to be applied to the traction motor. During accelerations, the motor is set to be in motor mode and is set to pump to provide braking torque. The pumping depends on the HiP accumulator pressure and once it reaches a higher threshold, it tapers down the pumping to protect the accumulator. It also makes sure the mode transition of the pump (pump-neutral-motor and vice-versa) happens gracefully. The motor controller also communicates with the transmission controller to read the current gear and 'shift in-progress' signal. On detecting a shift or a Neutral, torque to the motor is cut to prevent motor over speed.

Calibration and Implementation Issues

During gear shifts, the motor torque is cut to a low level to prevent motor over speeds. When the torque is reapplied to the transmission in the next gear, it causes a drivetrain NVH issue which can be mitigated by ramping up the motor torque. This ramp rate should be sufficient to reduce the drive train vibrations but not too sluggish to affect drivability.

Transmission controller

Functions

The transmission controller reads in the user's command from the electronic shifter and sends out a command to the transmission to optimize shift points. The optimum shift points are based on the required performance over the drive cycle as well as motor's efficiency map. UT's optimized transmission shift selection logic was also implemented as a part of the supervisory transmission controller. In addition to the shift points, the transmission supervisory control logic plays an important role in the standby stop-start feature that is crucial to increasing fuel efficiency of the HHV. Please see Section 'Standby Engine and Hydraulic Stop-Start' for more details.

Calibration and Implementation Issues

The transmission control was the major challenge due to the fact that the TCM relies on the many other components such as:

1. ECM to receive accelerator position. This is used to optimize shift points as well as to prevent TCM from going into fault
2. Gear shifter to receive shift commands. This has to be faked by the supervisory controller to optimize shifts
3. Body controller for 'Reverse Inhibit' signal to prevent reverse
4. Torque convertor speed signal which senses if the engine is turned off. If so, it goes into Neutral. This feature is undesirable in a series hydraulic hybrid configuration as it would prevent pure-hydraulic launch.

These functions were implemented within the supervisory controller (both software and hardware) to make sure the TCM responded appropriately.

Power distribution logic

This controls the power relays to ensure a graceful staged-shutdown occurs during fault shutdown. In case of a fault, it turns off power to the hydraulic pump and motor first while keeping power to itself on via a keep-alive relay. Once the user diagnoses the fault, he can then turn off power to the supervisory controller hardware.

2.6.5 Diagnostics

Diagnostics within the supervisory controller for the HHV can be broadly divided into sensor-level-Out of range failure detection and logic-level-In range failure detection. There were primary and secondary conditional faults too. Some faults are purely for monitoring and logging where as the crucial faults would result in a system shut-off to protect the components and are listed in Table 2.6.5-1.

Table 2.6.5-1. Important Diagnostics and Fault Reactions

Sub-system (s)	Fault type	Reaction	Occurrences
Crucial Sensors	Outside voltage range	Shut off	Brake, temperature, etc
Vehicle, TCM, ECM, Pump, Motor, Brake	CAN message error	Shut off	Message timeout, lose CAN connection, etc
Hydraulics, coolant	Temperature too high	Shut off	Radiator fan not working
Motor, Engine, Pump	Speed too high	Shut off	Motor being torque when transmission is in Neutral
Auxiliary motor	Speed too low	Shut off	Flow valve
Accumulator	Pressure too high/ low	Shut off	Improper control
Accumulator-Hydraulics	Pressure delta (gas-fluid)	Shut off	Valves not open, leaks, etc
Transmission	Inhibits, warnings	Shut off	Over CAN from TCM
Battery	Voltage too high/ low	Shut off	Battery drained, charger issue
ESTOP	ESTOP	Shut off	Emergency shutoff

2.9 UNIVERSITY OF TOLEDO ACTIVITIES

University of Toledo was subcontracted to perform two tasks: 1) develop and optimized shift schedule for the Allison 3700 transmission driven by a hydraulic hybrid motor, and 2) develop speed control algorithms for controlling a hydraulic motor for two different systems, an auxiliary pump system and a pump system to power hydraulic tools, called the HyToPS. The subcontract was an extension of work initiated by the University of Toledo (UT) under contract DAAE07-99-L053 to establish testing capabilities, control methodologies and system evaluation capabilities.

UT was to test hydraulic components supplied by Southwest Research Institute (SwRI) and develop control methodology for select components. Test items may include, but is not limited to pumps, motors, hydraulic conversion devices, manifolds, valves, tools, hoses, connectors and accumulators. For each component supplied, SwRI will specify the test protocol and provide a list of parameters to be measured. Parameters to be measured include, but are not limited to flow, pressure, differential pressure, temperature, rotational speed, leakage rates, vibration, noise, efficiency, electrical power consumption, and torque.

UT was responsible for all test setup and instrumentation and SwRI was responsible for providing the test hardware. SwRI provided specific control goals for the equipment, but would work with the contractor to adjust control goals as overall project goals change.

The UT team developed and simulated a optimized gear shift points to maximize the fuel efficiency and to be able to meet the drive cycles chosen. The final result was in the form of a table that was implemented within SwRI's supervisory controller which was discussed in section 2.6.4(Transmission controller). The final gear selection logic was based on current gear, shift status, motor speed, motor displacement, accelerator pedal rate, and certain parameters of the motor and transmission which can be further tuned to a drive cycle.

The result of these tasks would be implemented into the overall program and if successful, they would have the potential to provide additional fuel economy improvement. SwRI did have alternative controls and hardware plans that would be implemented as a contingency to provide the necessary functionality, but with less potential efficiency.

3.0 DISCUSSION OF RESULTS

The following sections provide a description of what was learned or accomplished during the performance of the project objective. In some cases, the work direction or orginal approach was changed based on data acquired or new information that was not available when the approach was developed.

3.1 VEHICLE PROCUREMENT

Working closely with the TARDEC PM and vehicle manager, a M1078 platform was identified that could be made available for this program. The vehicle had been used for a previous demonstration program, was several vehicle upgrades behind, and was damaged in shipping. The damage consisted of a the cab being crushed.

Under the direction of the TARDEC PM, SwRI contracted BAE Systems to replace the damaged cab and to upgrade the vehicle configuration to the A1R status. The vehicle was repaired and upgraded to an A1R configuration by installing the C7 GEN4 kit and delivered to SwRI for conversion into a HHV.

3.2 BASELINE VEHICLE TESTING

Vehicle fuel economy and performance tests for the conventional, non-hybrid vehicle were conducted both on-road and on a chassis dynamometer. The order of the tests is not important, but due to the local weather conditions, the on-road tests were performed first, when the weather at the SwRI test location was temperate (~ 25 °C). The on-road tests were completed in a few days, and the dynamometer tests – whose duration lasted many weeks – were performed shortly afterwards.

3.2.1 On-Road Testing

Vehicle testing using both on-road and dynamometer drive cycles produced a significant quantity of data. The on-road testing was not part of the original scope, but the characteristic distinctiveness of each vehicle – regarding overall drivetrain losses from engine to wheels – compelled SwRI to perform on-road coast down and acceleration tests. Specifically, the dynamometer setup data was generated using the on-road coastdown test results at two test weights (Figure 3.2.1-1). The dynamometer coefficient data setup was confirmed by attaining the on-road maximum speed using the dynamometer at the two test weights.

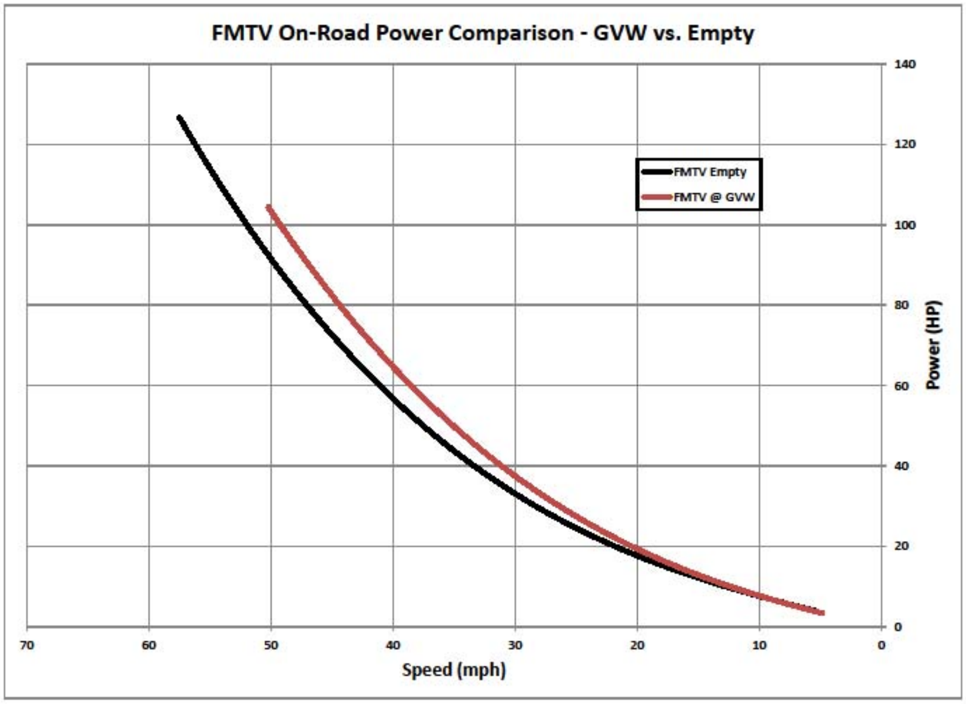

Figure 3.2.1-1. On-Road Coastdown Test Results – Vehicle Empty and at GVW

The two on-road test weights were designated as: empty (@ 18,400 lbs.) and GVW (@ 22,900 lbs.). Testing at two weights produced a better representation of the road load data than at a single weight. As might be expected, Figure 3.2.1-1 shows that the empty vehicle achieved a higher maximum speed. Although not used for establishing accurate dynamometer coefficients, the results from two test weights may be used to separate out the aerodynamic and rolling resistance effects in the coastdown data.

The on-road tests were run over several days on a course with flat, paved (asphalt), and under varying atmospheric conditions. The on-road tests were only run when the weather conditions were acceptable:

- Wind speed less than five miles per hour (mph)
- Zero precipitation and no wet driving conditions
- Moderate temperatures

To mitigate the wind speed and driving direction during the on-road testing, the test runs were made in pairs going in opposing directions. The purpose of two runs in opposing directions was that any small variations in the course altitude when travelling in one direction would be counteracted by the run in the opposite direction. The two opposing runs were executed back-to-back within 10 minutes of each other. The reason for the delay was to allow the testers to collect the data from each run including any notes regarding the test; *e.g.*, when engaged, cooling fan draws a substantial amount of power from the vehicle and this is noticeable during maximum speed testing.

During the on-road testing, data was collected using AutoTap® for Trucks. This is a heavy duty diagnostic scan tool for commercial vehicles that plugs in to the diagnostic CAN port in the cab of the vehicle. The CAN message ID database for the baseline vehicle was not a standard set that would be used for commercial heavy duty vehicles, but the AutoTap scanner was able to collect data for the signals shown in Table 3.2.1-1 at 100 mS intervals:

Table 3.2.1-1. On-Road CAN Message IDs and Units

CAN Message ID	Units
Percent Engine Load	%
Instantaneous Fuel Economy	mpg
Road Speed	mph
Engine Speed	rpm
Percent Accelerator Pedal	%
Instantaneous Fuel Rate	gal/hr
Transmission Output Shaft Speed	rpm
Transmission Range Attained (gear number and clutch state – locked or unlocked)	n/a

Not all of this data was used for the dynamometer setup, but the data has tremendous value applied to performing future modeling and simulation for this family of vehicles. The data provides addition parameters to validate the vehicle models against. In particular, vehicle fuel economy improvement is one of the targeted project metrics along with overall vehicle performance. The collected data allows for the determination on the improvement (or not) of both of these metrics.

3.2.2 Dynamometer Testing

The project test procedure was to run the baseline vehicle first and test the hydraulic hybrid conversion later. The test order for the baseline vehicle was to run the on-road tests first to determine the coastdown coefficient values to be used in the dynamometer testing, and to also measure the baseline vehicle fuel economy and performance. The data acquisition system (DAQ) used while the vehicle was on the dynamometer collected data directly from the vehicles CAN bus the same as the on-road testing. However, where the AutoTap system had a static CAN message ID database, the dynamometer DAQ system was programmable, and could collect additional data. The data set collected during the dynamometer testing is shown in Table 3.2.2-1.

Table 3.2.2-1. Dynamometer CAN Message IDs and Units

CAN Message ID	Units
Front Roll Speed	mph
Front Roll Load	ft-lb
Front Roll Power	hp
Rear Road Speed	mph
Rear Roll Load	ft-lb
Rear Roll Power	hp
Fuel Rate	l/hr
Engine Speed	rpm
Output Shaft Speed	%
Gear Number	n/a
Fuel Economy	l/100 km
Load	%
Pedal Position	%
Dynamometer Cell temperature	°C

For the dynamometer testing, a predetermined set of drive cycles would be run (see table 2.4.1-1) to best represent the vehicle operating in its native situation. However, during the dynamometer testing, the set of drive cycles was changed to include a group of military related drive cycles: Battlefield, Convoy, Hybrid Electric Vehicle Experimentation and Assessment (HEVEA) – Churchville B, HEVEA – Munson, and HEVEA – Harford. The final drive cycle set run on the dynamometer is shown in Table 3.2.2-2.

Table 3.2.2-2. Final Drive Cycle Set

Drive Cycles	Test Weights	Repeats	Grade
Central Business District (CBD)	Empty and GVW	3	No
New York Bus	Empty and GVW	3	No
Highway Dynamometer Driving Schedule (HUDDS)	Empty and GVW	3	No
Heavy Heavy-Duty Diesel Truck (HHDDT) Creep Mode	Empty and GVW	3	No
Heavy Heavy-Duty Diesel Truck (HHDDT) Transient Mode	Empty and GVW	3	No
Orange County Bus	Empty and GVW	3	No
California Air Resources Board (CARB) Low Speed Cruise	Empty and GVW	3	No
Battlefield	Empty and GVW	3	Yes
Convoy	Empty and GVW	3	Yes
HEVEA – Churchville B	Empty and GVW	3	Yes
HEVEA –Munson	Empty and GVW	3	Yes
HEVEA –Harford	Empty and GVW	3	Yes
Full Pedal Performance	Empty and GVW	3	No
5-minute Idle	Empty and GVW	1	No
Custom 1	Empty and GVW	3	No
Custom 2	Empty and GVW	3	No

All five of the added drive cycles have a grade component whereas none of the other drive cycles have any grade changes. Grade changes can be implemented by changing the dynamometer load to account for variations along the z-axis.

The vehicle was mounted in the SwRI Heavy Duty Vehicle Dynamometer Test Cell as shown in Figure 3.2.2-1. The vehicle tires were positioned on the dynamometer rollers, and the vehicle was held stationary by hold down chains fore and aft. The vehicle exhaust was captured and exported out of the test cell, and ambient air was blown over the vehicle – front to rear – to allow for engine cooling during testing. The figure also shows a cooling hose supplying air to the driver.

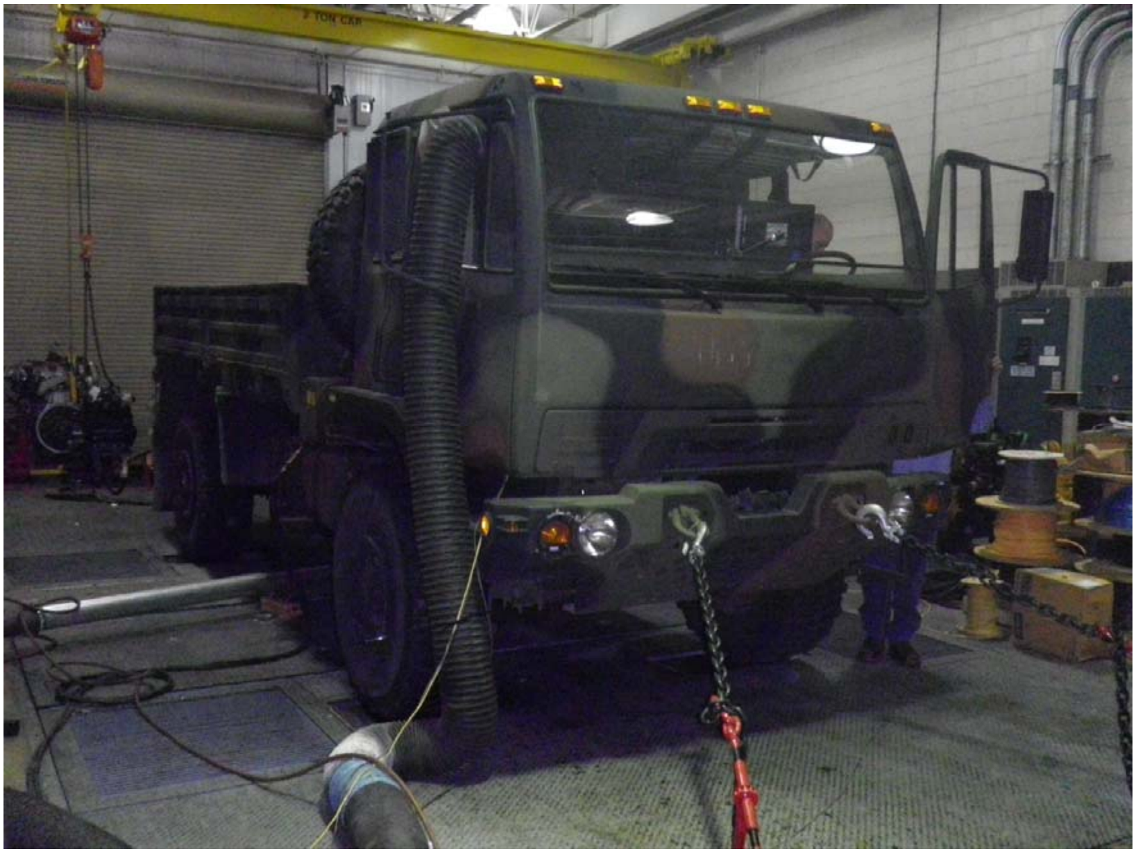

Figure 3.2.2-1. Dynamometer Test Setup

To perform the dynamometer testing, a driver sat in the vehicle and controlled the pedal while following a drive cycle trace on a computer display. Figure 3.2.2-2 shows a Central Business District (CBD) drive cycle trace and the driver following that trace as closely as possible. The only job of the driver is to follow the trace and halt the test in the event of a detected fault; *e.g.*, a malfunction indicator light.

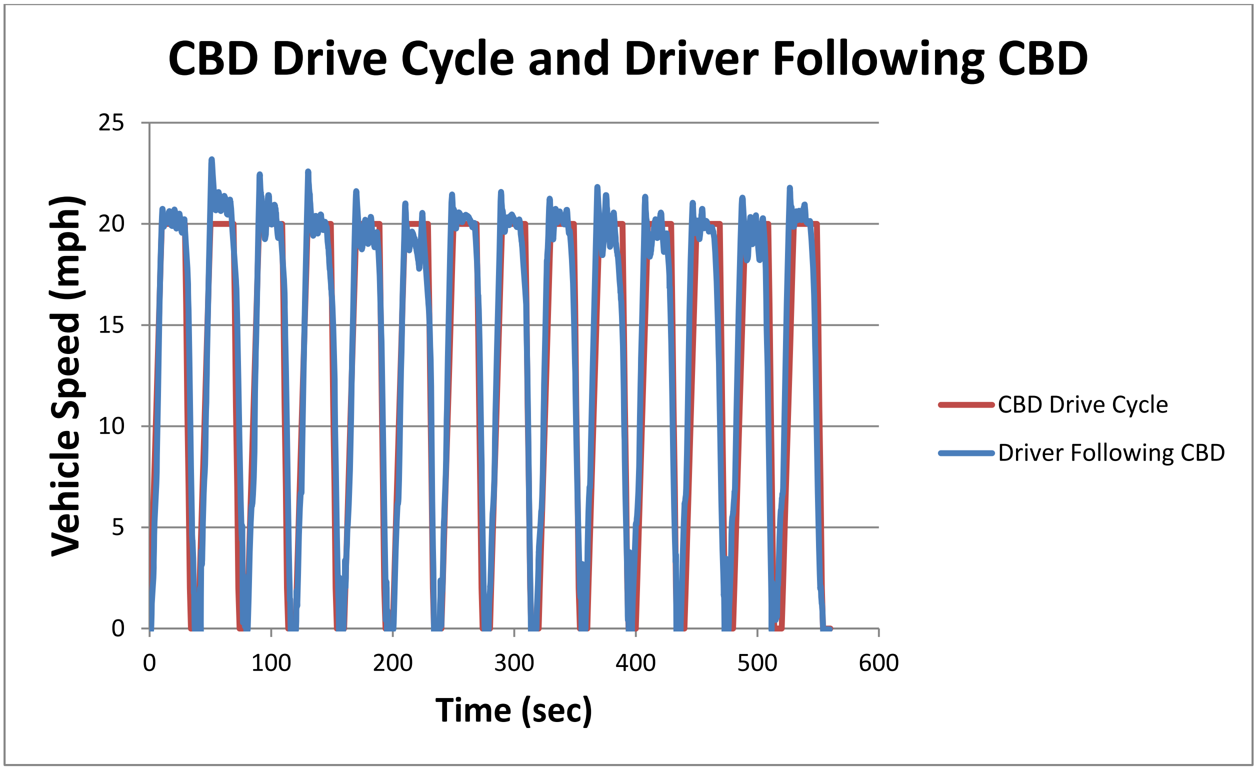

Figure 3.2.2-2. CBD Drive Cycle

All of the drive cycles required the vehicle to be warmed up before testing, and the fuel economy measured over each drive cycle was calculated using at least three drive cycle runs. Several of the military related drive cycles are lengthy (up to ~70 minutes), and this caused the vehicle to spend an extended amount of time on the dynamometer. Similarly, over 2,500 miles were run during the dynamometer testing. Representations of all drive cycles shown in Table 3.2.2-2 are included in Appendix A.

3.3 MAJOR COMPONENT SPECIFICATION AND PROCUREMENT

The results of the major component procurement are described below:

3.3.1 Use Hydraulic Pump, Motor, and Megablock

The hydraulic pump, motor, megablock, and accumulator control valve were procured under a subcontract to Eaton Corporation. Both the hydraulic pump and motor had their own integrated switching block attached to them to control the mode between neutral or pumping and motoring.

Two sets of pumps and motors were supplied. One set was a standard efficiency using a standard efficiency assembly. The second set of pump and motor were very a high efficiency version of the standard efficiency. The primary difference was that improved bearings were used to reduce friction losses. In addition, a displacement sensor was incorporated to feed back actual displacement for more accurate control. Each pump and motor also had a dedicated Electronic Control Unit (ECU) to control the mode selection and displacement control. The lead time to procure the high displacement pumps and motors was expected to extend beyond the schedule for vehicle integration, therefore, the standard efficiency pump and motor were procured much earlier. Figures 3.3.1-1, 3.3.1-2, 3.3.1-3 and 3.3.1-4 show the motor, pump, megablock, and accumulator control valve, respectively.

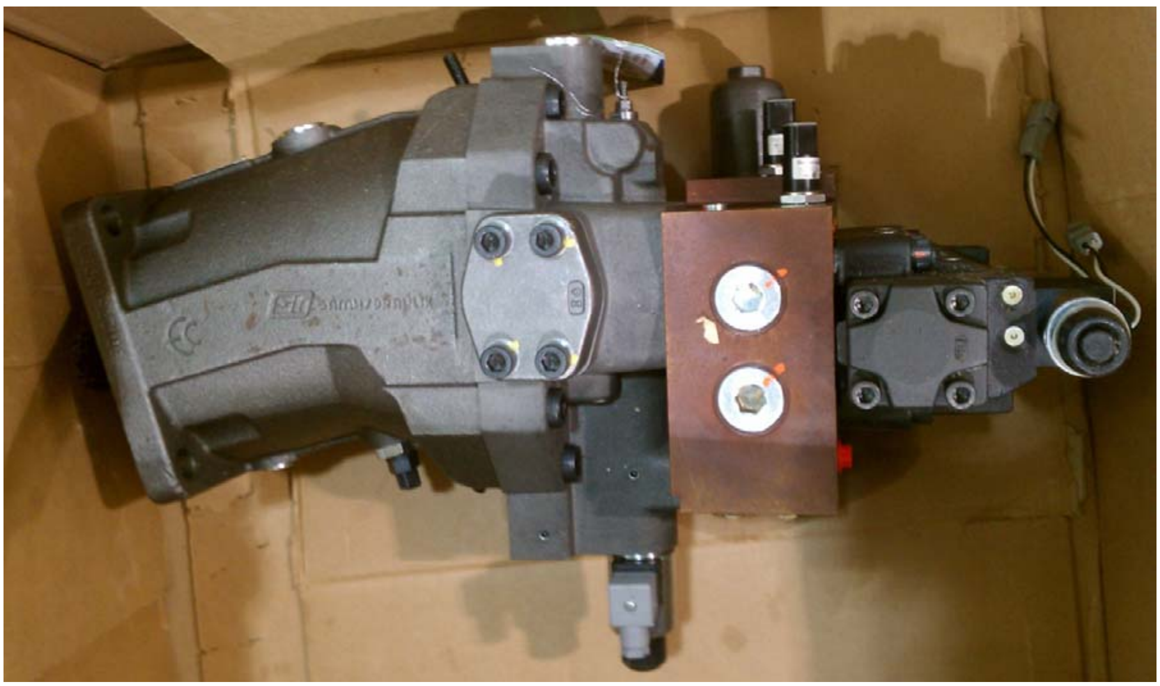

Figure 3.3.1-1. Eaton Motor with Mode Switching Valve

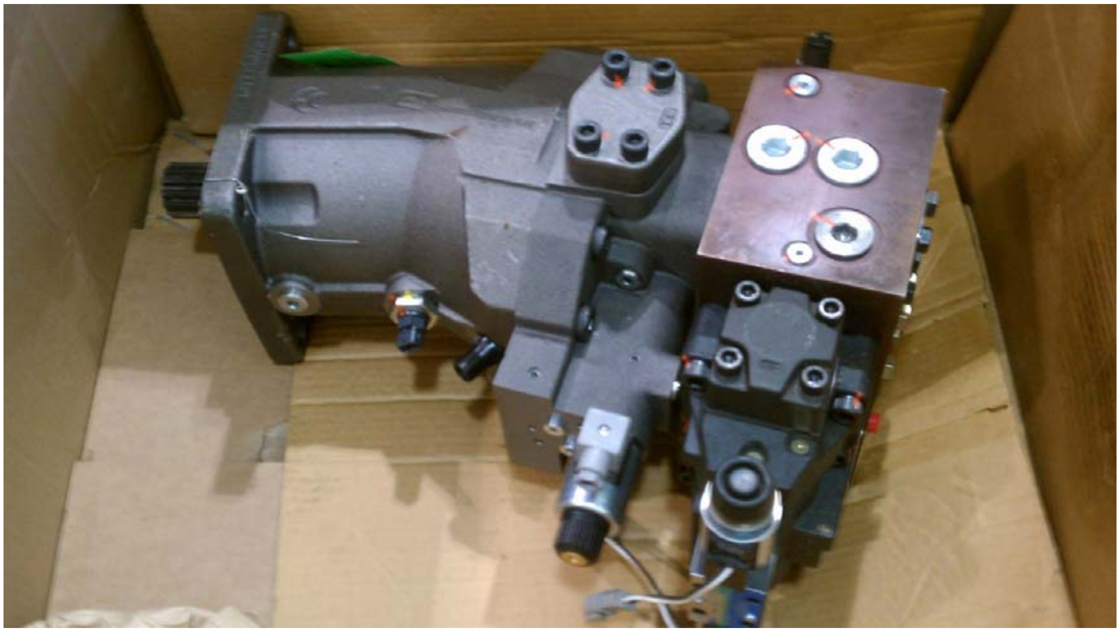

Figure 3.3.1-2. Eaton Pump with Mode Switching Valve

Figure 3.3.1-3. Eaton Megablock

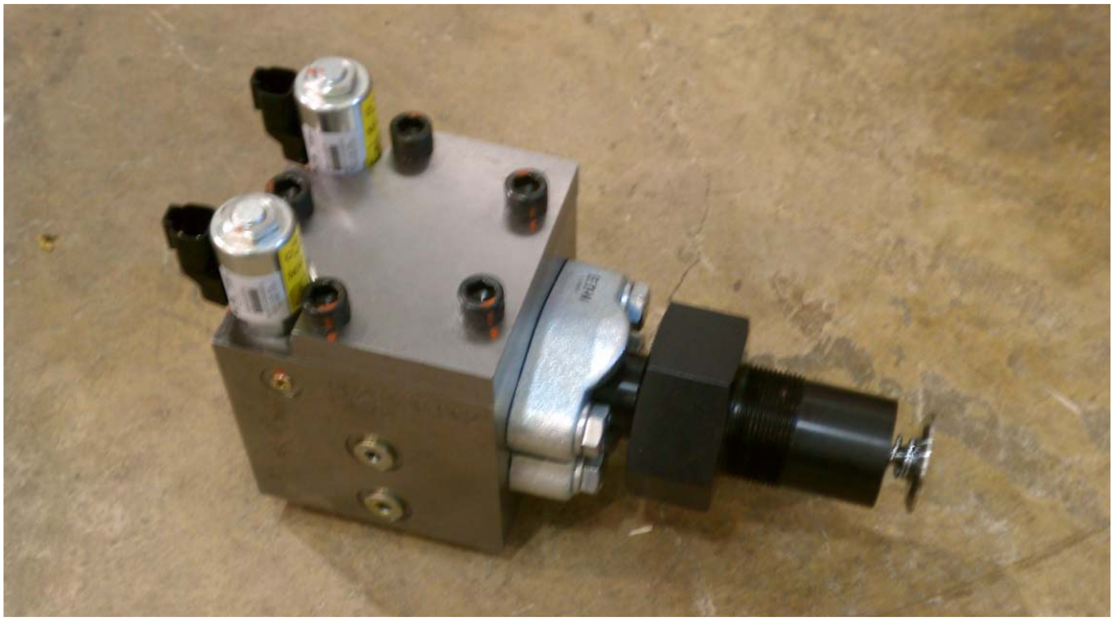

Figure 3.3.1-4. Eaton Accumulator Control Valve

Before delivery to SwRI Eaton conducted performance tests on each of these pump and motor units. The test results for the high efficiency pump and motor are included in the appendix. Eaton also tested the functionality of the Megablock and preset the pressure reducing valves and relief valves.

3.3.2 Step-up Gearbox

To match the engine power output curves with the hydraulic pump power curves a step-up gear box was purchased. A Durst build to order gearbox purchased. Specific specifications are listed below:

- 0.8:1 – speed increaser gear ratio
- 9 O'clock pump mounting pad (relative to the center line) in rear view For reference Durst drawing 1PDD06D shows a 6 O'clock mounting.
- SAE No 2 Engine adapter
- SAE 11.5 Drive plate bolt pattern.
- 29T- 12/24 30˚ Pressure angle input shaft
- SAE "D" Pump Adapter
- SAE "D" 13T-8/16 spline output shaft

As part of the purchase a Torsional Vibration Analysis (TVA) was performed to insure the proper coupling was selected to prevent the diesel engine torsinals from damaging the gearbox.

3.3.3 Hydraulic Accumulators

Two high pressure and two low pressure accumulators were procured assembled and tested at SwRI. The vessels were procured from Lincoln Composites, the rubber bladders were procured from Martin Rubber, and SwRI fabricated other required hardware. The accumulators are shown installed in the vehicle in Figure 3.3.3-1. Over pressure burst discs, supplied by LaMOT, were installed on the accumulator gas manifold, with 10,000 psi burst discs on the high pressure accumulators and 360 psi burst discs on the low pressure accumulators.

Figure 3.3.3-1. Composite Hydraulic Accumulators

3.3.4 Vehicle Control System

The supervisory controller was a dSPACE microAutobox with IO conditioning boards and solid state relays to control the solenoids. The figure 3.3.4-1 (below) shows the various components that comprised the supervisory controller hardware. The dSPACE microAutobox was a 1401/1505/1507 multi-board controller with sufficient I/O and processing capability needed for this application. Custom I/O conditioning units were used to provide voltage shifting, additional current source capability, temperature and pressure normalization, and driving solenoids using relays. It had three CAN busses over which it communicated with the peripheral ECUs of the HHV-FMTV(e.g. engine, transmission, pump, motor, etc).

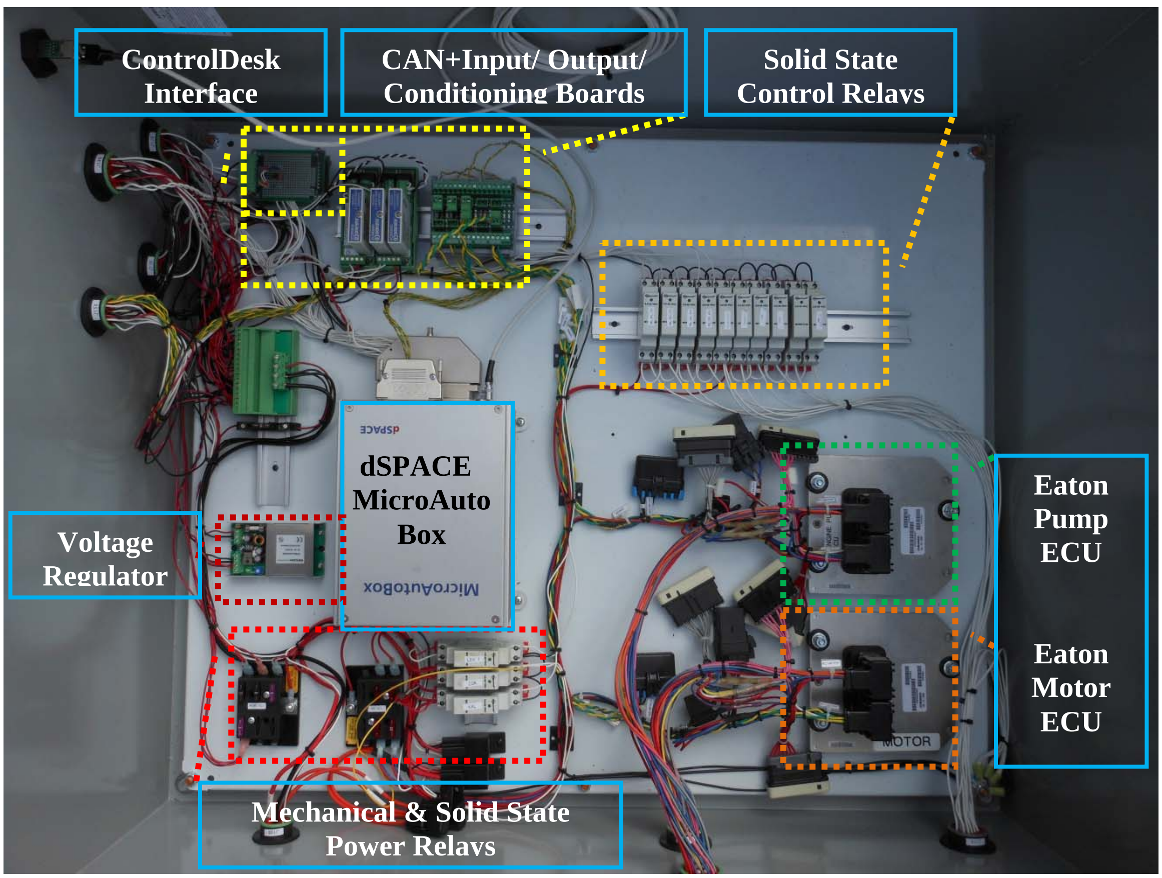

Figure 3.3.4-1. dSPACE microAutobox Supervisory controller

3.3.5 Closed Center Steering Gear

A closed center steering gear was purchased from R.H Shephard. The component was custom built to replace the stock steering gear. The closed center steering gear used high pressure fluid from the hydraulic hybrid system. The high pressure supply passed through a pressure reducing valve, located in the Megablock, to limit the maximum pressure applied to the steering gear to its rated pressure of 2600 psi. It replaced the stock open center steering gear and engine driven fixed displacement pump. This new steering gear was necessary to enable engine off operation and maintain power steering. The closed center steering gear only requires high pressure flow when steering actuation occurs. When there is no steering actuation, the flow is limited to an insignificant leakage rate.

3.4 POWERTRAIN VEHICLE INTEGRATION

3.4.1 Drivetrain Integration

The integration of the hydraulic hybrid drivetrain began with the removal of the bed and the removal of the Allison transmission. The Durst gearbox was then installed on the CAT C7 engine. This required the replacement of the engine flywheel and the engine bell housing. Figure 3.4.1-1 shows the new flywheel and Figure 3.4.1-2 shows the Durst gearbox mounted on the engine.

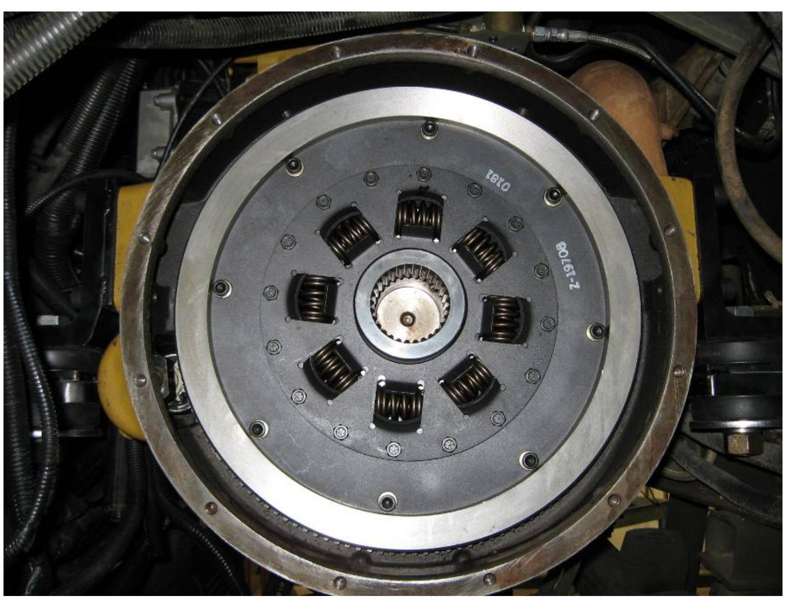

Figure 3.4.1-1. New Engine Flywheel

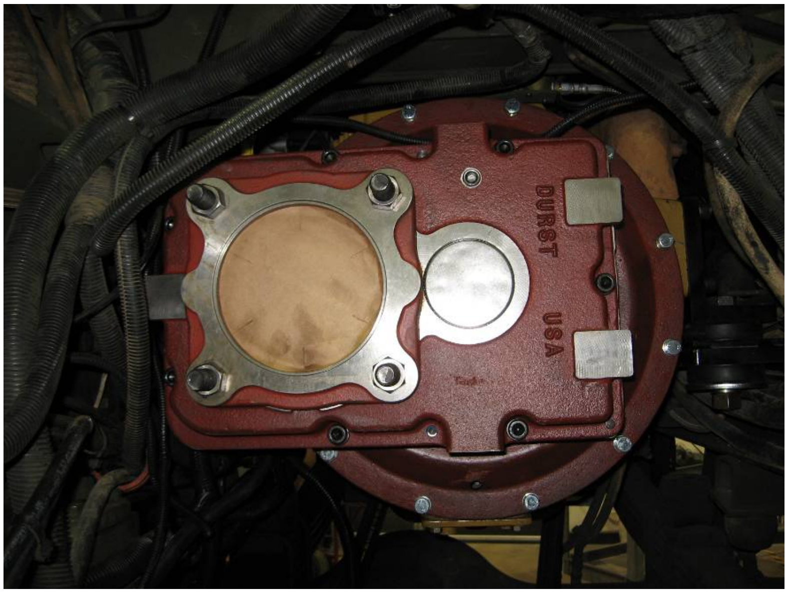

Figure 3.4.1-2. Durst Gearbox Mounted on Engine

The engine driven pump was then mounted, which required removal of the standard cross-member, which would have had interference, and replacing it with a new curved cross-member as shown in Figure 3.4.1-3

Figure 3.4.1-3. Engine Driven Pump and New Cross-member

The transmission assembly with the drive pump motor were then mounted. Additional modifications to the originally designed transmission support frame were required as well as to the cross-member located at the rear of the transmission. Inconsistencies between the supplied CAD models of the vehicle and transmission to the actual vehicle and transmission required that modifications be made to the mounting brackets for the transmission support frame. Figure 3.4.1-4 shows the transmission and drive motor assembly mounted in the vehicle chassis.

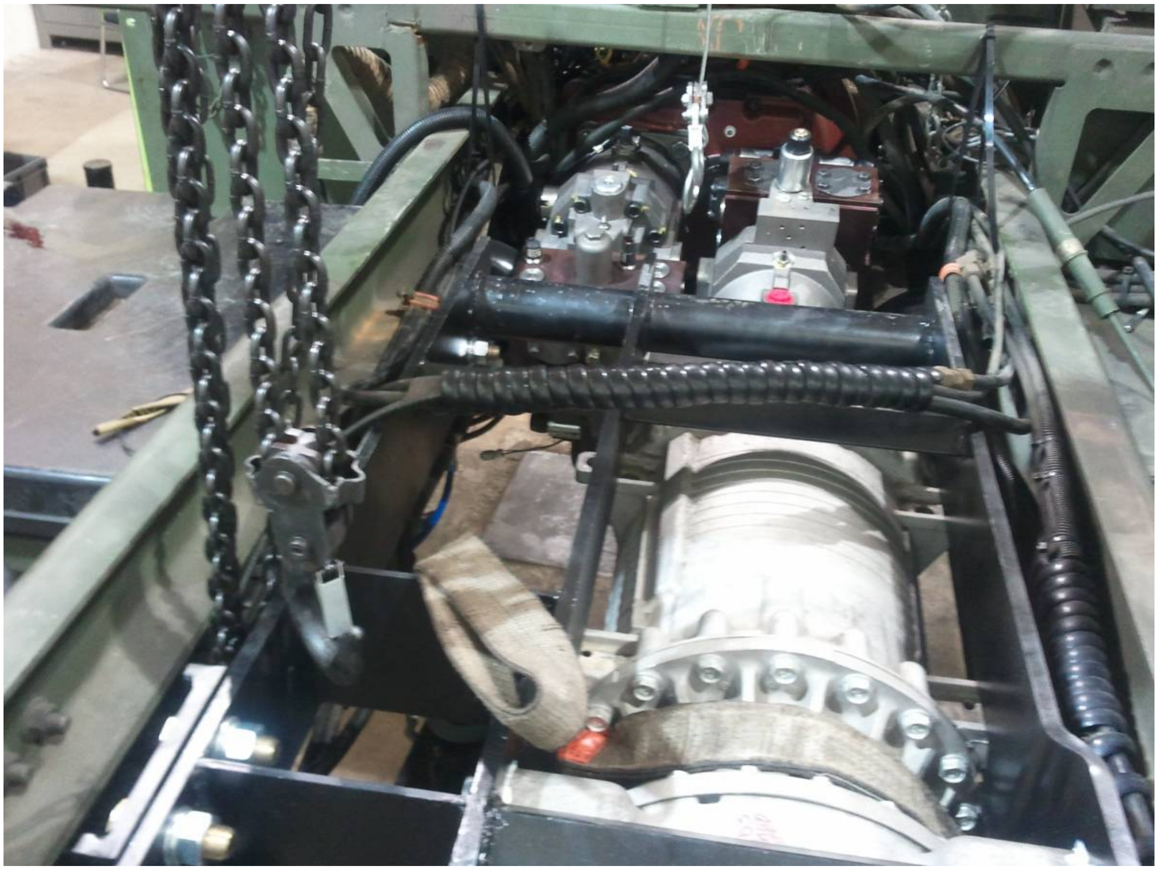

Figure 3.4.1-4. Transmission and Motor Assembly Installation

Figures 3.4.1-5 and 3.4.1-6 show the installation of the transmission and motor assembly along with the megablock located to the rear of the transmission. The main hoses between the megablock and the engine driven pump and the drive pump motor are also shown.

Figure 3.4.1-5. Installation of the Transmission and Drive Motor Assembly and the Megablock

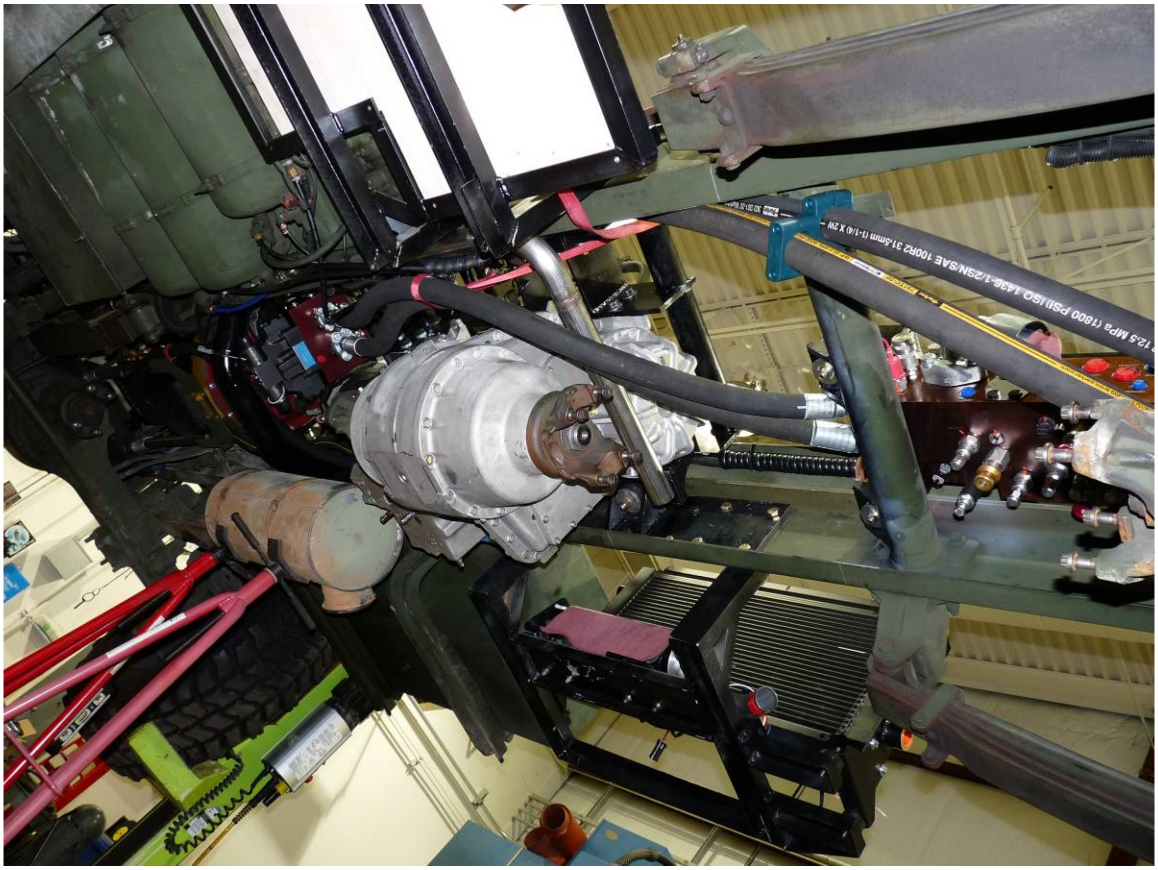

Figure 3.4.1-6. Installation of the Hybrid Driveline with the Megablock along with Main Hoses Viewed from the Bottom

3.4.2 Hydraulics Integration

The hydraulic system was assembled and integrated according to the schematic shown in Figure 2.1.2-1 In addition, an auxiliary circuit providing the transmission lubrication functions and acting as the hydraulic hybrid system charge pump as well as filtration and cooling. The final assembly of the auxiliary motor and pumps included the following displacements of Parker gear pumps and motor.

Table 3.4.2-1. Auxiliary Motor Pump Stack

	Pressure, psi	Selected Displacement, cc/rev	Speed, rpm	Flow Rate, gpm
Motor	2000	8	1400	3.0
Charge Pump	250	19	1400	7.0
Transmission Control Pump	160	14	1400	5.2
Transmission Lubrication Pump	30	23	1400	8.5
Transfer Case Scavenge Pump	1	19	1400	7.0

A photograph of the assembled pumps and motor is shown in Figure 3.4.2-1 and a hydraulic schematic of the auxiliary pump and the rest of the auxiliary system is shown in Figure 3.4.2-2.

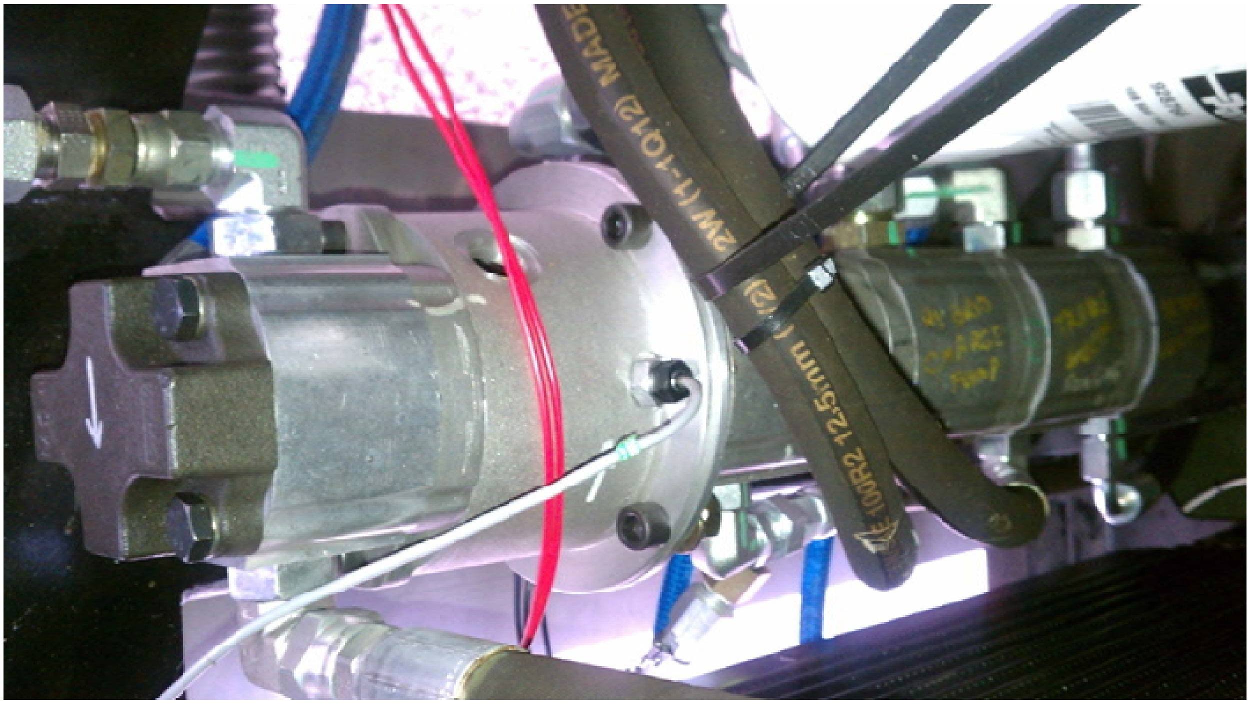

Figure 3.4.2-1. Transmission Lubrication Pumps and Charge Pump Assembly

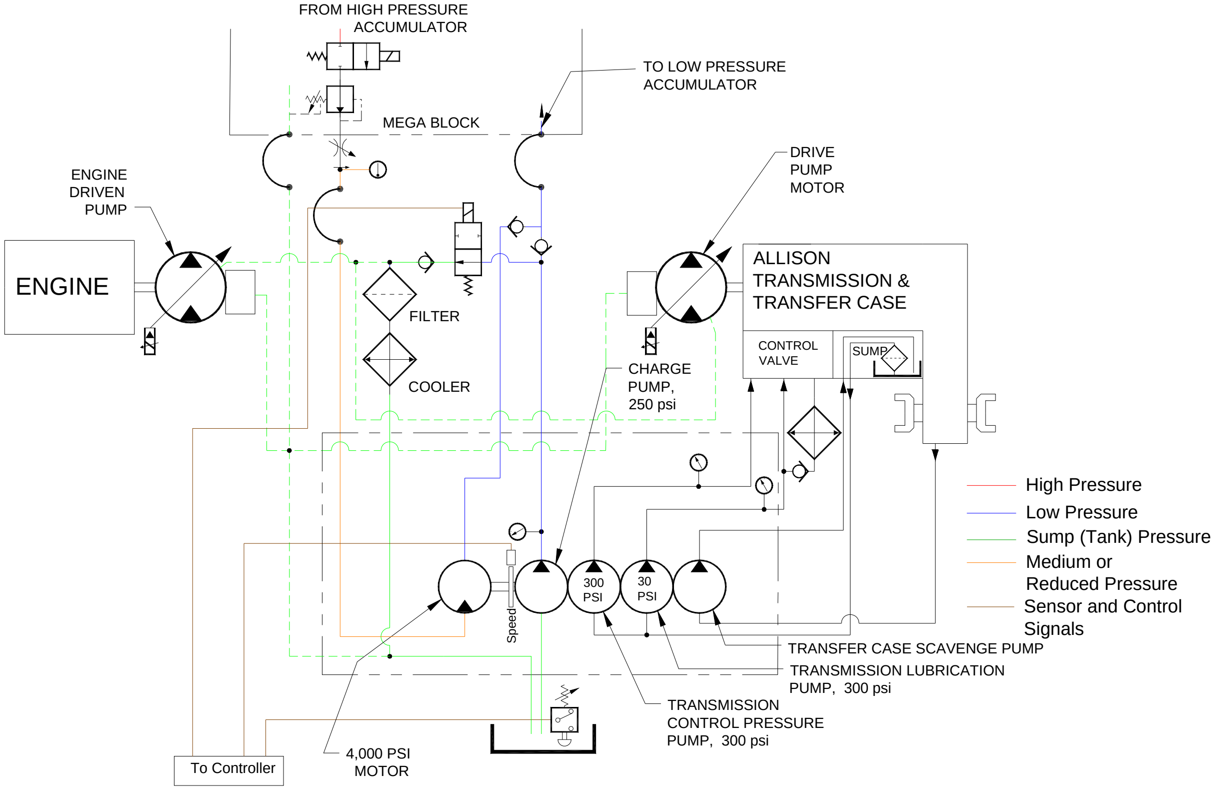

Figure 3.4.2-1. Schematic of Transmission Lubrication Pumps, Charge Pump Assembly, and Fluid Conditioning Circuit

Figure 3.4.2-3 Is a photograph of the reservoirs, located on the left hand side of the vehicle. The large reservoir is for the hydraulic hybrid system and the smaller reservoir is for the HyToPS system. Figure 3.4.2-4 is a photograph of the hydraulic hybrid system oil cooler, located on the right hand side of the vehicle. This cooler is an AKG D70 air to oil cooler rated for nominal heat rejection of 70 horsepower. It has two 24 VDC fans that are thermostatically controlled to turn on and off.

Figure 3.4.2-2. Hydraulic Reservoirs

Figure 3.4.2-3. Hydraulic Oil Cooler

3.5 HYDRALIC TOOL POWER SYSTEM

The results of the controls development by the University of Toledo, to be described in a later section, revealed that the actuator hardware selected for the motor displacement control was undersized and not capable of operating over the full range of pressures. It was therefore decided to abandon the implementation of the HyToPS system on the vehicle.

The objective of implementing the Hydraulic Tool Power system was to demonstrate the functionality of the system and the ability to tap into the hydraulic hybrid system and efficiently export power to hydraulic tools. The hardware, as previously described, was procured, assembled and made available by University of Toledo for developing the control algorithm. University of Toledo was able to demonstrate controllability for part of the operating range, but they were unsuccessful in developing adequate control for the complete operating range. The HyToPS system was therefore not integrated into the HHV FMTV. More details of the University of Toledo development on the HyToPS controls is included in a later section of this report.

3.6 DRIVE CYCLE TESTING

The program approach for developing the fuel economy comparisons between the baseline FMTV and the modified HHV version of the FMTS has been to test the HHV on the SwRI heavy duty dynamometer, running the same drive cycles as were run on the baseline vehicle. Prior to running the final fuel economy tests with the drive cycles a developmental effort was undertaken to tune and calibrate the HHV control system. The planned developmental effort consisted of the following basic tasks or milestones.

- Validate transmission shift control
- Validate hydraulic accessory system supply
- Calibrate control system (both drive train and accessory systems)
- Validate/calibrate pressure control strategies
- Validate that the HHV can successfully run individual drive cycles
- Optimize control for maximizing fuel economy

During activities associated with validating the pressure control strategies, a hardware failure occurred internal to the hydraulic motor driving the Allison Transmission. Although the hydraulic motor would still function as a motor, the control of the motor displacement was lost.

Eaton hydraulic design engineers were contacted, and system operational data was acquired and sent to Eaton for analysis. After several telecon between the Eaton and SwRI design teams, Eaton confirmed that the failure could not be repaired while the motor was in vehicle. What would be required is to remove the hydraulic motor from the vehicle and send it back to Eaton for repair or replacement. Both the motor options of either repair or replacement would require a lead time longer than the program remaining period of performance.

Based on the motor failure and the inability to continue developmental testing, the HD dynamometer development was completed and fuel economy tests were not performed. The following sections report on the data acquired and associated analysis performed on two drive cycles that were being used for initial system validation. The results presented do not represent a valid comparison between a baseline FMTV and the potential of a HHV FMTV. They are included in this report in support of any future program activity and associated analysis..

3.6.1 New York Buss Cycle

The NY bus cycle is particularly suited to a series hybrid application due to its lower speeds and frequent stop-starts. It has 385 seconds of stop time out of 600 seconds of total test time which could be used to improve the fuel efficiency over the entire cycle. It is also a representative test for vehicles like busses and short haul trucks.

FMTV Result

The SHHV FMTV was able to comfortably go through the cycle. The only available data for this cycle is a baseline test with the standby shut-off modes disabled and is shown in Table 3.6.1-1. The dyno coast downs are based on the same unloaded weight of the stock empty FMTV of 18,400 lbs. The weight of the FMTV after the SHHV installation is 19,710 lbs.

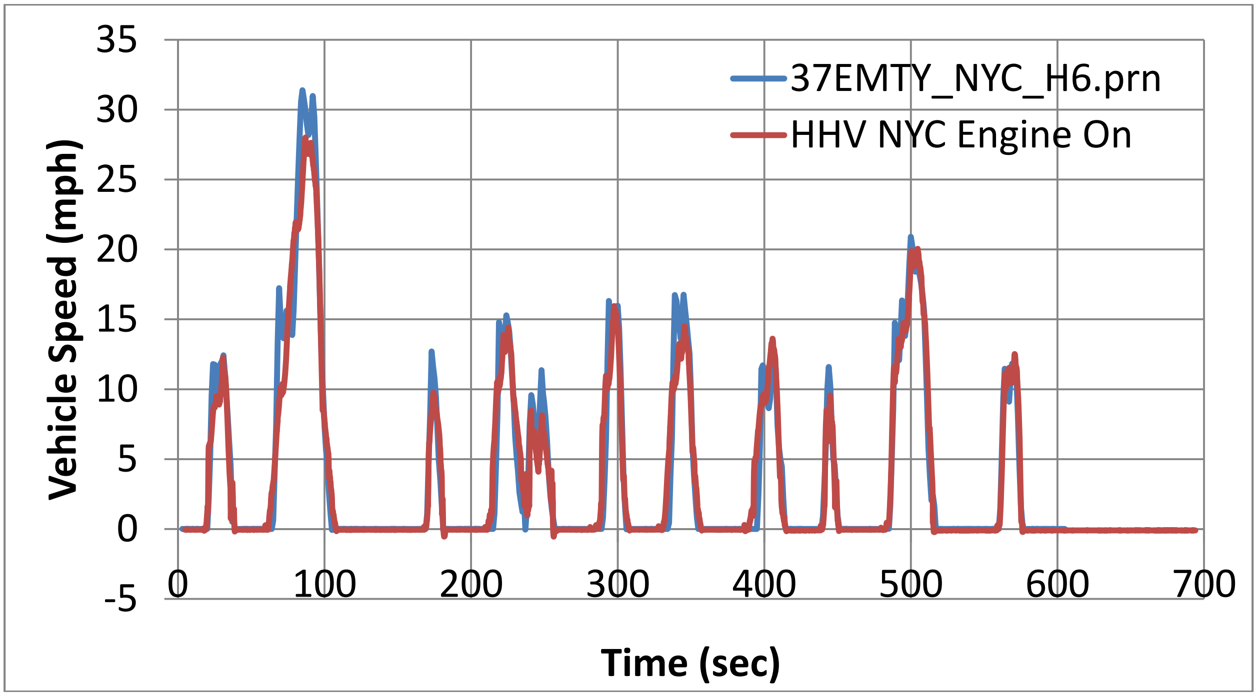

Figure 3.6.1-1. New York Bus Cycle Results

Table 3.6.1-1. New York Bus Cycle Results

Test Name	MPG	Notes
Empty Stock FMTV		
32EMT_NYC_H2.prn	2.09	
33EMT_NYC_H3.prn	2.13	
36EMTY_NYC_H5.prn	2.10	
Average	**2.11**	**Average MPG of stock FMTV Over New York bus cycle**
Empty SHHV-FMTV		
NAC_20120907_Nybus_engON.xlsx hd01a_nybus_baseline_engon.xls	**1.09**	Engine ON baseline with Hybrid-Hydraulic configuration
Activating Standby Stop-Start	1.2483 to **1.6110**	Based on combined Engine-on cycle data with fuel savings during a standby-off. Note: An actual test was not performed

Analysis

In addition to the baseline We were able to collect fuel saving data due to standby-shut off modes implemented and discussed in section 2.6.3 (6e-off and 6h-off). With this we were able to extend the available data for the NY bus cycle to include the standby shut-off modes. From data, we have the fuel savings and overhead of a hydraulic start which is used to calculate fuel savings due to the same over the NY bus cycle tabulated below.

Consumption/ Savings	Hydraulic Start	10 sec Standby-off
Fuel (gals)	0.001138	0.0056

Table 3.6.1-2. Fuel savings over New York bus cycle using Standby Stop-Start feature

Vehicle Stops	Cumulative Time Stopped	Individual Stop Times	Available Stop time	hyd SS viable	Total Standby fuel saved	Fuel used for Hydraulic Start	Net fuel saved due to standby-shutoff
#	sec	sec	sec		gals	gals	gals
0	26.8	26.8	21.8	1	0.01268	0.001138	0.011542
1	55.2	28.4	23.4	1	0.01364	0.001138	0.012502
2	121.2	66	61	1	0.0362	0.001138	0.035062
3	152	30.8	25.8	1	0.01508	0.001138	0.013942
4	184.8	32.8	27.8	1	0.01628	0.001138	0.015142
5	213.2	28.4	23.4	1	0.01364	0.001138	0.012502
6	252	38.8	33.8	1	0.01988	0.001138	0.018742
7	280	28	23	1	0.0134	0.001138	0.012262
8	312	32	27	1	0.0158	0.001138	0.014662
9	355	43	38	1	0.0224	0.001138	0.021262
10	386	31	26	1	0.0152	0.001138	0.014062
Total Fuel saved from Standby Stop-Start over NY bus cycle							**0.181682**

Total miles covered during the cycle is 0.642 miles and total fuel consumed is 0.58 resulting in a baseline MPG of 1.106. With an estimated fuel savings of 0.1817 gallons over the cycle, the corrected MPG comes out to 1.611 improving the baseline SHHV MPG by 45.5%. The final MPG results for the SHHV-FMTV still falls short of the stock by 23%. These MPG results are purely indicative of the ability of the system from existing data. Further optimization will have included engine operation optimization and target pressure optimization to use the engine and pump more efficiently.

Energy Loss within the system

Table 3.6.1-3 shows a breakdown of the energy losses within the SHHV-FMTV as a lumped estimate measured by the fuel consumed. These numbers are based on a some tests performed to measure the fuel consumed to hold a steady HiP accumulator pressure against the intrinsic operational losses of the system with the vehicle being stationary.

Table 3.6.1-3. Lumped energy loss within the SHHV-FMTV

Component	Potential Improvements	Engine On NY bus cycle Fuel consumed (gal)	Lumped Loss with Standby Shut-off (gal)
Fuel		600s x 0.00056gal/s = **0.336**	0.336-0.1817 = **0.1543**
Engine (hydraulic losses)	Optimized engine-pump operating points		
Pump (hydraulic losses)			
Accumulator	Reduce bladder-to-fluid gas osmosis		
Auxiliary Motor	Isolate charge pump from transmission		
Engine (motive)	Optimized engine-pump operating points	**0.244**	**0.244**
Pump (motive)			
Traction Motor	Optimized shift points		
Transmission			
Vehicle	Reduce overall weight		
Total		**0.580**	**0.3983**

To understand the energy losses within the system, a detailed energy walkthrough of the system should be performed which will need additional sensors and ability to observe and optimize the losses in real-time over the cycle.

3.6.1 Heavy Duty Urban Dynamometer Drive Schedule (HD UDDS) Cycle

The HD-UDDS cycle is a high speed test with maximum speed crossing 55 mph. We were not able to complete a successful run of HD-UDDS before the motor failed.

Results

Figure 3.6.2-1 (HD-UDDS cycle) shows the level of achievement for the cycle. The high speed operation was intentionally avoided at that point as it required some target pressure tuning and additional logic to make sure the accumulator wasn't drained during constant high power as demanded by cycle.

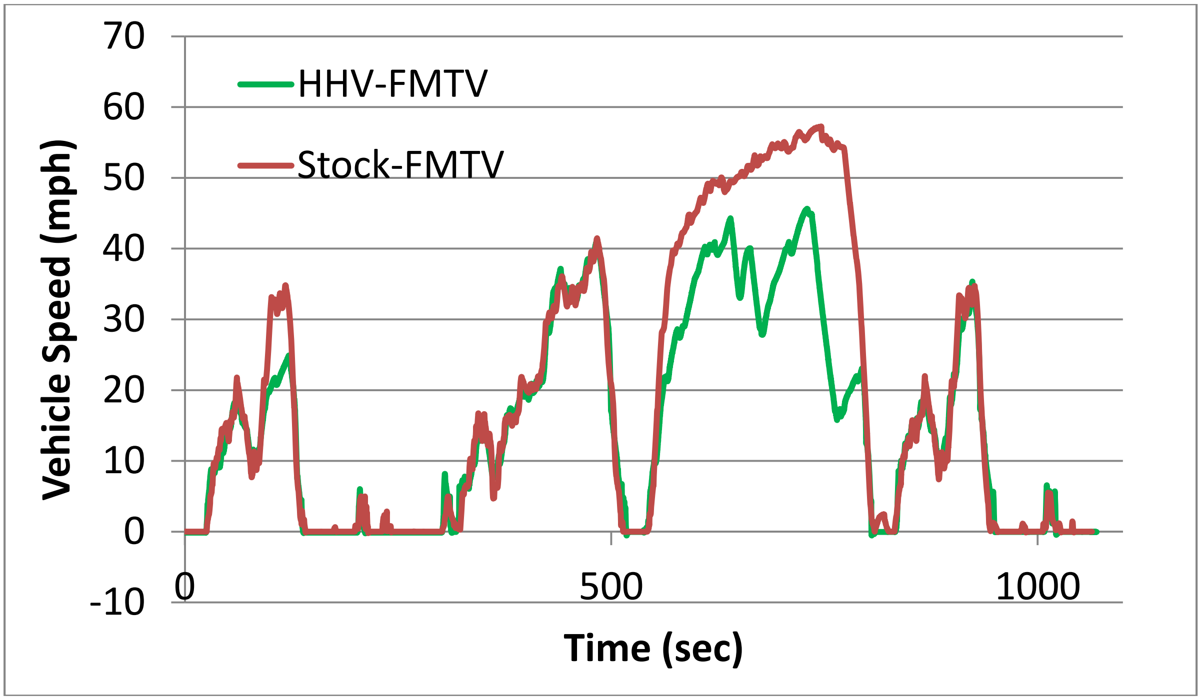

Figure 3.6.2-1. HD-UDDS Cycle Results

The results of the HD-UDDS helped calibrate the system to achieve the higher speeds.

Table 3.6.2-1. HD-UDDS Cycle Results

Test Name	MPG	Notes
Empty Stock FMTV		
44army_udds_H1.prn	5.48	
45army_udds_H2.prn	5.50	
46army_udds_H3.prn	5.55	
47army_udds_H4.prn	5.46	
Average	**5.50**	**Average MPG of stock FMTV Over HD-UDDS cycle**
Empty SHHV-FMTV		
NAC_20120907_HDUDDS_h1h2h3partial_engON.xlsx	**4.98**	Engine ON baseline with Hybrid-Hydraulic configuration. Unable to follow cycle completely

Analysis

Based on the results, we observed that the road demand for the accelerations in the 3rd hill of the cycle required us to increase the gains on the pressure controller to keep up with the demand. Logic was added to make sure that the pressure controller pumping into the accumulator exceeded the losses and the drain caused by the motor driving the wheels. We were not able to complete a successful run of HD-UDDS before the motor failed.

3.7 UNIVERSITY OF TOLEDO ACTIVITIES

University of Toledo (UT) was assigned two tasks in support of this project:

1. Develop an optimal shift strategy for an Allison 3700 transmission driven by an Eaton Hydraulic motor.
2. Develop a speed control system to power to auxiliary pumps including a charge pump and lubrication and cooling for an Allison 3700 seven speed transmission using hydraulic stored energy from a hydraulic hybrid vehicle and
3. Develop a speed control system for a hydraulic power takeoff for powering hydraulic tool from stored energy from a hydraulic hybrid vehicle

3.7.1 Development of an Optimal Shift Strategy

The shift strategy programmed in the Allison 3700 Transmission Control Module was designed to optimize efficiency and performance while being driven by a CAT C7 diesel engine. A new shift strategy to optimize performance and efficiency with the Eaton hydraulic motor is manufactured by Brevini with a 160 cc/rev maximum displacement. The hydraulic motor is a variable displacement bent-axis type motor. Its efficiency is a function of pressure, displacement, and speed. UT was tasked to come up with a shift schedule to maximize the efficiency of the motor while meeting the driver torque commands. Some of the constraints placed on the control were as follows:

- Drive motor can be operated up to 3000 rpm at full displacement.
- Supply pressure range varies between 2400 and 6500 psi

UT developed a shift schedule to maximize motor operational efficiency within these constraints. Efficiency maps of the pump/motor, the HUDDS drive cycle, and a vehicle model were utilized as Matlab S-function models to the vehicle simulation along with optimization algorithms in m-file format created by UT. They generated a spreadsheet with a recommended shift strategy that was implemented on the actual vehicle.

3.7.2 Develop a speed control system to power to auxiliary pumps and develop a speed control system for a hydraulic power takeoff for powering hydraulic tools (HyToPS)

Both speed control systems used similar components although their operating conditions were quite different. The basic concept of the designs was that a Rotary Power A70 Variable Displacement Hydraulic Pump/Motor (A70) would be used to drive pumps that would provide the functions needed. For the hydraulic power takeoff, the A70 in motor mode would drive a gear pumps on a separate hydraulic system to provide hydraulic power takeoff for open-center hydraulic tools. The hydraulic fluids were separated to prevent contamination of the vehicles driving system. The operating conditions for this system required 2000 psi service at 8 gallons per minute and operated intermittently as needed by the tool used externally by an operator.

Although the flow would be constant, the pressure could vary, thus the load on the A70 motor would have considerable variation.

The second function required an A70 (separate and distinct from above) to drive ganged pumps with a total service load of 250 psi at 8 gallons per minute. The loading conditions for this system required less variation although a pump for removing reservoir oil would be cycle on and off to a bypass mode during normal operation. However, pumps for lubrication and cooling to the transmission would operate continuously.

While manufacturer literature for the A70 provided operating characteristics, more information was needed to design a suitable control system for the applications. Therefore, characterization testing of the A70 was performed to both validate the manufacturer's data as well as to provide data for design of the control systems.

In further testing, control systems were developed and tested based on pressure modulation using the A70 in a "Hardware in the Loop" simulation with LabView and Simulink. Load was provided by a dynamometer. When the pump and actuator components became available, the A70 with these components were then retested. The control systems were modified for the performance of the components.

The significant findings are:

1. The Rotary Power A70 variable displacement pump/motor as tested performed according to manufacturer's provided data with minor exceptions.

2. The control of the A70 displacement control servo-actuator requires more force that was originally anticipated. Initial destroking from full displacement required a force that varied from approximately 1 kilogram of force to 8 kilograms of force. It appeared that this was a function of both inlet pressure and the length of time that the A70 was in a full displacement mode.

3. The Copely electromechanical linear actuator with 1:1 ratio lever arm was insufficient to provide the force needed to destroke the A70. When the arm was increased to 2:1, the actuator functioned successfully. Testing with the actuator arm increased to 3.2:1 resulted in less lag time.

4. The linear actuator operational conditions were dependent on the supply current, the supply voltage and control voltage provided. Only the supply and the control voltage could be measured. The supply voltage was 29 volts and the supply current was 6 amps.

5. Actuator performance was highly variable, due to the variability of forces needed for stroking and destroking of the A70 and resulted in a wide spread of performance times.

6. Actuator controller tuning using the Copely Controls software package CME2 is critical for best performance.

7. The actuator heats at a rate of 1.4^{o}C for every minute of use while loaded and would overheat is used continuously.

8. The mean system lag is approximately 377 milliseconds with a standard deviation of 0.286 seconds. In some circumstances, particularly on the first destroking of the A70, the lag time was greater than two seconds. This is attributed to the variability of the force required and showed significant improvement with the increase of the actuator lever arm ratio.

9. The rise time from 10% to 90% of the command speed had a mean of 1.864 seconds The majority of rise times were less than 1 second.

10. There was an average of 37% speed overshoot (undershoot). The overshoot was greater, particularly on destroking, sometimes as high as 90%, which often resulted in shutting down the motor temporarily. This was improved with increasing the actuator lever arm length.

11. The mean settling time for the system (where the speed achieves and holds a speed within 5% of the command) was 5.886.

12. The control response was determined to be unsatisfactory for implementation on the vehicle because it is critical that the main hydraulic pump to the transmission maintain sufficient flow to supply pressure for clutch actuation. The inability to maintain transmission control pressures could result in damage to the transmission or the hydraulic system.

13. The testing and development schedule was modified because of difficulty in achieving satisfactory control for the auxiliary pump system and the risk that the electromechanical servo-actuator would overheat. The overall project was reoriented which removed the focus from the HyToPS system. As a result the HyToPS configuration was mainly used to set up the testing environment but was not itself fully tested under the design conditions.

Figure 3.7.1 shows a sample of the speed control response to changing speed commands for a complete test cycle and Figure 3.7.1 shows the details of the speed control response to a single speed step. Notice the large amount of overshoot and undershoot as well as the relatively larger time delay.

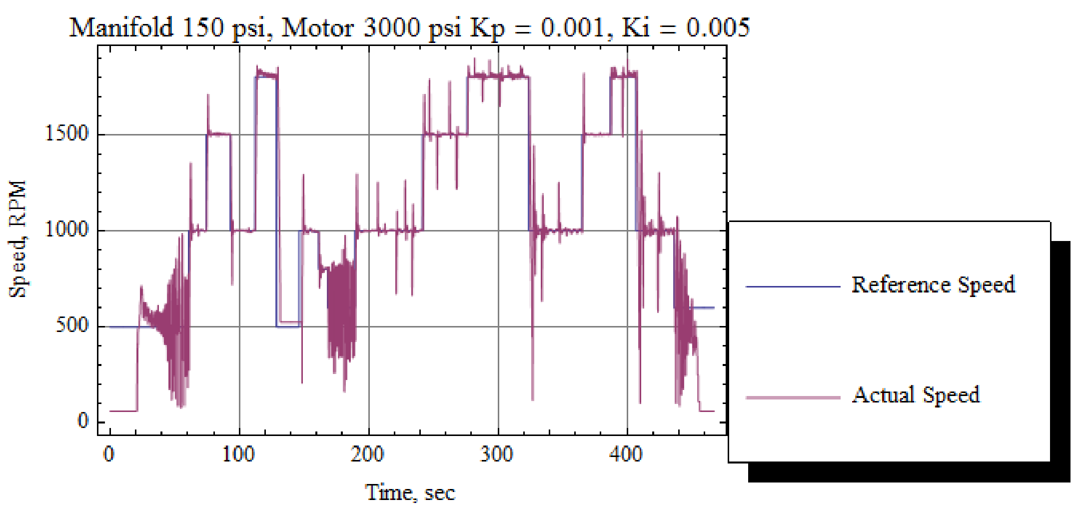

Figure 3.7.2-1 Speed Control Response of Auxiliary Pump System for a Complete Test Cycle

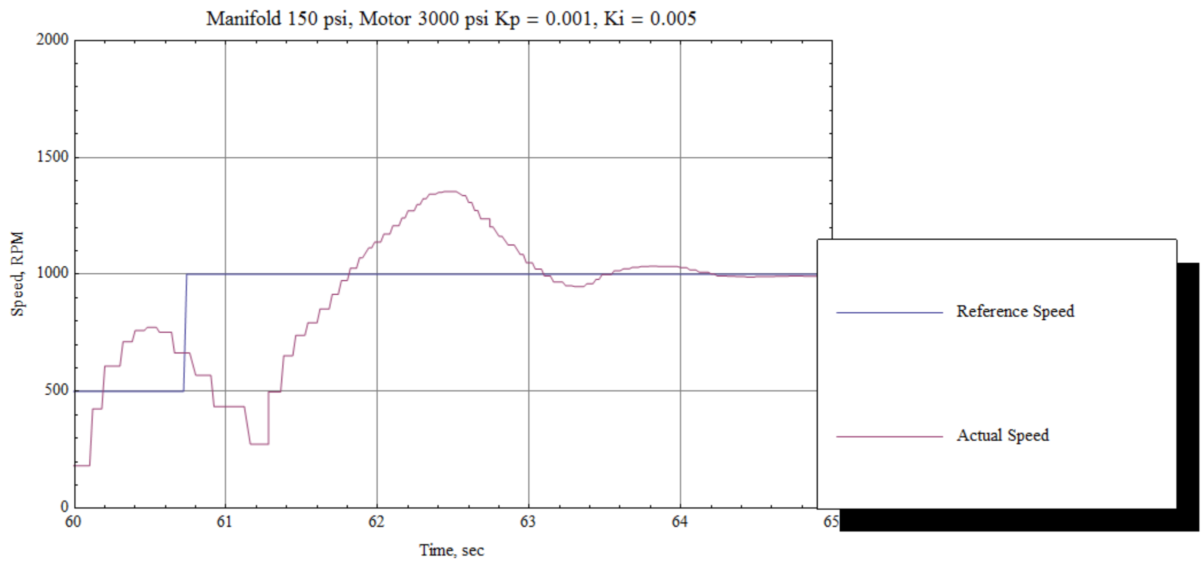

Figure 3.7.2-2 Speed Control Response of Auxiliary Pump System for a Single Speed Step

4.0 PROGRAM SUMMARY

SwRI completed the detailed design of a heavy duty hybrid hydraulic powertrain for a M1078-A1R platform. As part of the vehicle detailed design, a modification package was created for the Allison 7-speed 3700 transmission to allow it to be driven by a hydraulic motor. The existing 3700 transmission was then, characterized, modified, and then tested in a hardware-in-the-loop environment to validate shifting in the new HHV configuration. Using 3-D modeling tools, the designed hybrid hydraulic drivetrain components were then placed in the vehicle chassis. The existing drivetrain components were then removed and the vehicle HHV drivetrain was then installed. A dSPACE microautobox control system was developed and integrated into the HHV system to control the vehicle.

SwRI completed the design of a Hydraulic Tool Power System to allow for the HHV hydraulic system to power a range of off-board hydraulic tools. Using 3-D modeling tools, the system was placed in the HHV vehicle. The system was partially fabricated but not integrated in the vehicle based on the tool power system developmental hydraulic motor system not meeting speed control performance requirements.

The University of Toledo completed two tasks in support of the program. The first task was the development and validation of optimal shift strategy for an Allison 3700 transmission driven by an Eaton hydraulic motor. The second task was the development of speed control systems for HHV auxiliary pumps and the Hydraulic Tool Power System drive hydraulic motor.

Integration of the HHV system into the M1078-A1R was completed and preliminary on-road testing and vehicle shakedown was successful. The HHV was moved to the SwRI heavy duty dynamometer for continued system shakedown and developmental testing. During developmental testing, a hydraulic motor failure occurred. Although not catastrophic, the motor could not be repaired to allow for final fuel economy testing to be completed.

5.0 RECOMENDATIONS

Based on the execution of this program the following five recommendations are made regarding the use of hybrid hydraulic technology for military applications. For each recommendation, a brief description is provided.

1. **Monitor commercial development of hydraulic components.** The hydraulic industry continues to develop advanced technology components and system control advances materially impact what would be considered as fundamental limitations of current hybrid hydraulic components.

2. **Identify high value heavy duty vehicle duty cycles.** Hybrid hydraulic vehicle designs should be considered where vehicle duty cycle contains varying speeds, grades, and start stop, and short duration high power modes.

3. **Ground up hybrid hydraulic drivetrain designs.** Due to the expense and challenges with effectively integrating hybrid hydraulic technology into existing platforms, consider a complete hybrid hydraulic drivetrain as the basis for the next series of TARDEC investigations into hybrid hydraulic technology.

4. **Split power path hybrid hydraulic transmission.** To maximize the benefit of the high power hybrid hydraulic systems as well as limiting the impact of hydraulic losses during high speed or steady state operation, a heavy duty transmission needs to be developed that will allow series hydraulic operation during low speed or transient operation and direct mechanical drive for high speed or constant speed operation.

5. **Series hybrid hydraulic heavy duty testbed.** To quickly be able to test and validate developing hydraulic technology, TARDEC should consider developing a series hybrid hydraulic testbed on a heavy duty platform to quickly be able to implement and test and validate new hydraulic technology and components as they come to market.

APPENDIX A. Eaton Efficiency Data

Table A-1. Eaton Efficiency Data for Brevini Pump/Motors in Pumping Mode

			Std			Brevini 5-2 w/ feedback			Brevini 5-3 w/ feedback		
Displacement	Speed	Pressure	Mech	Vol	Overall	Mech	Vol	Overall	Mech	Vol	Overall
25	500	2500				94.1	81.5	76.7	94.2	85.5	80.5
25	1125	3771	91.4	82.0	75.0	94.5	80.4	76.0	93.9	84.0	78.9
25	1125	4351	92.0	76.6	70.5	93.8	74.8	70.1	94.2	79.4	74.8
25	1125	5076	92.6	70.1	64.9	94.0	65.7	61.7	94.2	72.7	68.5
50	1000	4000				95.6	89.5	85.6	95.6	91.5	87.5
50	1125	3771	94.4	92.6	87.4	95.6	91.8	87.8	95.5	92.8	88.5
50	1125	4351	94.8	90.0	85.4	95.6	88.7	84.8	95.8	91.0	87.2
50	1125	5076	95.3	86.7	82.6	96.1	84.9	81.6	95.8	87.0	83.3
75	1125	3771	94.4	95.3	90.0	95.1	95.2	90.5	95.3	95.5	91.0
75	1125	4351	94.8	93.9	89.0	95.5	93.5	89.3	95.6	93.9	89.8
75	1125	5076	95.2	91.9	87.4	95.7	91.1	87.1	95.7	91.9	88.0
75	1500	5000				94.4	92.9	87.7	94.7	93.2	88.2
75	1600	3771	93.1	96.3	89.7	92.9	95.6	88.9	93.5	95.5	89.2
75	1600	4351	93.8	95.4	89.4	93.9	94.2	88.4	94.0	94.5	88.8
75	1600	5076	94.4	93.8	88.5	94.5	92.3	87.3	94.6	93.1	88.0
75	1875	2321	88.6	98.2	87.0	90.0	97.9	88.1	90.1	97.4	87.8
75	1875	2901	90.6	97.7	88.4	91.6	96.9	88.7	91.4	96.9	88.6
75	1875	3626	92.1	96.6	89.0	92.8	95.4	88.5	92.7	95.9	88.9
75	1875	3771	92.3	96.4	89.0	92.6	96.3	89.1	93.0	95.6	88.9
75	1875	4351	93.0	95.4	88.8	93.5	94.2	88.1	93.7	94.9	88.9
75	1875	5076	93.8	93.9	88.1	94.0	92.8	87.2	94.4	93.4	88.1
100	1125	3771	90.3	97.1	87.7	93.2	97.4	90.7	93.7	97.6	91.4
100	1125	4351	90.8	96.2	87.3	93.7	96.5	90.4	94.1	96.8	91.1
100	1125	5076	91.2	94.7	86.3	94.0	94.9	89.2	94.5	95.5	90.3
100	1600	3771	90.7	97.6	88.6	90.8	97.5	88.5	90.9	97.6	88.7
100	1600	4351	91.6	96.8	88.7	91.5	96.9	88.7	91.8	97.0	89.0
100	1600	5076	92.3	95.9	88.5	92.2	96.2	88.6	92.5	96.1	88.9
100	1875	2321	84.0	98.8	83.0	84.7	98.9	83.8	84.3	98.6	83.2
100	1875	2901	86.7	98.5	85.4	86.9	98.0	85.1	86.8	98.3	85.3
100	1875	3626	88.7	98.0	86.9	88.9	97.7	86.8	88.9	97.8	86.9
100	1875	3771	89.0	97.8	87.0	89.1	97.8	87.1	89.2	97.7	87.1
100	1875	4351	90.2	97.2	87.7	90.1	97.3	87.7	90.4	97.1	87.8
100	1875	5076	91.2	96.4	88.0	91.0	96.4	87.7	91.5	96.5	88.2
100	2000	6500				91.8	94.7	86.9	92.2	94.8	87.3
100	2500	6500				90.7	91.5	83.0	90.8	90.8	82.4

Table A-2Eaton Efficiency Data for Brevini Pump/Motors in Motoring Mode

Displacement	Speed	Pressure	Standard			Brevini 5-2 w/ feedback			Brevini 5-3 w/ feedback		
			Mech	Vol	Overall	Mech	Vol	Overall	Mech	Vol	Overall
25	500	2500				11.19	83.03	9.29	35.68	81.43	29.05
25	500	3771	81.50	73.54	59.94	86.76	70.70	61.34	90.49	69.47	62.86
25	500	4496	72.10	65.46	47.20	72.23	61.84	44.67	88.16	61.67	54.37
25	500	5366	71.13	55.82	39.70	71.20	53.49	38.08	71.88	50.36	36.20
25	500	5511	70.87	53.25	37.74	71.00	50.95	36.17	71.57	49.13	35.16
25	500	5947			0.00	70.48	46.91	33.06	70.98	44.31	31.45
25	500	6527			0.00	70.23	42.75	30.02	69.77	38.07	26.56
25	1500	3771	86.73	86.39	74.93	88.57	86.87	76.94	90.65	84.11	76.25
25	1500	4496	86.84	82.00	71.21	88.35	83.26	73.56	91.08	79.24	72.17
25	1500	5366	74.00	74.35	55.02	88.32	77.11	68.10	89.93	72.84	65.51
25	1500	5511	73.93	74.08	54.77	88.53	76.15	67.42	89.71	71.61	64.24
25	1500	5947	74.36	71.50	53.17	88.18	72.93	64.31	89.15	68.71	61.25
25	1500	6527			0.00	87.79	68.72	60.33	75.04	60.83	45.65
25	2500	3771	86.26	90.19	77.80	79.33	91.08	72.25	82.67	87.67	72.48
25	2500	4496	86.99	87.02	75.70	81.96	87.40	71.63	83.25	83.79	69.76
25	2500	5366	87.16	82.78	72.15	86.77	82.58	71.65	87.76	79.14	69.45
50	500	3771	82.69	82.13	67.91	91.31	84.95	77.57	92.44	81.86	75.67
50	500	4496	82.49	76.78	63.34	91.00	81.12	73.82	91.66	75.31	69.03
50	500	5366	81.42	69.05	56.22	83.78	72.54	60.77	91.25	69.96	63.84
50	500	5511	81.76	68.52	56.02	83.76	71.14	59.59	90.39	67.42	60.94
50	500	5947	81.29	65.59	53.32	83.41	67.68	56.45	84.54	62.90	53.18
50	500	6527			0.00	83.06	63.04	52.36	83.66	57.60	48.19
50	1000	4000				91.93	89.83	82.58	93.57	87.53	81.90
50	1500	3771	90.54	92.12	83.41	91.09	93.17	84.87	92.63	91.70	84.94
50	1500	4496	90.72	89.66	81.34	91.17	90.66	82.65	92.79	89.01	82.59
50	1500	5366	84.49	85.21	71.99	91.25	87.28	79.64	92.25	84.69	78.13
50	1500	5511	84.75	84.90	71.95	91.37	86.58	79.11	92.63	84.64	78.40
50	1500	5947	84.70	82.61	69.97	91.30	84.97	77.58	92.26	82.42	76.04
50	1500	6527			0.00	91.04	82.38	75.00	91.62	78.93	72.32
50	2500	3771	88.90	94.02	83.58	89.77	95.00	85.28	90.03	94.41	85.00
50	2500	4496	89.62	92.40	82.81	89.27	93.42	83.40	90.34	92.88	83.91
50	2500	5366	90.08	89.74	80.84	90.35	91.07	82.28	89.73	89.86	80.63
75	1500	4496	86.91	92.25	80.17	92.41	94.27	87.11	92.78	92.98	86.27
75	1500	5000				92.62	93.04	86.17	92.84	91.21	84.68
75	2500	6500				91.15	91.73	83.61	88.91	89.81	79.85
85	2500	6500				89.95	92.55	83.25	89.59	91.59	82.06
100	2000	6500				92.27	91.14	84.09	90.99	90.63	82.46
100	2500	6339				88.98	91.92	81.79	83.99	91.78	77.09

APPENDIX B. DRIVE CYCLES

The drive cycles used for the vehicle dynamometer testing are shown below. Some drive cycles are defined by vehicle speed as a function of time. Other drive cycles are defined by vehicle distance travelled as a function of time. Additionally, drive cycle grade will also be shown for those drive cycles that include grade. Some drive cycles begin at a flying start; *i.e.*, the drive cycles starts with the vehicle already in motion. Additionally, some of the military related drive cycles have maps accompanying them to show the vehicle course being simulated on the dynamometer.

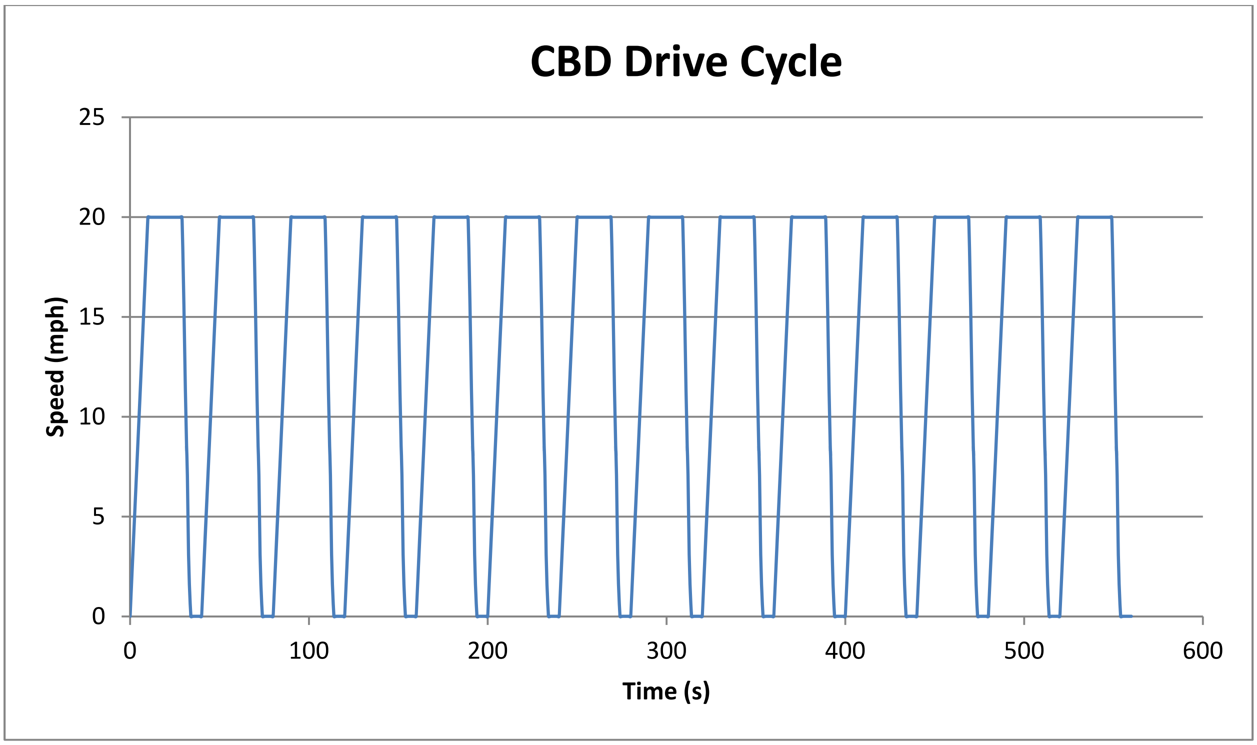

Figure B.1. CBD Drive Cycle

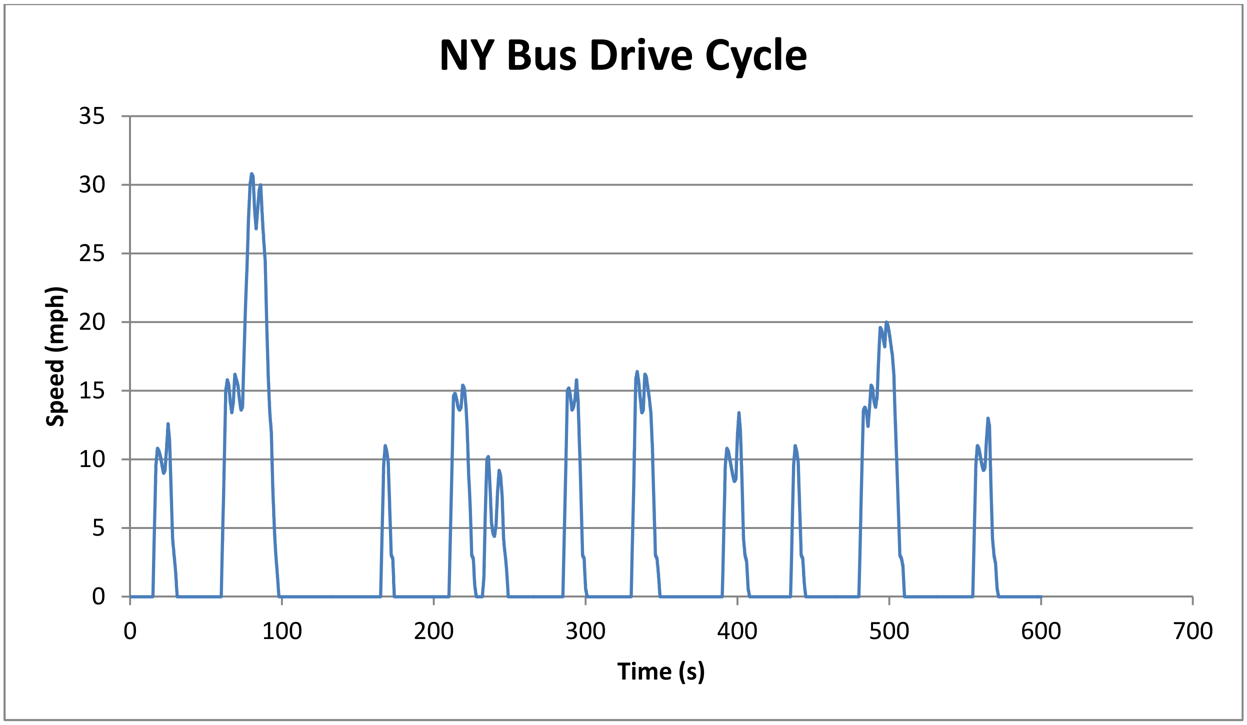

Figure B.2. NY Bus Drive Cycle

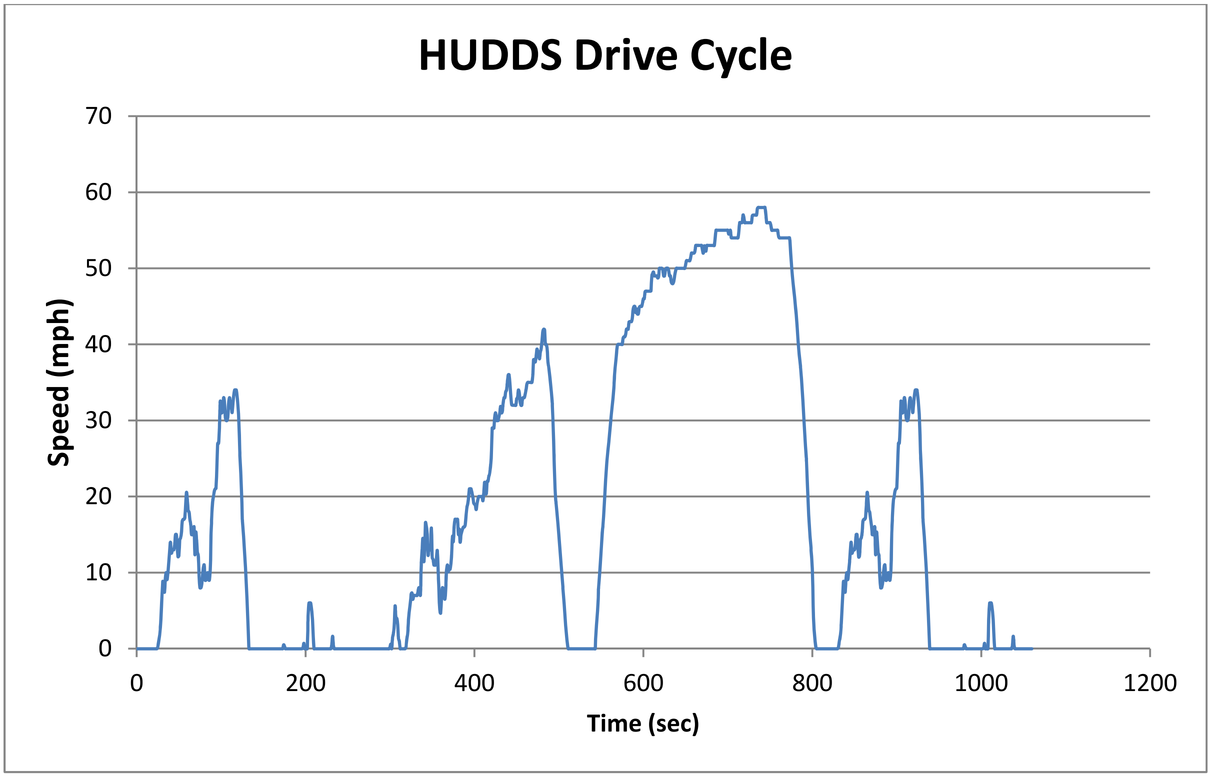

Figure B.3. HUDDS Drive Cycle

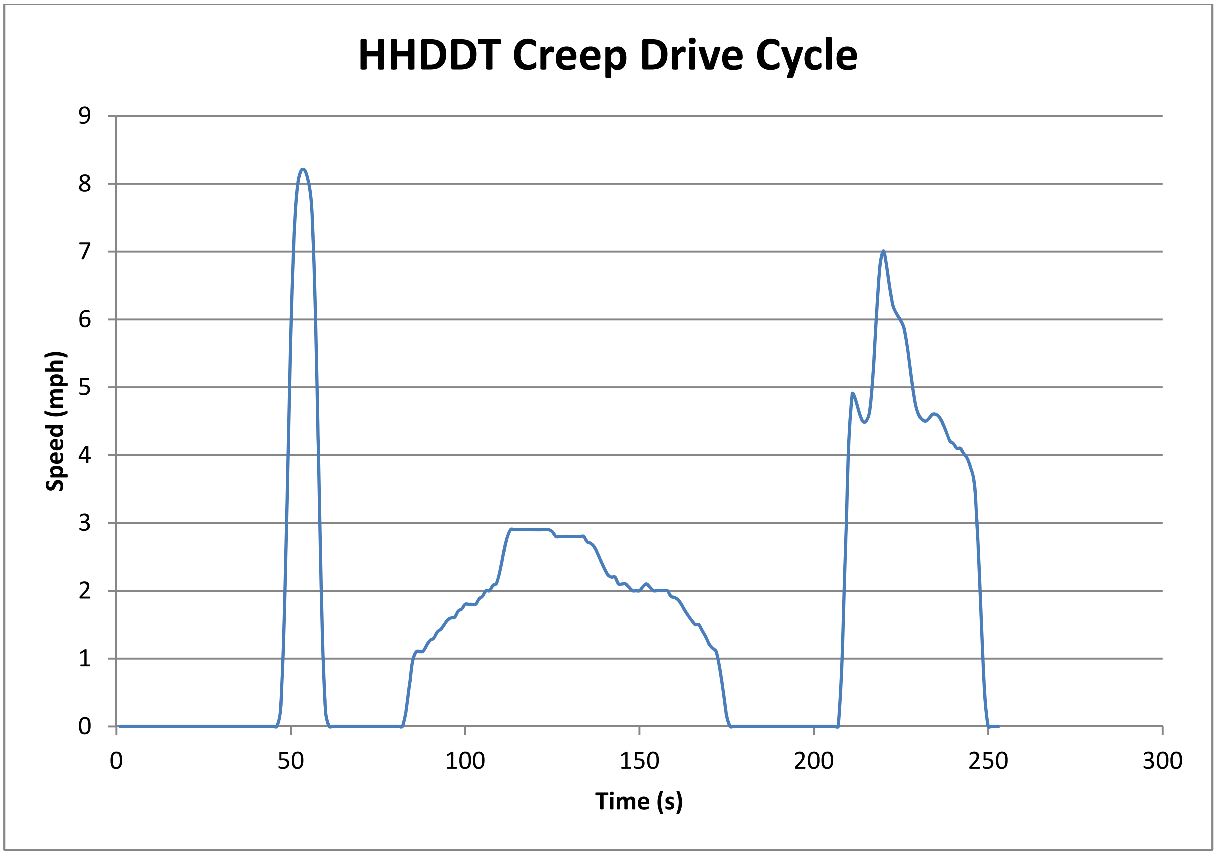

Figure B.4. HHDDT Creep Drive Cycle

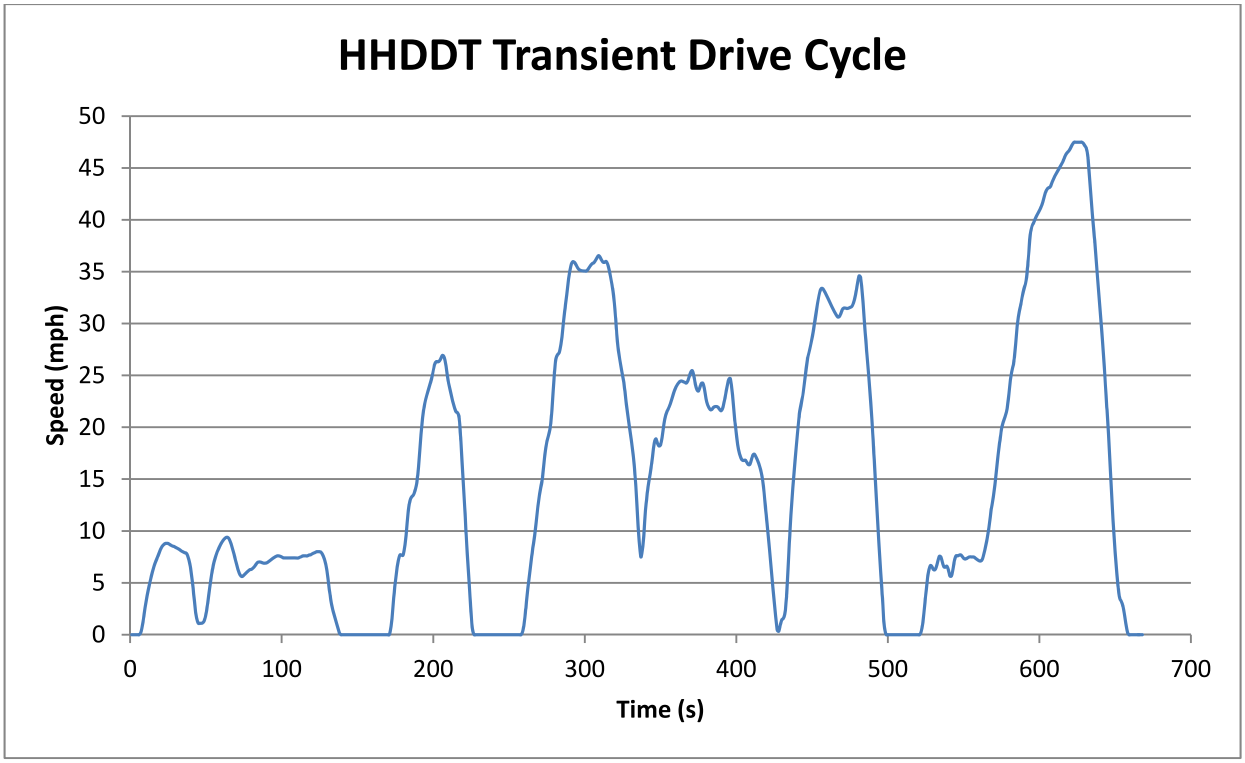

Figure B.5. HHDDT Transient Drive Cycle

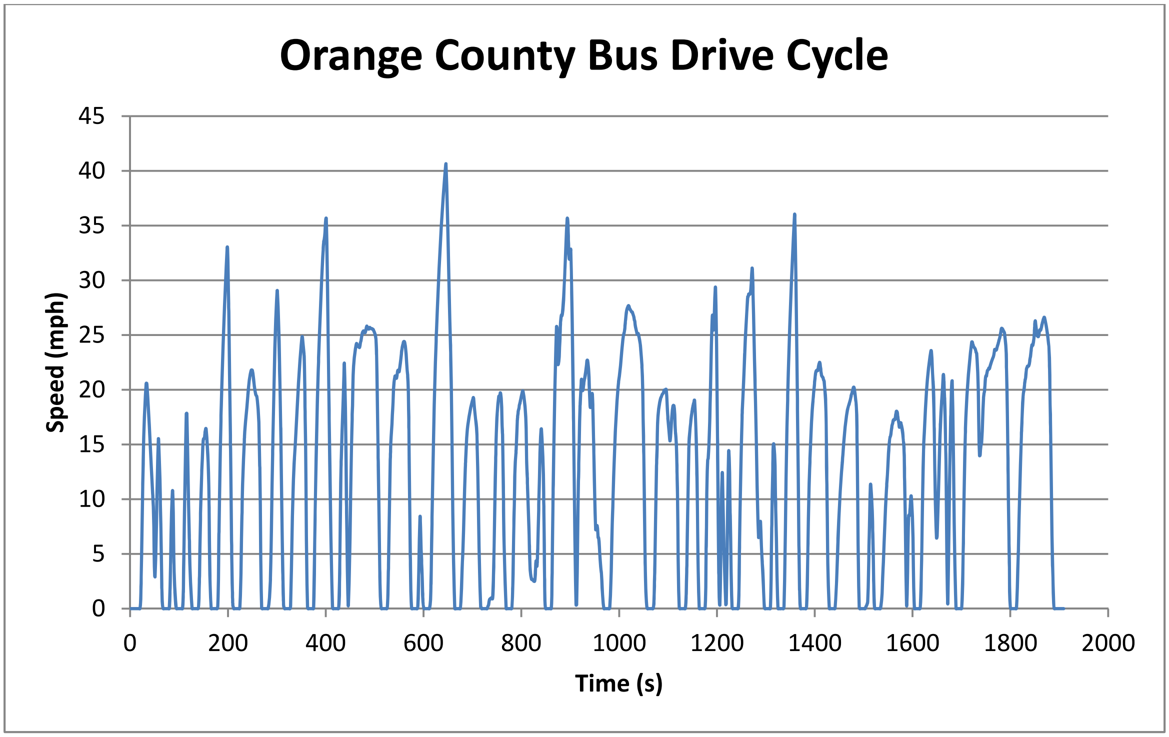

Figure B.6. Orange County Bus Drive Cycle

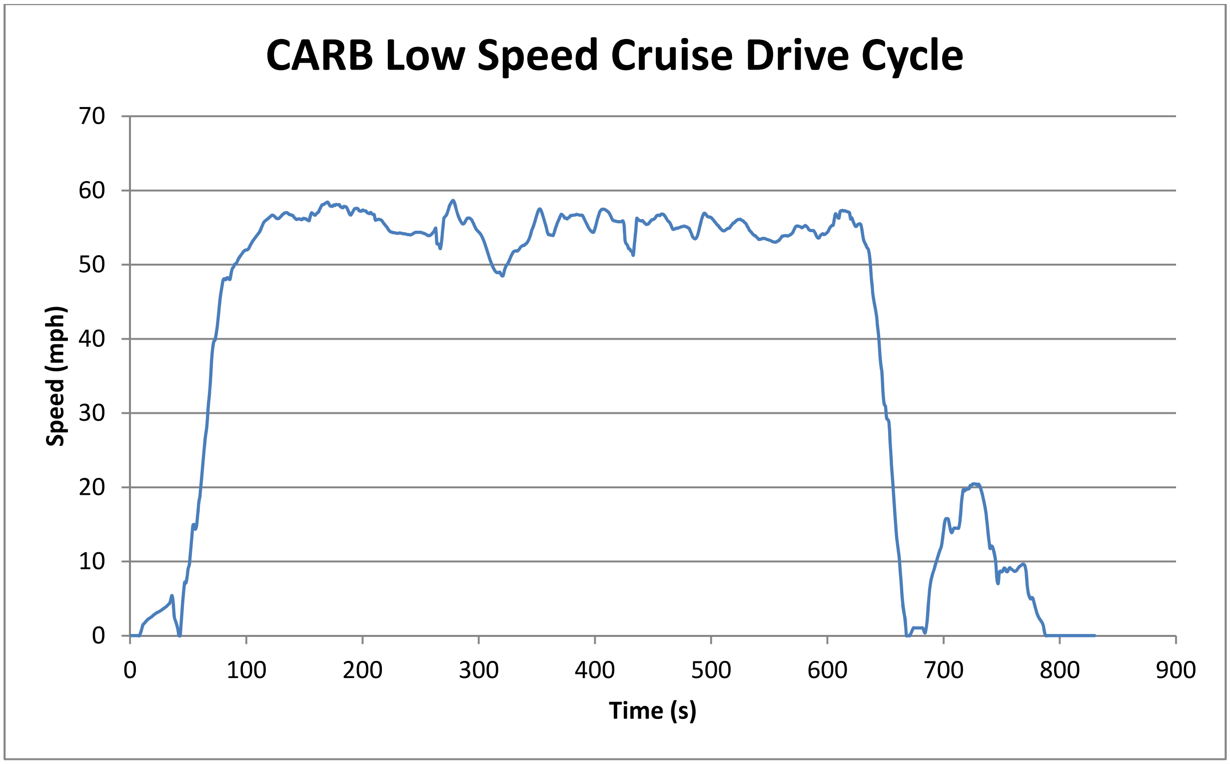

Figure B.7. CARB Low Speed Cruise Drive Cycle

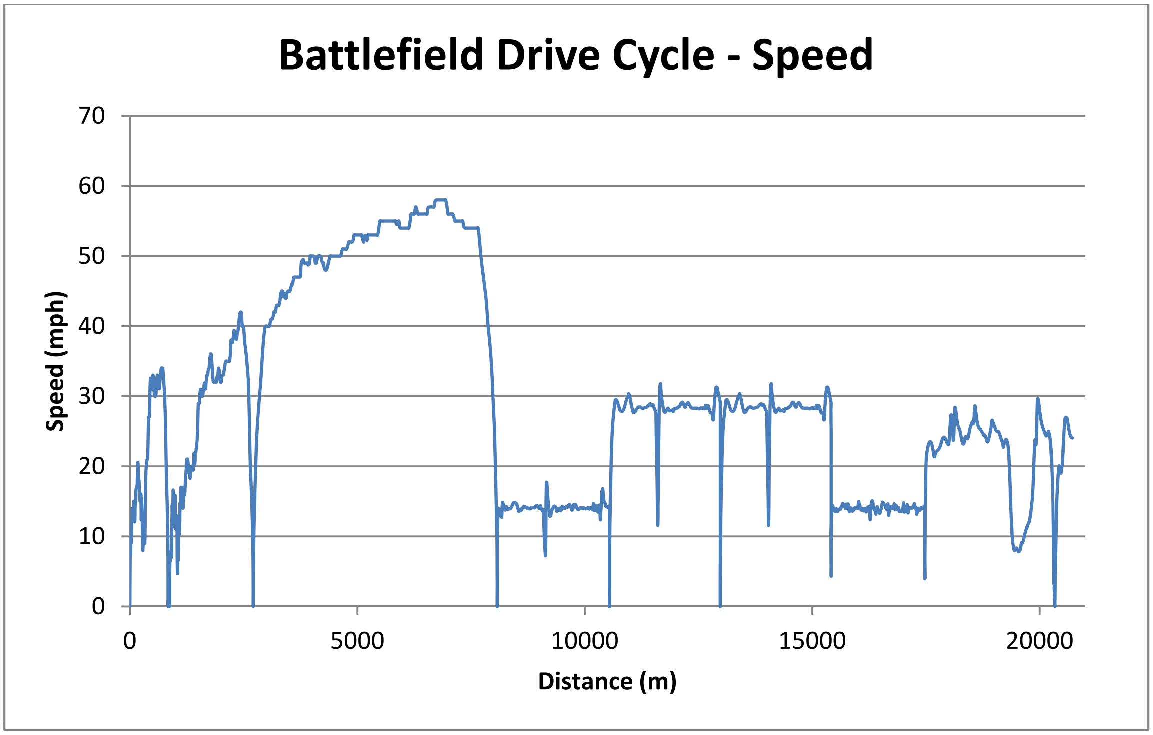

Figure B.8. Battlefield Drive Cycle - Speed

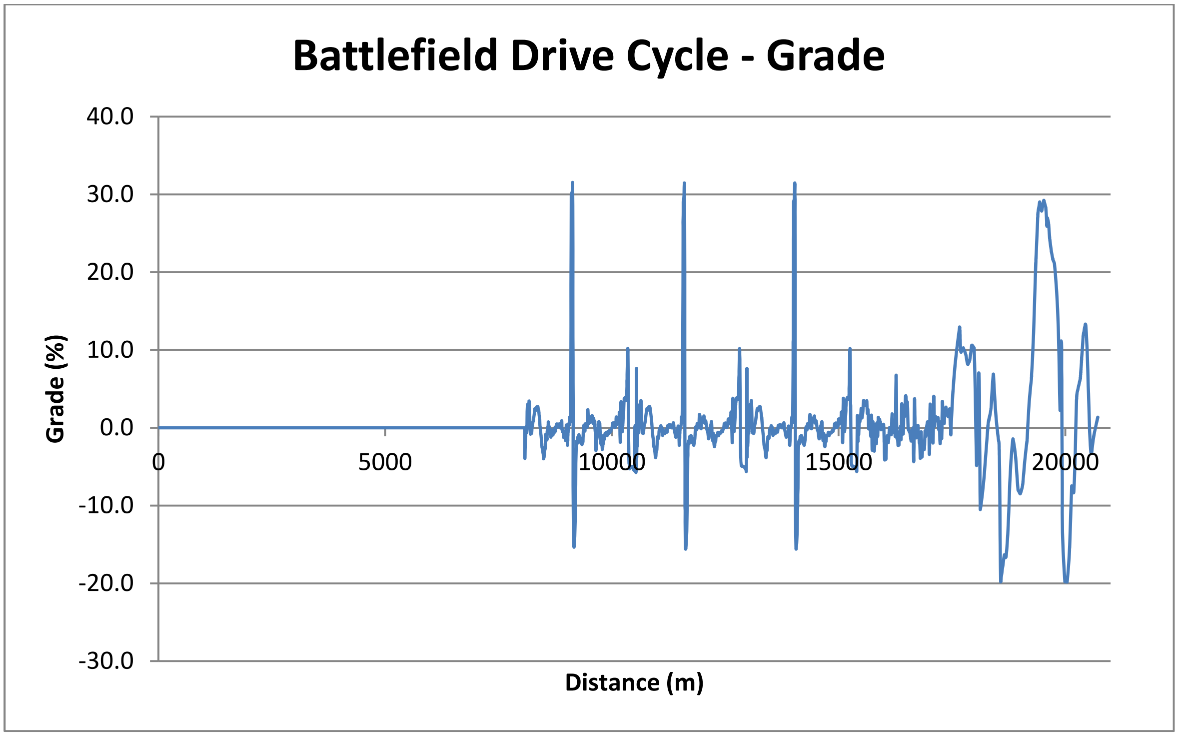

Figure B.9. Battlefield Drive Cycle - Grade

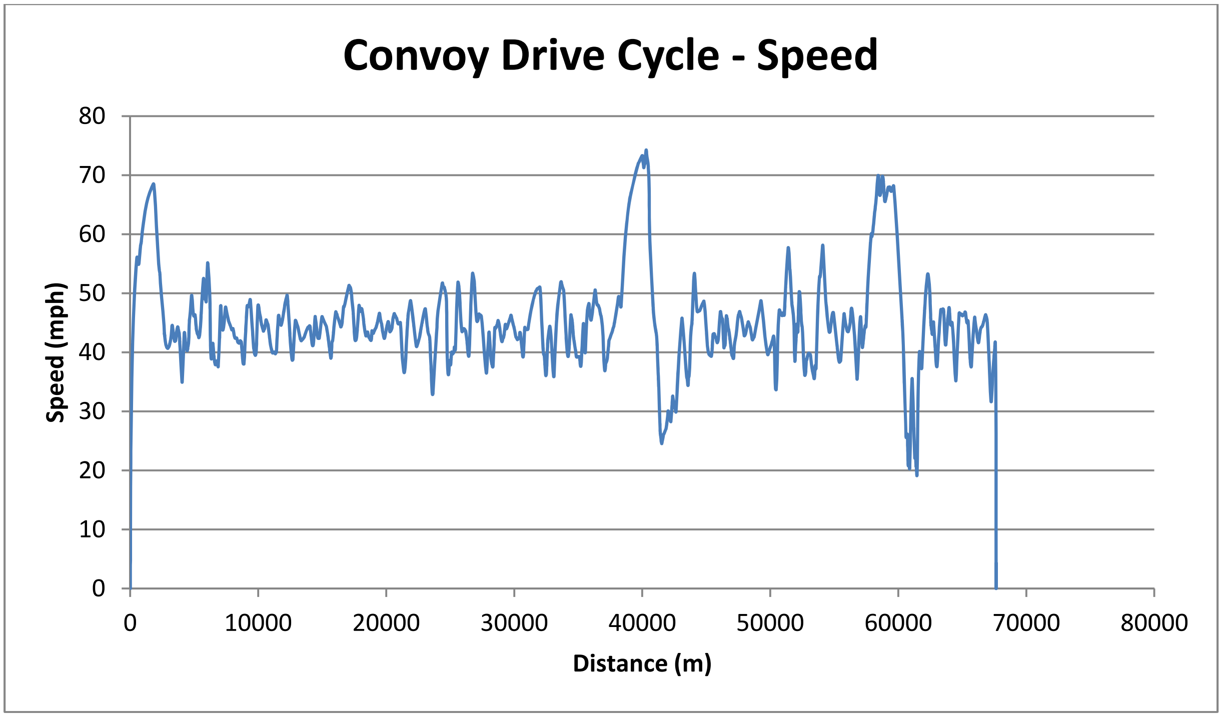

Figure B.10. Convoy Drive Cycle - Speed

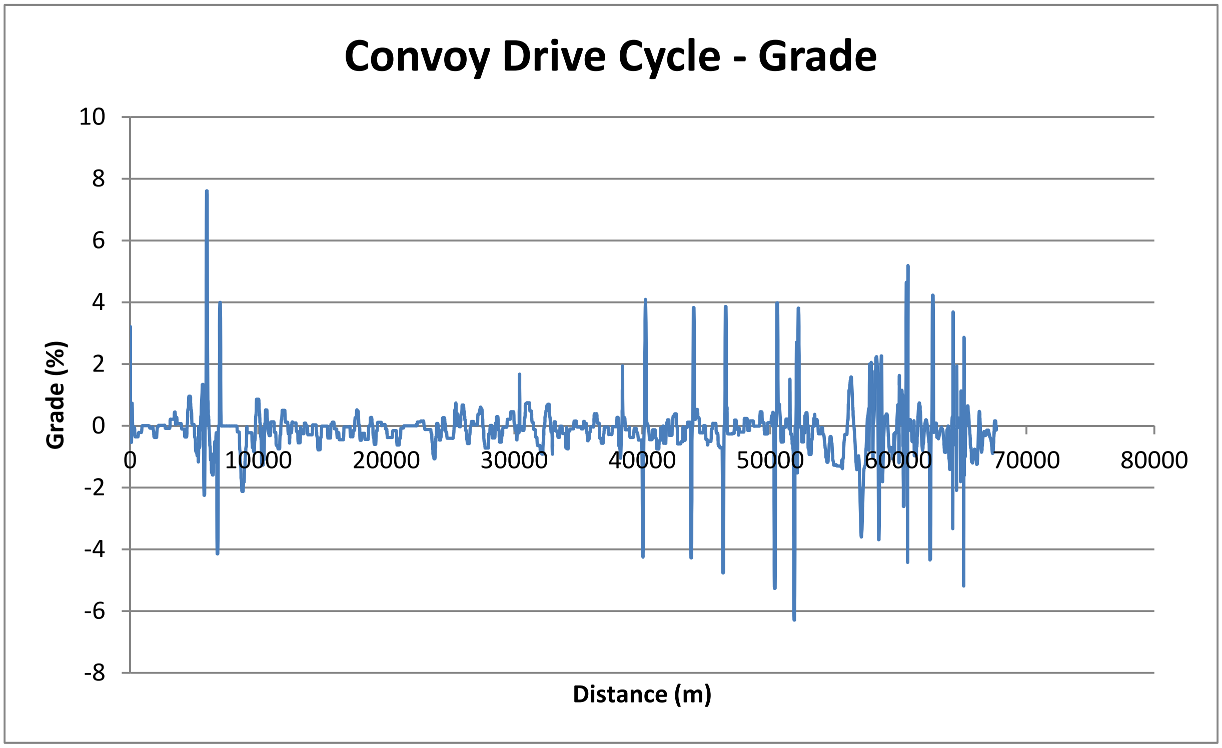

Figure B.11. Convoy Drive Cycle - Grade

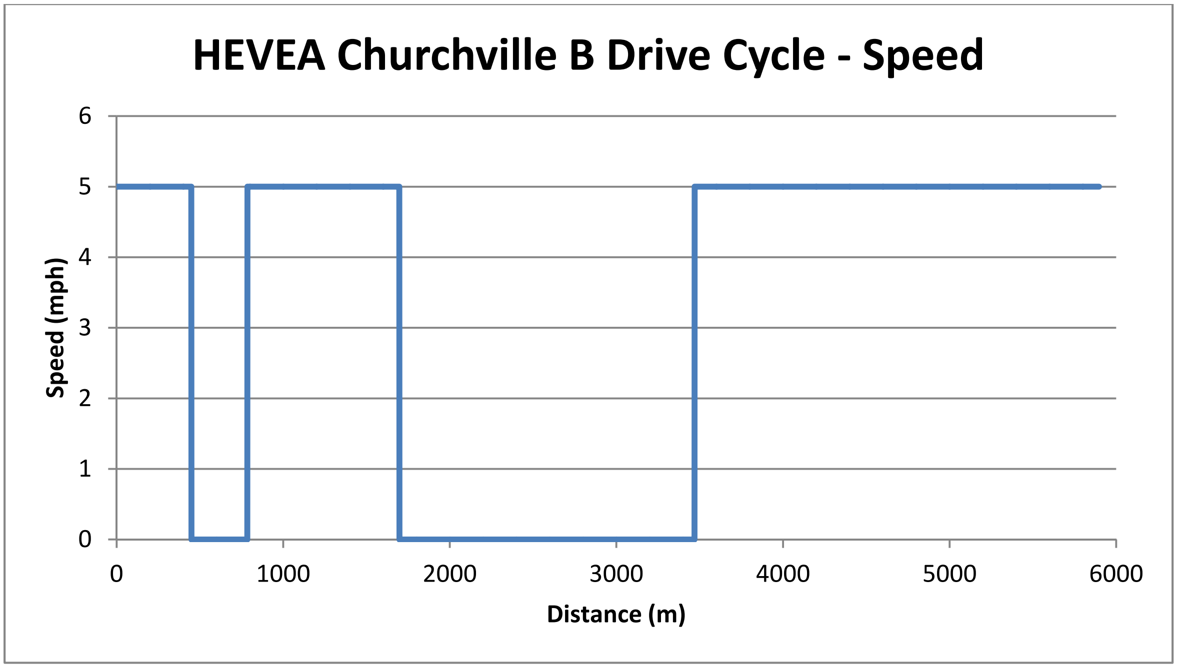

Figure B.12. HEVEA Churchville B Drive Cycle - Speed

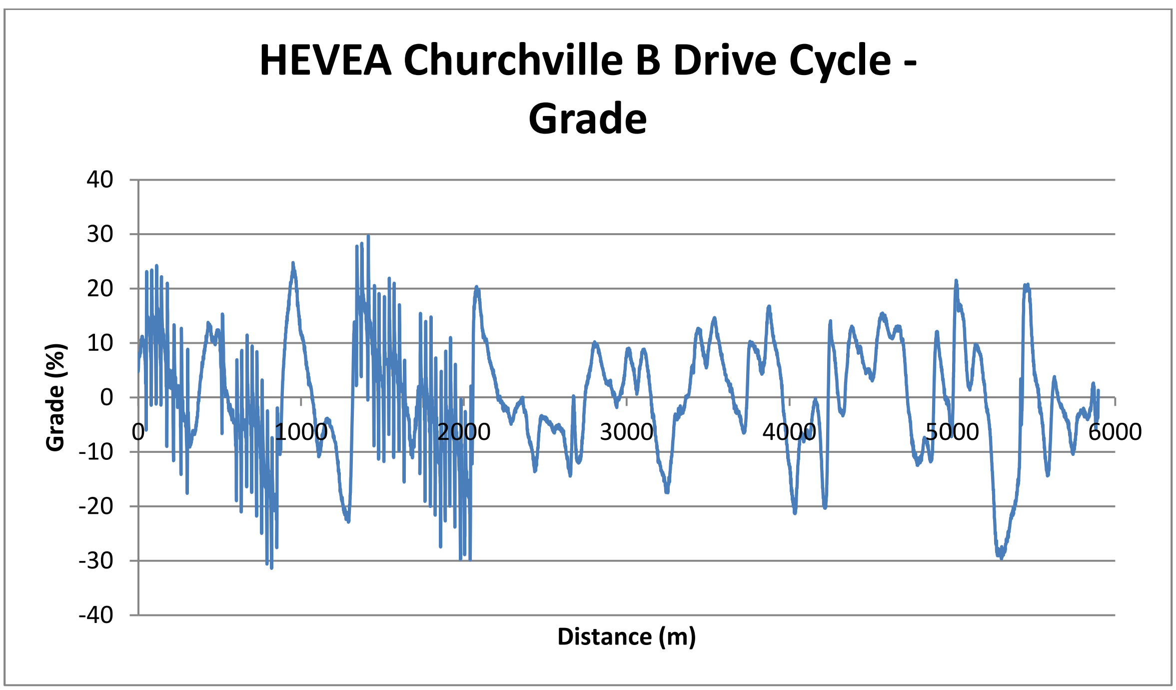

Figure B.13. HEVEA Churchville B Drive Cycle - Grade

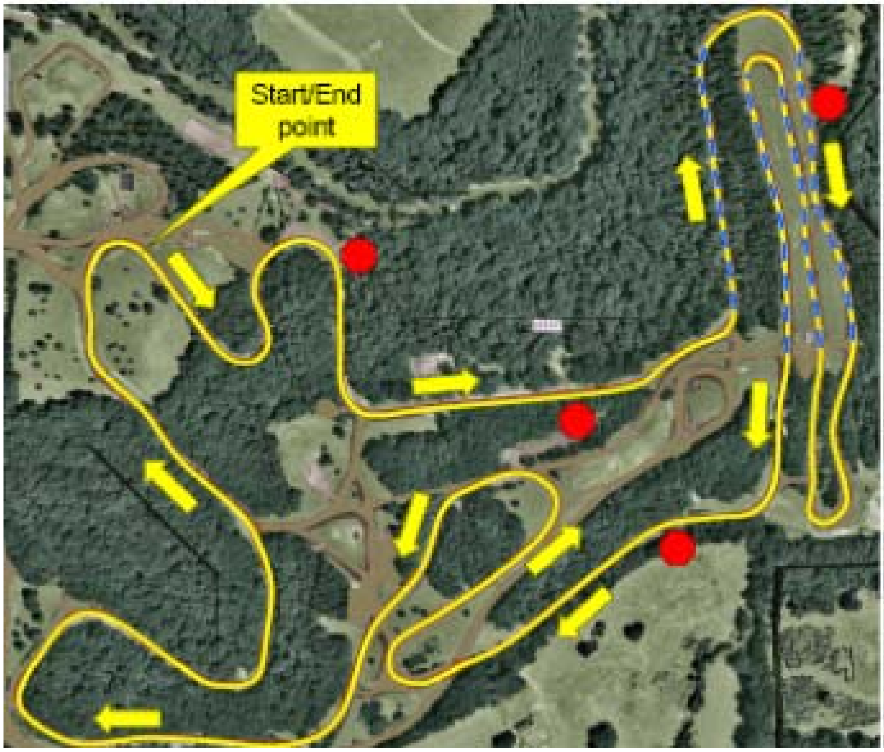

Figure B.14. Map of HEVEA Churchville B Drive Cycle

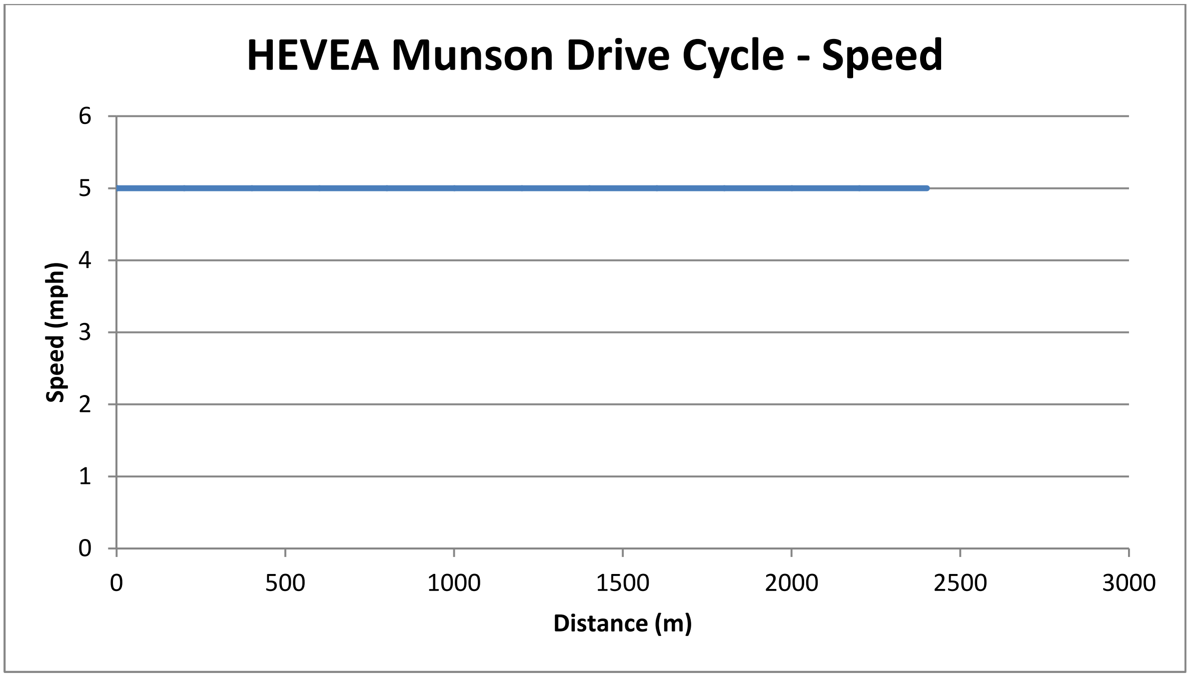

Figure B.15. HEVEA Munson Drive Cycle - Speed

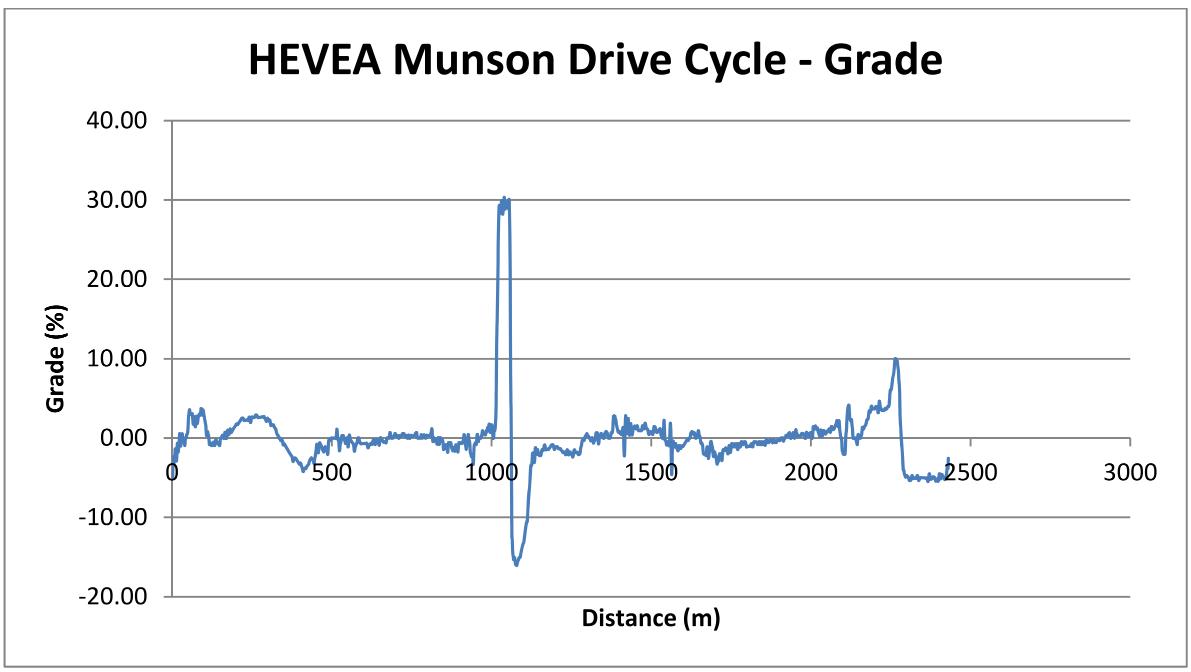

Figure B.16. HEVEA Munson Drive Cycle - Grade

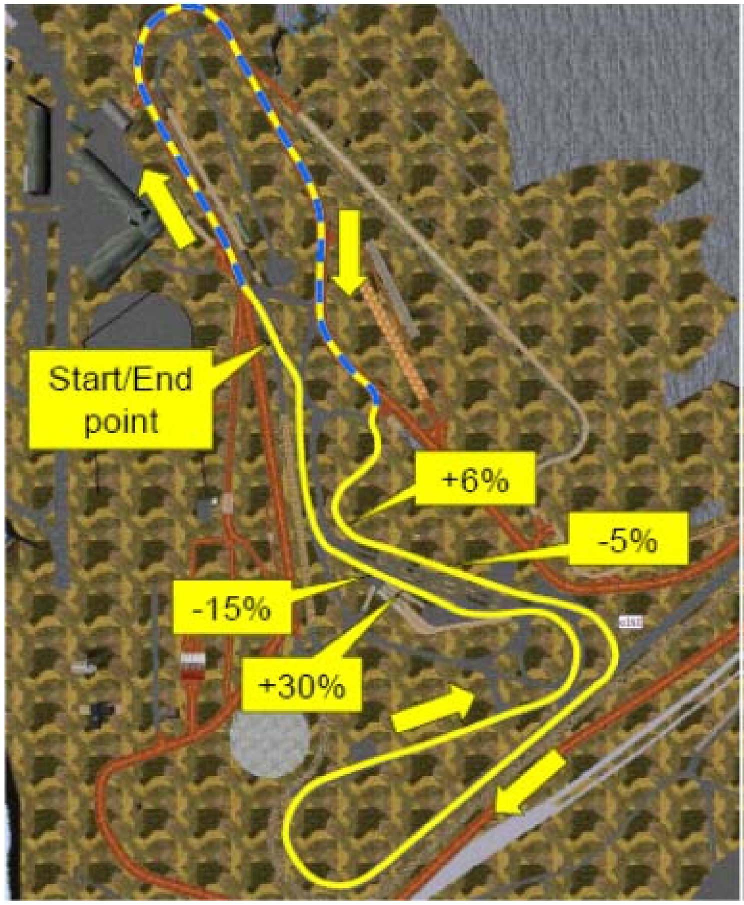

Figure B.17. Map of HEVEA Munson Drive Cycle

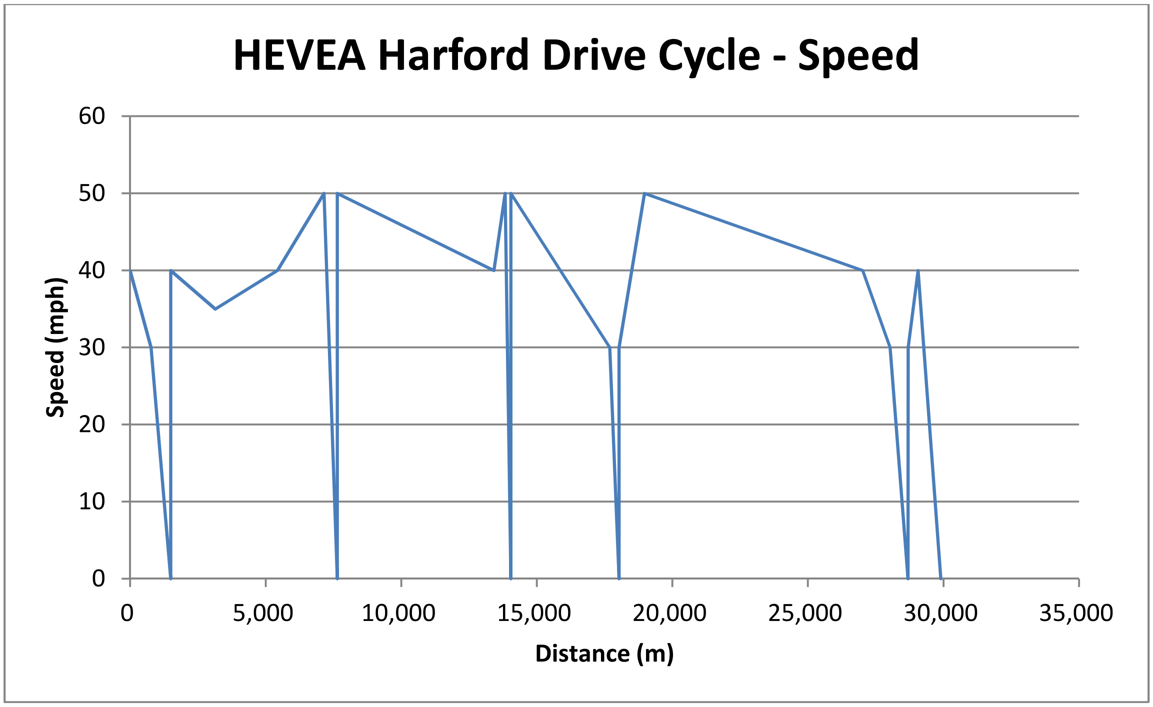

Figure B.18. HEVEA Harford Drive Cycle - Speed

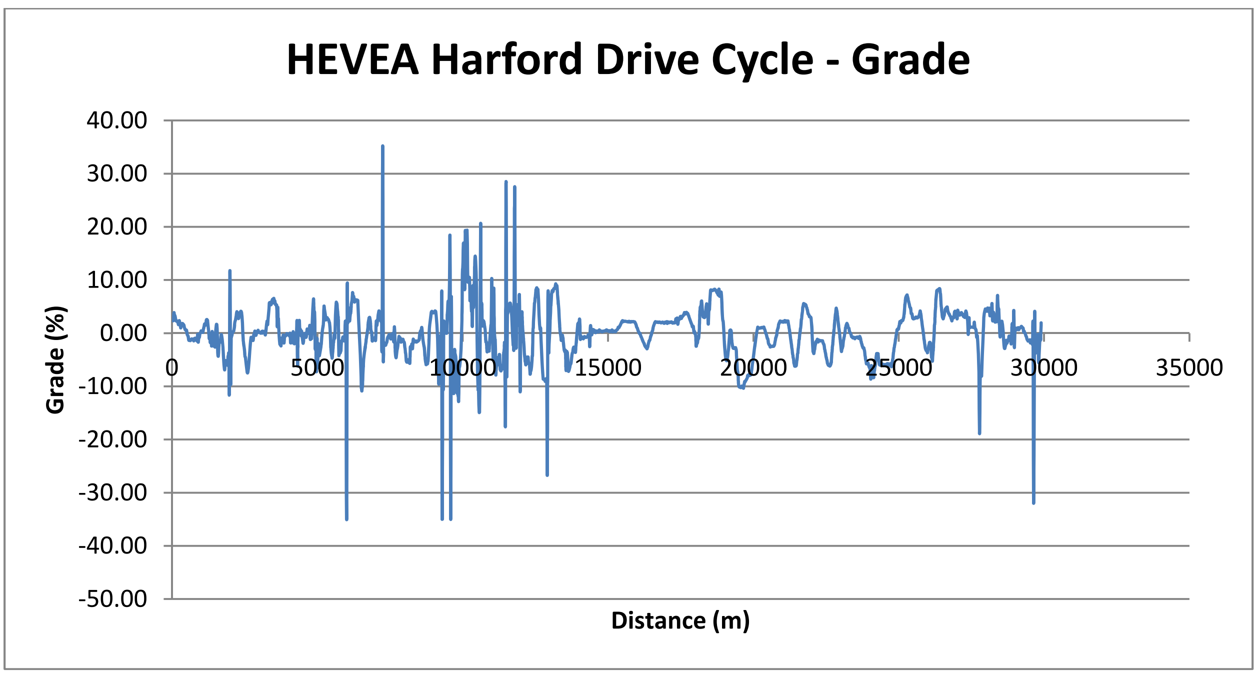

Figure B.19. HEVEA Harford Drive Cycle - Grade

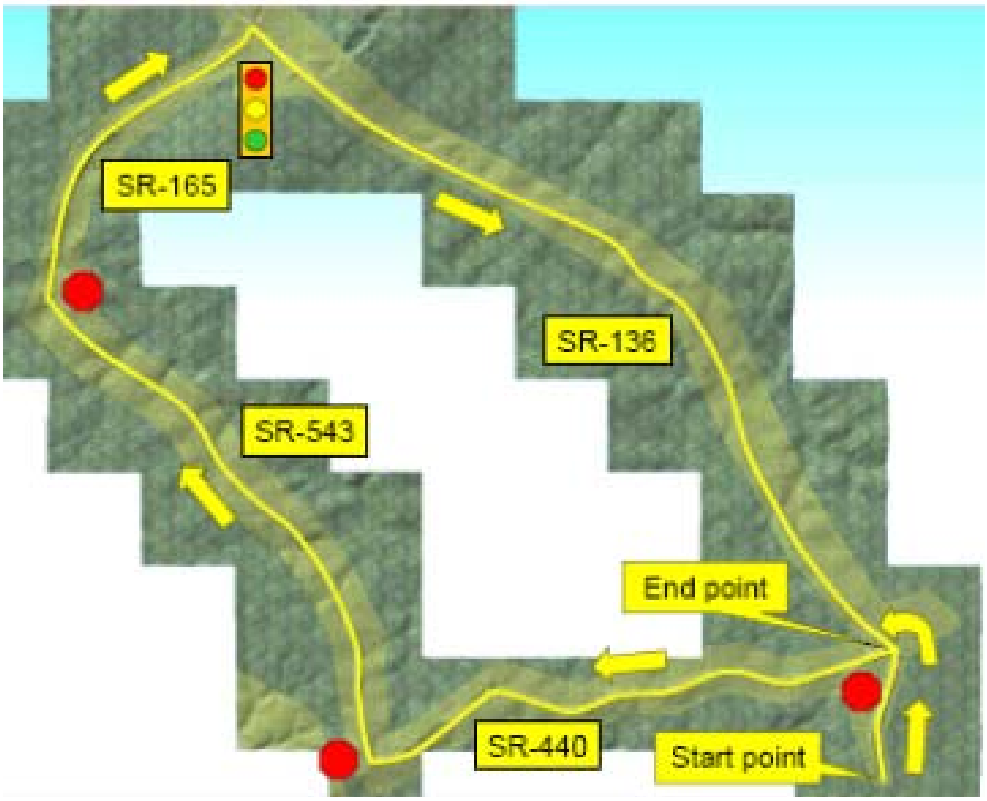

Figure B.20. Map of HEVEA Harford Drive Cycle

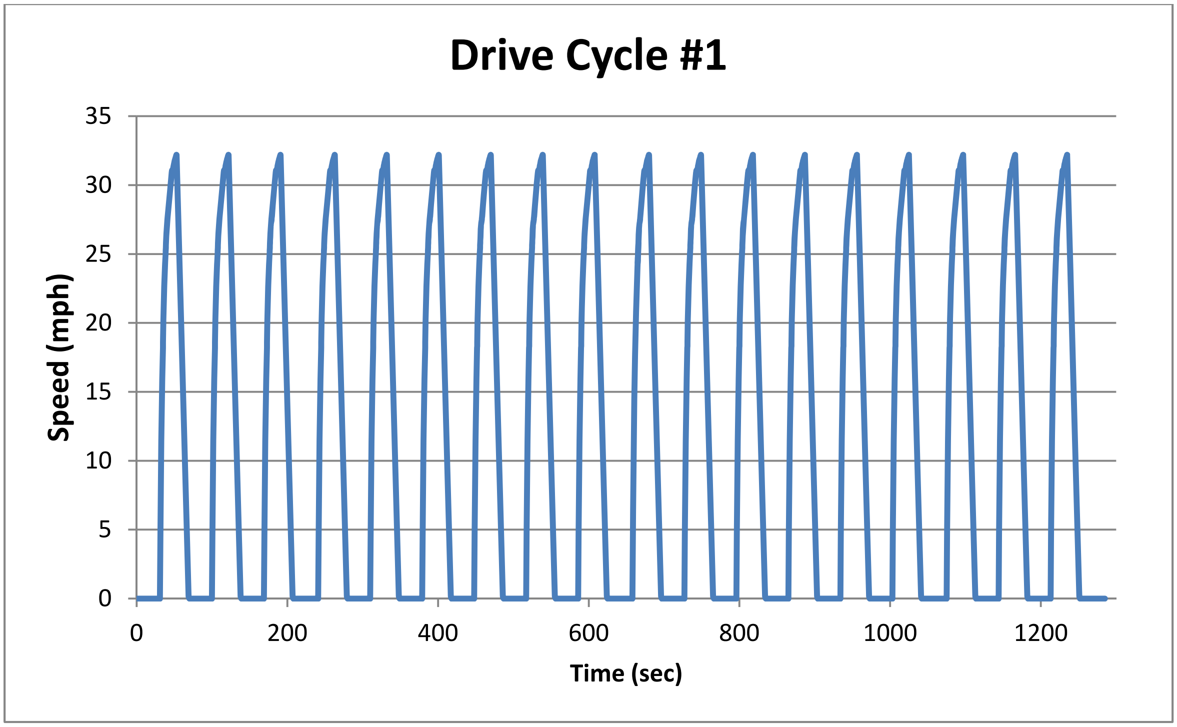

Figure B.21. Drive Cycle #1

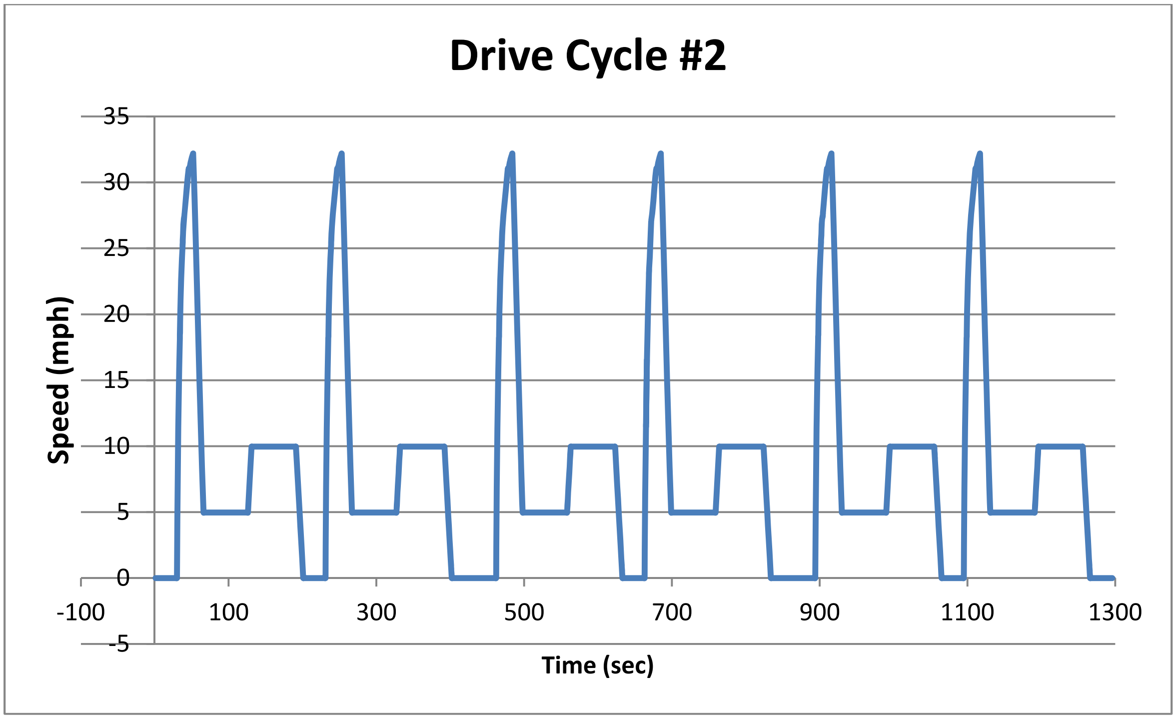

Figure B.22. Drive Cycle #2