Design and Construction of a Hydrostatic Dynamometer for Testing a Hydraulic Hybrid Vehicle

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Abstract

A hydrostatic dynamometer has been constructed to test a hydraulic hybrid passenger vehicle. Hybrid vehicles reduce their fuel consumption by capturing energy normally lost during braking and reusing it during acceleration in addition to managing the engine to enable it to run at a higher efficiency operating point. The dynamometer is designed to test a particular hybrid vehicle which stores and transmits energy hydraulically. Measurement of fuel consumption while the vehicle completes standard drive cycles is necessary to refine and validate the performance and efficiency of the vehicle. The dynamometer provides repeatable, convenient, inexpensive, safe, and flexible indoor testing for the vehicle.

The dynamometer connects directly to the output shaft of the vehicle's transmission. It is mounted on a bedplate installed behind the vehicle. The dynamometer's hydraulic pump loads or motors the vehicle to simulate driving. It can be controlled either manually or automatically. Automatic controls allow the dynamometer to calculate and apply the appropriate load based on vehicle speed.

MATLAB and Simulink simulations aided the design of the dynamometer. A MATLAB simulation of the vehicle determined torque and speed requirements for standard drive cycles. Another MATLAB simulation calculated pressures, flow rates, and energy storage requirements on the dynamometer to size components. A Simulink simulation aided controls development.

The dynamometer has demonstrated open and closed loop performance with and without load. It has demonstrated fast torque tracking. However, vehicle reliability issues have prevented drive cycle tests from being completed.

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1 Introduction

The purpose of this chapter is to introduce the project and explain the motivation for it. It describes the dynamometer and the vehicle it is designed to test in section 1.1. It describes the goals of the project in section 1.2. This chapter also summarizes some other hydrostatic and non-hydrostatic dynamometers designed to test vehicles in section 1.3. An outline of the rest of this thesis appears in section 1.4.

1.1 Background

The Center for Compact and Efficient Fluid Power (CCEFP) is developing a prototype hydraulic hybrid vehicle on a utility vehicle chassis. This vehicle is named the Generation 1 Hydraulic Hybrid Passenger Vehicle (HHPV) and is shown in Figure 1.1.

Figure 1.1: The Generation 1 HHPV

The goal of this project is to demonstrate fuel savings by using a hydraulic hybrid powertrain. Simulations show that fuel economies of over 60 miles per gallon are possible from this particular vehicle.

A hybrid vehicle stores energy from braking and when power demanded at the wheels is less than what the engine is able to produce. It releases this energy during acceleration when demand peaks to supplement engine power. This allows the engine to be downsized. In a hybrid vehicle, the engine can be sized to produce the average power requirement rather than the highest power necessary for driving. The engine operates with increased efficiency because it produces close to its peak power during most of its duty cycle.

Hybrid vehicles commonly store energy in two ways: electrically or hydraulically. Electric hybrids store energy in batteries and are commonly available as passenger vehicles. Hydraulic hybrids store energy by compressing a gas in an accumulator. Hydraulic components offer greater power density than electric components, but the energy storage density of hydraulic accumulators is less than that of batteries. Therefore, a limited selection of commercially available hydraulic hybrid heavy trucks exists, but the technology has not yet penetrated the passenger vehicle market.

Hydraulic hybrid powertrains can be built in three classes of configurations: series, parallel, and power split. Each configuration requires an accumulator to store energy. In the series configuration, the engine drives a pump which sends fluid to spin a motor connected to the wheels. The engine is not mechanically linked to the wheels. This configuration loses energy as a result of the transmission of all energy through hydraulics. The advantage of this architecture is that it allows for excellent engine management with a simple control strategy [1]. In the parallel configuration, the engine drives the wheels directly through a driveshaft, and a single pump motor is coupled to the driveshaft. This configuration offers excellent power transmission efficiency, but the engine does not operate at its most efficient point. For example, Stecki et al. [2] implemented a parallel hydraulic hybrid system in a Freightliner truck. Using hydraulic power to supplement engine power at the wheels, they achieved an average 37% fuel savings on an urban drive cycle.

The CCEFP has chosen to pursue an "input coupled" power split configuration [3]. A block diagram appears in Figure 1.2. A planetary gear train splits engine power to travel both through hydraulics and a driveshaft. The "torquer" pump motor is geared directly to the engine, and its displacement can be varied to change the torque on the engine. The "speeder" pump motor is connected to the engine and wheels through a planetary gear train, and its displacement can be varied to change their speed. Engine management improves because the engine speed does not depend directly on wheel speed, yet transmission losses are reduced because some of the power travels through a driveshaft. Little practical demonstration of the power split architecture in a hydraulic hybrid passenger vehicle exists, so the CCEFP is building a prototype vehicle and creating simulations based on this hardware. The power split architecture also allows fuller control flexibility than the series or parallel architectures.

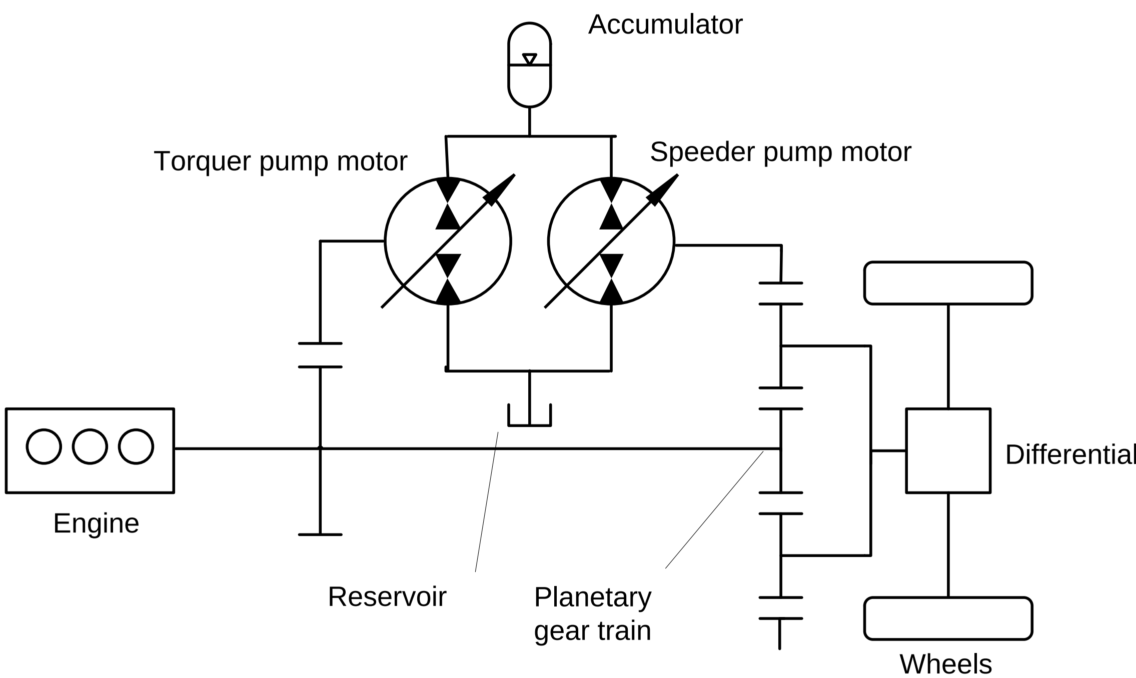

Figure 1.2: Input coupled power split architecture

Simulations can predict fuel economy and performance of the vehicle. However, measuring these parameters on the physical vehicle is difficult and unreliable without a dynamometer. A dynamometer allows repeatable indoor replication of an arbitrary drive cycle, including the Environmental Protection Agency's (EPA) Urban Dynamometer Driving Schedule and Highway Schedule[4]. The current vehicle is not capable of completing these cycles outdoors on the road. Furthermore, a dynamometer allows comparison, development, and tuning of various control strategies.

The purpose of this thesis project is to design, construct, and implement a dynamometer capable of testing the Generation 1 HHPV.

1.2 Requirements and goals

The main goal of dynamometer testing is to mimic outdoor driving in a controlled indoor environment. Fuel economy can be measured and compared to that of other vehicles on a standard drive cycle. Although possible, outdoor road testing is difficult and not repeatable. Dynamometer testing eliminates variables from outdoor testing such as wind, weather, tire pressure, and traffic. The dynamometer loads (absorbs power from) the vehicle's powertrain as though it is driving. This dynamometer must also be able to motor (provide power back into) the powertrain to replicate braking. The motoring feature is necessary to measure the vehicle's effectiveness of capturing and storing braking energy.

The EPA publishes standard drive cycles for measuring fuel economy. A drive cycle is a vehicle land speed trajectory as a function of time. The dynamometer is responsible for producing the torque appropriate for the vehicle operating at the speed and acceleration specified by the drive cycle. A plot of an example cycle, the Urban Dynamometer Driving Schedule, appears in Figure 1.3 [4]. Another popular cycle is the Highway Fuel Economy Driving Schedule. The dynamometer must be able to reproduce these dynamic cycles, arbitrary other cycles within the power capability of the vehicle, and loading at a constant speed and torque. Section 3.5 beginning on page 35 describes the loading that the dynamometer must provide in order to follow these cycles.

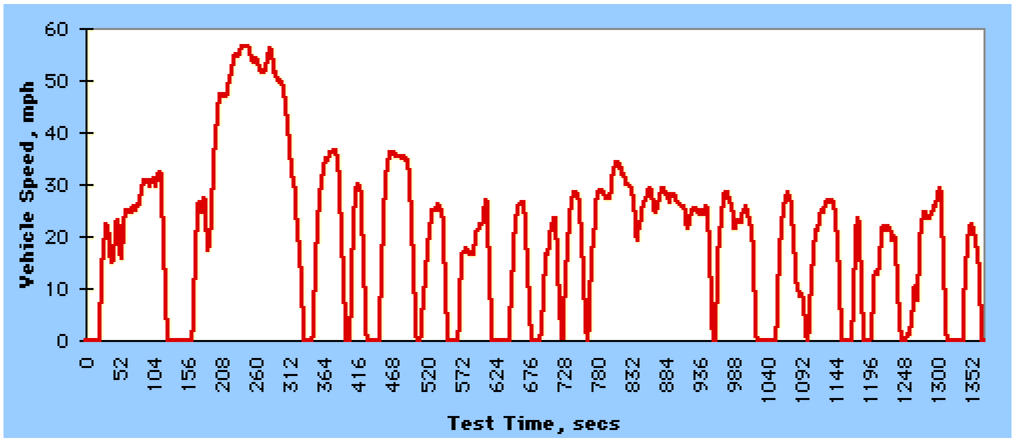

Figure 1.3: EPA's Urban Dynamometer Driving Schedule[4]

Other limitations added additional requirements for the dynamometer. The vehicle and dynamometer needed to fit in a 5 by 6m space. It is on the fourth floor of a building, and the floor and doorway could not be modified. Therefore, all components needed to fit in an elevator, through a doorway, and in the room. The room has exhaust ventilation, tap water, and three phase 20A power available. Adding additional utilities such as chilled water or a high current power supply would cost tens of thousands of dollars. The project's materials budget is $5,000. Dynamometer operators are graduate students; new students must be able to quickly learn how to use it safely.

1.3 Literature review

This section outlines the EPA fuel economy test procedure the dynamometer is intended to follow. It also summarizes some other hydrostatic and non-hydrostatic dynamometers used in similar applications.

1.3.1 EPA procedure

The EPA publishes a thorough procedure for evaluating vehicle fuel economy in the Code of Federal Regulations [5]. Vehicles complete the Urban Dynamometer Driving Schedule to receive their official fuel economy values. A human drives the vehicle through the schedule's speed trajectory on a chassis dynamometer. The driver must follow the schedule within 2 mph at any point on the speed trace within 1 second of the current time. An illustration of these tolerances appears in Figure 1.4. Speeds lower than those prescribed are acceptable, provided the vehicle is operated at maximum available power.

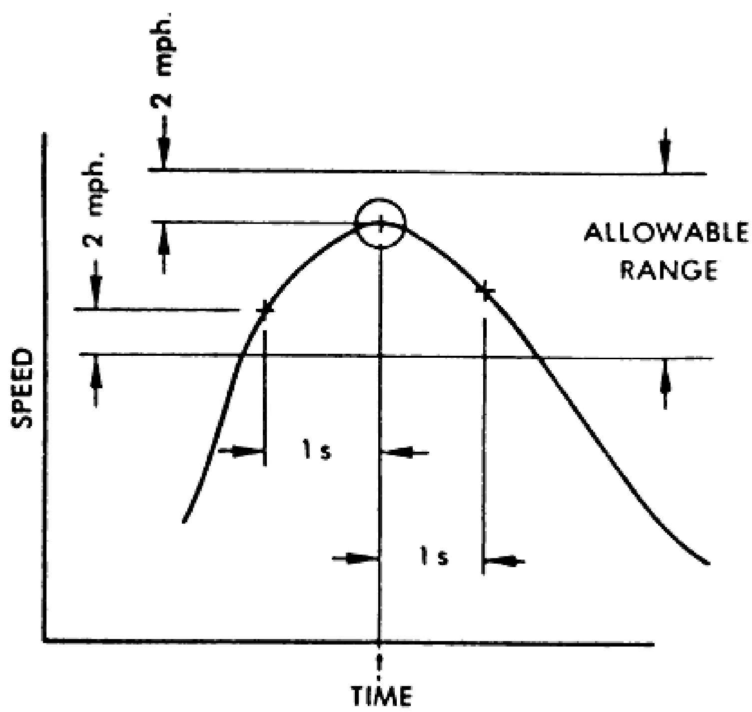

Figure 1.4: EPA drive cycle speed tolerances[5]

The vehicle's manufacturer supplies rolling and aerodynamic drag coefficients at speeds between 9mph and 72mph. The dynamometer uses this data along with other vehicle parameters to calculate the proper torque to apply to the vehicle's wheels at each instant throughout the drive cycle based on measured speed and acceleration. The parameters for the Generation 1 HHPV are listed in Table 3.4 on page 36.

Reference [5] also specifies a procedure for preparing the vehicle for the test. It includes instructions on fueling, temperature, and emissions measurement. Depending on the test being run, the vehicle may be warmed, driven, and filled with a precise volume of fuel prior to testing.

1.3.2 Construction of other hydrostatic dynamometers

The following papers describe construction of custom hydrostatic dynamometers for various purposes.

Rolewicz [6] built a system of hydrostatic dynamometers to test various components of a four-wheel drive military vehicle. It used pumps to absorb power from components and a servovalve controlled hydraulic motor to provide input power to the components that were not self-powered. This project required a modular and compact solution to allow interchanging of various components under test. It considered other types of dynamometers but settled on a hydrostatic variety because the low inertia and high stiffness improve controllability. The project reduced costs by reusing existing components. Rolewicz [6] is an example of a similar project that selected the hydrostatic option in order to reduce cost. The hydrostatic advantage of robust capabilities to pump and motor a large variety of equipment is appealing, although the current project is purpose built for a specific vehicle.

Longstreth et al. [7] build off of the work in [6]. Its goal was to realize a cost effective high-bandwidth dynamometer capable of producing transient torques. It used a fixed displacement pump coupled to a servovalve to control load. The low inertia of the pump compared to traditional dynamometers improved bandwidth and reduced vibrations. This paper shows that a hydrostatic dynamometer is a good choice for transient loading, which the current project requires.

Wang et al. [8] constructed a hydrostatic engine dynamometer to emulate the dynamics of a hybrid powertrain. The power density, low inertia, and high bandwidth are listed as reasons that "the hydrostatic dynamometer is an ideal candidate for the next-generation dynamometers." It could motor the engine when no combustion is occurring. A variable displacement pump was connected to the engine under test. A proportional valve and load sensing compensator controlled the load with fast tracking.

Holland et al. [9] built a low-cost absorbing hydrostatic dynamometer to test small and medium sized engines for educational purposes. It used a fixed displacement pump and proportional relief valve to control load; it did not offer motoring capabilities. The high power density of hydraulic systems allowed the dynamometer to be portable. It was durable and easy for inexperienced operators to learn to use.

Like this project, many of the previous papers identified low cost, low inertia, and high power density as advantages of hydrostatic dynamometers. The major disadvantage compared to electric dynamometers is that they are not an established commercial product and must be custom built.

1.3.3 Related work

Hydrostatic dynamometers are far from the only viable option for testing hybrid vehicles. Nowell used a dynamometer with inertia weights and a hydro-viscous absorber to test trucks on the EPA Federal Test Procedure duty cycle [10]. The inertial flywheels allow motoring without a driving power device. The truck drives the dynamometer either through its driveshaft or a single wheel. Although the approach is different, the goals and configuration of this is similar to those of the current project.

Wilson [11] constructed an eddy current chassis dynamometer to test a racecar and other vehicles. The vehicle drove on rollers loaded by an eddy current dynamometer. See subsection 3.3.2 for a description of eddy current technology. Wilson elected to construct his own roller apparatus to reduce cost.

1.4 Overview

Chapter 2 describes the final dynamometer: its physical construction, electronics, controls, sensors, and safety features. Chapter 3 describes the design process which used computer models to guide component selection. Chapter 0 presents the experimentally observed performance of the dynamometer. Chapter 5 concludes the thesis and presents possible future work and opportunities for the dynamometer. Appendix A contains a bill of materials for the dynamometer. Appendix B contains electrical schematics. Appendix C contains instructions for calibrating the torque sensor. Appendix D contains the MATLAB code used for design. Appendix E shows a portion of the Simulink model used for controller design. Appendix F contains operating instructions for the dynamometer.

2 Description of Final Physical Dynamometer

This chapter describes the physical dynamometer and how it works. It presents a high level overview of the dynamometer's method of operation in section 2.1. A hydraulic schematic of the dynamometer is presented, and each component is explained in section 2.2. The physical configuration of the machine is described in section 2.3. The electrical system is described along with the manual and automatic methods of controlling the dynamometer in sections 2.4, 2.5, and 2.6. Sensors and the data acquisition are described in sections 2.7 and 2.8. Safety features and ergonomics are explained in sections 2.9 and 2.10. Cost reduction techniques are described in section 2.11.

2.1 Overview of operation

To simulate driving, the dynamometer is connected to the output of the Generation 1 HHPV's transmission. A block diagram of the physical setup appears in Figure 2.1. Dynamometer testing does not use the rear differential gear or wheels; however, the dynamometer provides a load on the engine, transmission, and hydraulic system similar to that from driving on a road.

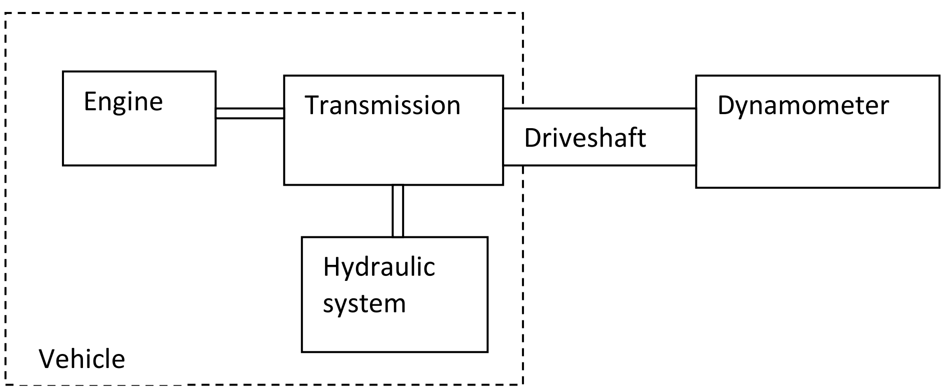

Figure 2.1: Block diagram of vehicle and dynamometer

Figure 2.2 shows the vehicle connected to the dynamometer with the safety shield removed.

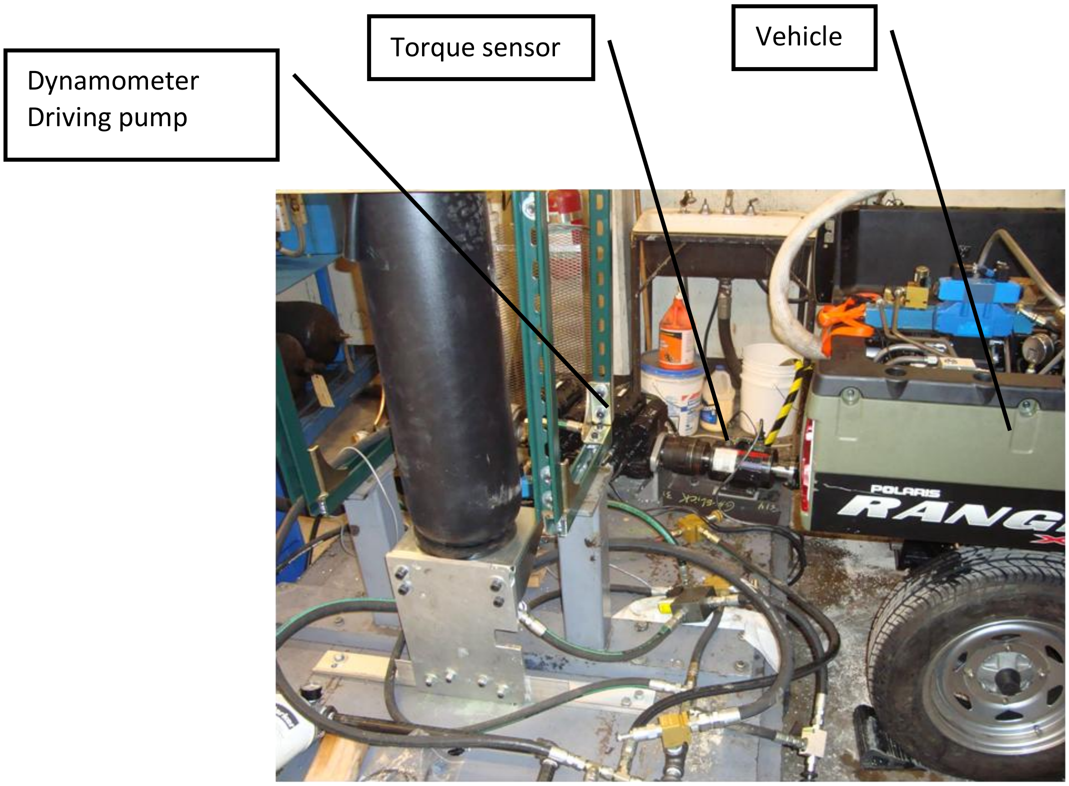

Figure 2.2: Vehicle connected to dynamometer

2.2 Hydraulic schematic and description

A hydraulic schematic of the dynamometer appears in Figure 2.3. A complete bill of materials appears in Appendix A.

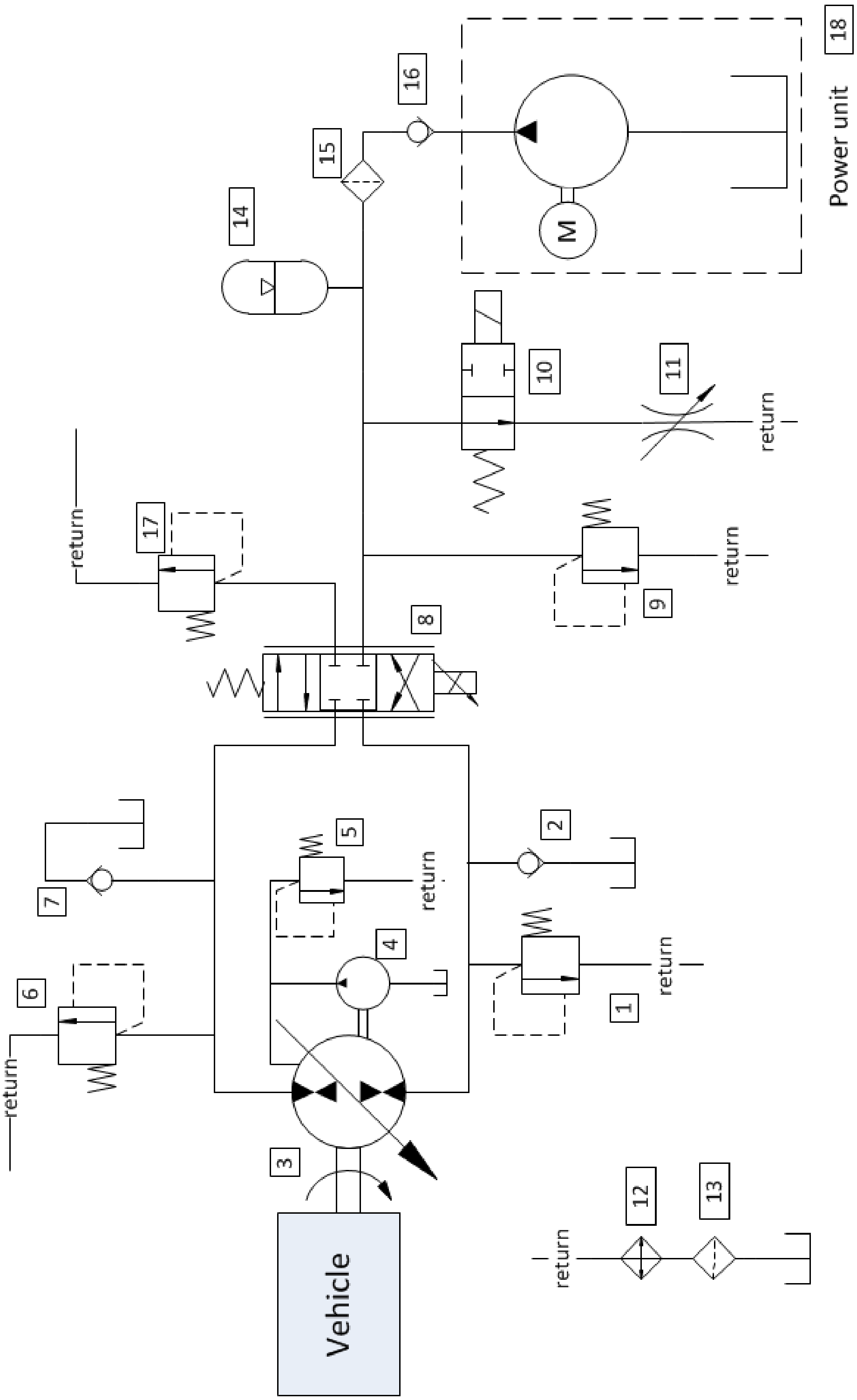

Figure 2.3: Dynamometer hydraulic schematic

Table 2.1: Hydraulic schematic key

1	High pressure relief valve
2	Anti-cavitation check valve
3	Dynamometer driving pump
4	Charge pump
5	Charge pressure relief valve
6	High pressure relief valve
7	Anti-cavitation check valve
8	Proportional directional valve
9	High pressure relief valve
10	Accumulator dump valve
11	Dump needle valve
12	Oil cooler
13	Return filter
14	Accumulator
15	High pressure filter
16	Power unit check valve
17	Low pressure relief valve
18	Hydraulic power unit

The output shaft of the vehicle's transmission directly turns tandem pump (3) which couples two 28cc/rev pumps in parallel. This pump is called the "dynamometer driving pump," although it can both pump and motor. A 16cc/rev charge pump (4) connects directly to the tandem pump and provides charge flow. Pump (3)'s displacement can be adjusted to change the load on the vehicle. It can pump or motor to simulate driving or braking, respectively. Normally, a proportional valve (8) remains in a fully open position to allow fully unobstructed fluid flow, but if the pump displacement does not change fast enough, the valve can more quickly adjust load to compensate. A continuously running hydraulic power unit (18) provides flow for motoring events. A 38 liter accumulator (14) maintains a somewhat constant high pressure and supplements the power unit's flow during motoring events.

Valves (1), (2), (6), and (7) prevent cavitation and trapped pressure when valve (8) is in the center position. A relief valve (9) is redundant to the power unit's internal relief valve but can accommodate the higher flow rate that the accumulator (14) or dynamometer driving pump (3) are capable of producing. Normally open solenoid valve (10) dumps the accumulator's stored energy if the operator presses the emergency stop button or if electrical power is lost.

Low pressure filter (13) and high pressure filter (15) remove contamination from both the pump and return lines. Filter (15) does not meet the cleanliness requirements of the proportional valve, but filter (13) does. The proportional valve requires a 3 micron or finer filter. The heat exchanger (12) cools the hot oil by passing its heat to tap water. The warm water drains to the sewer.

During normal loading, the dynamometer driving pump (3) draws fluid through a check valve (2 or 7, depending on flow direction) from the tank via suction. The fluid is pressurized by the dynamometer driving pump and pumped through a relief valve (1 or 6). The fluid passes through the low pressure filter (13) and oil cooler (12) before returning to the tank.

2.2.1 Description of hydraulic components

The following subsections describe each of the components in the above hydraulic schematic in more detail.

2.2.1.1 Dynamometer driving pump (3)

The dynamometer driving pump is directly connected to the driveshaft exiting the vehicle's transmission. It can absorb power or motor. It consists of two 28cc/rev Sauer-Danfoss Series 42 axial piston pumps coupled together in a tandem configuration. The shaft only spins in one direction. The pump's displacement can be varied infinitely from full to negative full, allowing bidirectional flow. The pump's displacement is proportional to the electrical command current between 14 and 85mA. A negative command current corresponds to negative displacement.

2.2.1.2 Hydraulic power unit (18)

The power unit runs continuously to fill the accumulator and provide flow for motoring. It is a model J-8985 assembled by Air Hydraulic Systems and can provide about 19LPM at 200bar. It has been modified to add a suction line and the capacity for more flow to prevent excessive pressure drops. A directional valve was removed, and the pressure line was connected directly to the dynamometer. All return oil passes through a single separate filter and oil cooler. The power unit's original return line filter was removed and replaced with one capable of meeting a higher cleanliness rating and flow rate. The new filter is described in subsection 2.2.1.10. The power unit did not have an available suction line, so one was installed for the charge pump (4) and check valves (2 and 7) using a pipe inserted through a new hole in the top of the reservoir. After passing through a Mueller Steam Specialty 20 micron Y type strainer (not shown in the schematic), the suction flow passes through a suction hose. A filter is not installed in the suction line to reduce the risk of cavitation. The hydraulic fluid is Mobil DTE 25.

2.2.1.3 Accumulator (14)

A Parker BA 38 liter bladder accumulator maintains a somewhat constant high pressure. If the dynamometer driving pump (3) requires more power than the power unit can provide while motoring, the accumulator supplements its flow. The accumulator fills when the dynamometer driving pump is pumping.

2.2.1.4 Proportional valve (8)

The proportional valve can vary the load faster than by changing the dynamometer driving pump's displacement. The manufacturer advertises a 50ms response time [12, p. 4]. The valve is an Eaton KFDG5V-7 two stage proportional directional valve rated for flows up to 160LPM. Its amplifier is in the dynamometer's power box. The amplifier receives a signal from the controller and feedback from the valve's spool position sensor. It sends current to the valve's solenoid. A reducing valve (not shown in schematic) is sandwiched between the main and pilot stage to provide pilot pressure by reducing the main system pressure to about 30bar. Pilot pressure from this valve is separate from the pump's charge pressure.

2.2.1.5 Charge pump(4) and relief valve (5)

The charge pump connects to the tandem dynamometer driving pump. It is a Hydreco HMP3 162025A2 16cc/rev gear pump. The relief valve is set to maintain charge pressure at about 14bar, as required by the manufacturer of the dynamometer driving pump [13, p. 9]. A reducing valve had been used to supplement charge flow during startup, but the reducing valve is excluded from the final design to prevent the dynamometer from motoring in reverse. See section 5.3 for more information.

2.2.1.6 Relief valve (17)

An Eaton RV5-10 cartridge relief valve maintains the low pressure at about 14bar. Without this minimal backpressure, the dynamometer driving pump's charge flow consumption increases substantially, and the charge pump is unable to keep up. Only return oil from the dynamometer driving pump passes through this valve.

2.2.1.7 Relief valves (1 and 6) and check valves (2 and 7)

When proportional valve (8) is not throttling flow (the spool is at an end position), these check and relief valves remain closed. When the proportional valve is throttling and the dynamometer driving pump is pumping, one check valve opens to prevent cavitation. The relief valve may also open on the opposite side if the pressure reaches the maximum system pressure. For example, if directional valve (8) is closed and the dynamometer driving pump (3) is pumping flow in the counterclockwise direction, check valve (7) and relief valve (1) open. One check valve is a Parker

C-2020-S in-line check valve; the other is an Eaton CV1-16 cartridge check valve. Both relief valves are Eaton RV5-10 cartridge relief valves. Relief pressures are set to the maximum system pressure of 200bar.

Check valve (16)

A Sun Hydraulics CXDA-XAN cartridge check valve prevents flow from the accumulator (14) from spinning the power unit backwards after it is shut off and prevents reverse flow through the high pressure filter (15).

2.2.1.8 Relief valve (9)

This Eaton RV5-10 cartridge relief valve relieves flow from the dynamometer driving pump (3) and power unit when the accumulator is full. Although the power unit also has an internal relief valve, it is not designed for the high flows that the dynamometer driving pump can produce. The new relief valve is capable of relieving 114LPM at 200bar.

2.2.1.9 Dump valve (10) and needle valve (11)

Dump valve (10) is a normally open Eaton SV13-16-O solenoid cartridge valve. Needle valve (11) is a Sun Hydraulics NFCC needle valve. The dump valve is energized and closed when the dynamometer is operating normally. When the operator presses the emergency stop button or if power is lost, the dump valve opens, releasing the accumulator's stored energy. The needle valve prevents the flow exiting the accumulator from being too fast; it must be adjusted upon installation and periodically to provide the desired flow rate to empty the accumulator in about 10s. A fixed orifice will replace the needle valve in the future so that it cannot be closed.

2.2.1.10 Filters 13, 15

The proportional valve (8) is the most sensitive component in the system to contamination and requires an ISO cleanliness level of 18/16/13 [12, p. 16]. The valve's manufacturer recommends that a filter be installed upstream of proportional valves [14]. The high pressure Eaton HF2P filter (15) does not meet the ISO cleanliness level for the valve, so a Parker 50AT return line filter (13) is also used. The return line filter has a pressure gauge to measure backpressure. The filter element should be replaced with a Parker 926541 canister when backpressure exceeds 1.7bar or after 250 hours of service.

2.2.1.11 Oil cooler (12)

Energy from the vehicle is converted to heat by throttling fluid. All return oil passes through this oil cooler and filter (13). The Thermasys EKS shell and tube heat exchanger removes up to 15kW of heat from the oil to a continuously running cold water supply. The warm water is dumped to the sewer. A solenoid water valve shuts off the water flow automatically when the dynamometer is shut down, but it is good practice to manually close the gate valve on the water supply as well to reduce leaks.

2.3 Physical design

Figure 2.4 shows a photograph of the physical dynamometer.

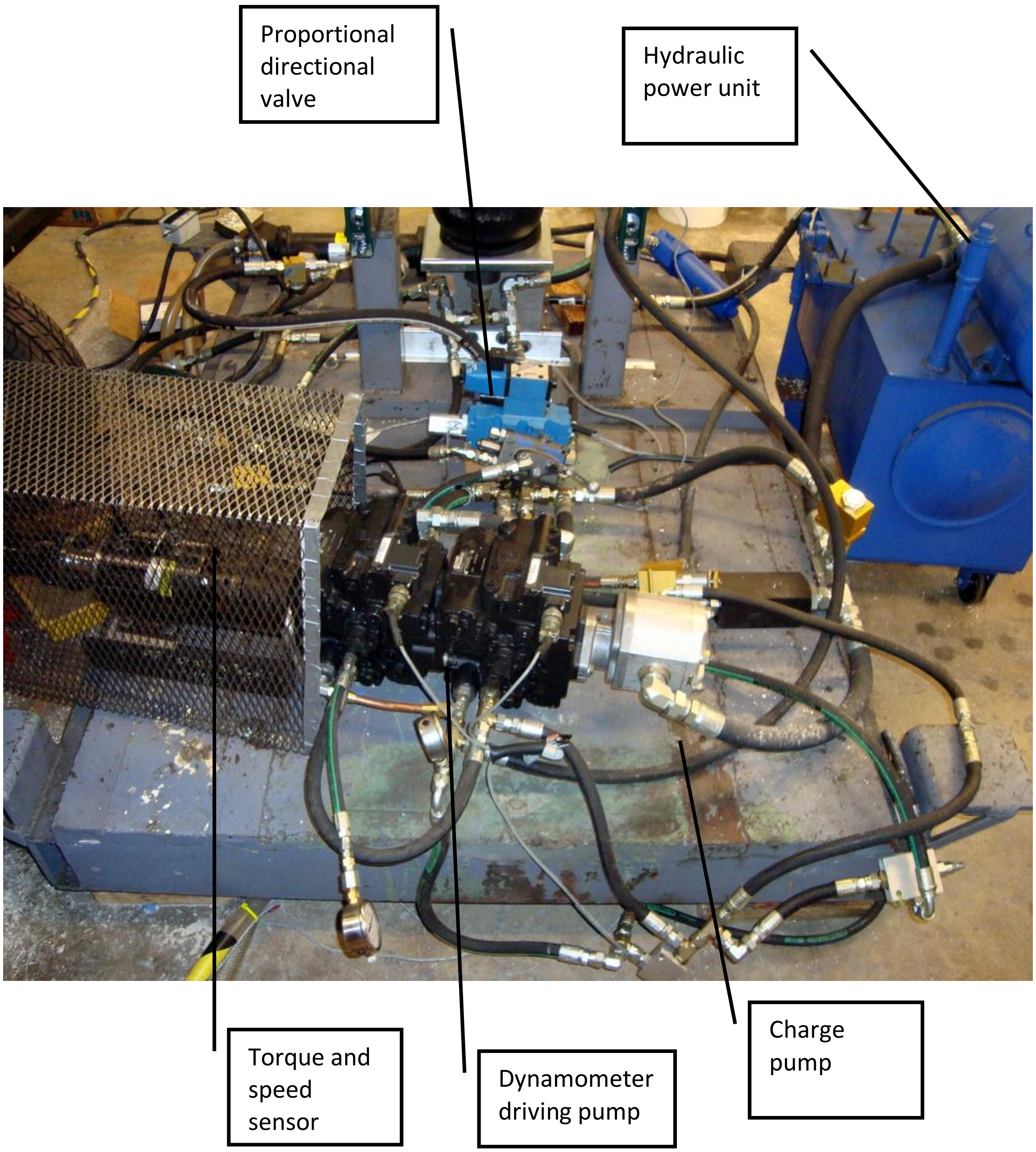

Figure 2.4: Overall physical dynamometer

Most components rest on a concrete bedplate. The dynamometer driving pump is supported by a commercially available foot bracket, which is connected to the bedplate with concrete anchors. The bladder accumulator must be held vertically according to the manufacturer's instructions. The top of the accumulator is supported by a strut and plywood structure; the bottom sits in a commercially available base. Valves and other lightweight components simply rest on the bedplate.

Most of the fluid connectors are hose with JIC fittings, but the low pressure and suction lines use some pipe. Although rigid steel tubing offers many advantages, it is impractical for this project. Since this is a prototype project, hose offers the flexibility to move or change components without replacing a tube. Plastic tubing connects the oil cooler to the city cold water and sewer at a nearby sink.

A hydraulic power unit on wheels is parked next to the bedplate. The rear of the vehicle is positioned facing the dynamometer driving pump near the edge of the bedplate.

A rubber isolated tube style driveshaft supplied by Machine Service, Inc. connects the vehicle to a torque and speed sensor (see section 2.7), which measures the torque and speed at the vehicle's output shaft. The second shaft on the torque sensor connects to the dynamometer driving pump.

2.4 Power and electronics

A high level block diagram of the electrical schematic appears in Figure 2.5. Full electrical schematics appear in Appendix B.

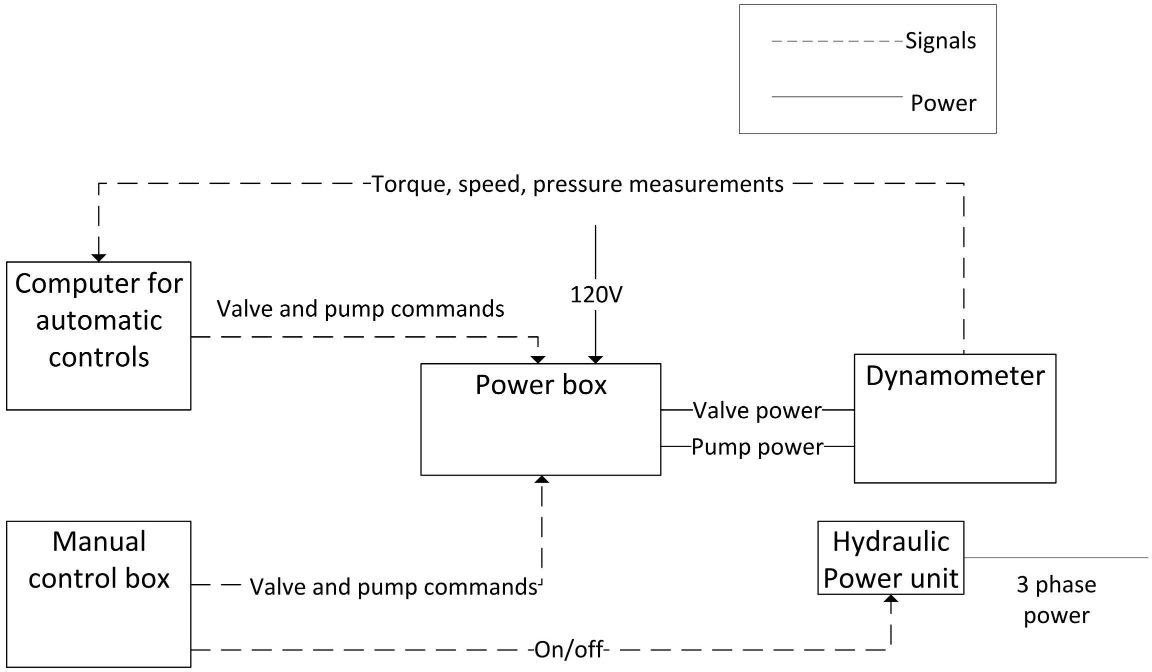

Figure 2.5: Block diagram of electrical system

The dynamometer's electronics control the dynamometer driving pump's displacement, proportional valve spool position, cooling water valve, dump valve, and the vehicle kill switch. The electrical systems consist of a power box and a control box. A computer may be added to control the proportional valve spool and dynamometer driving pump displacement automatically.

The power box houses 12V and 24V power supplies along with the proportional valve's amplifier, relays, and circuitry, to provide displacement control current to the dynamometer driving pump. It receives signals from the control box and/or computer and supplies the appropriate electrical power to all components on the dynamometer. The power box rests on the floor and is wired to the components on the dynamometer, the manual control box, and optionally the computer.

2.5 Manual controls

Figure 2.6: Manual control box

A handheld control box (Figure 2.6) allows the operator to adjust dynamometer load using knobs. This feature is useful for troubleshooting. The control box is separate from the power box, allowing it to be portable. It is wired to the stationary power box and the hydraulic power unit. One knob adjusts the dynamometer driving pump's displacement between 0 and full. An adjacent switch changes the sign of the pump's displacement to allow over center operation. Two knobs are necessary to adjust the proportional valve's spool position: one for each side of center. Operating the valve by turning more than one of its knobs away from the zero position is not recommended because their signals will cancel each other. An emergency stop button shuts down the system and puts it in a safe mode. Buttons allow the user to start and stop the hydraulic power unit remotely. The original start/stop buttons remain in place on the power unit.

2.6 Automatic controls

Two personal computers running xPC Target from MathWorks control the dynamometer automatically. The user creates a block diagram controller on the host computer running Windows XP and downloads it on to the target computer, which has no operating system. The target computer contains the data acquisition and voltage output cards. It controls the dynamometer in real time. The user can change some basic controller parameters while the real time program is running from the host computer.

The purpose of the automatic controller is to control the dynamometer driving pump's displacement and the proportional valve's spool position to provide the appropriate torque based on the vehicle's speed. Inputs to the controller are driveshaft speed and dynamometer driving pump torque and pressure. Outputs are dynamometer driving pump displacement and proportional valve position. The controller is currently a proportional integrator (PI) with feedforward. The feedforward portion calculates the dynamometer driving pump displacement that theoretically produces the desired torque. The PI portion of the controller compares actual to desired torque to correct the displacement command from the feedforward portion. This controller currently leaves the proportional directional valve in its fully open position. However, control of the proportional valve could be added later if the dynamic response needs to be improved.

The manual control box has two electrical connectors. The first is for the emergency stop and hydraulic power unit signals. The second is for the proportional directional valve and dynamometer driving pump displacement signals. The cable for the first always remains connected. The cable for the second can be removed from the manual control box and connected through the computer for automatic controls.

2.7 Sensors

Sensors on the dynamometer measure pressure at both ports of the dynamometer driving pump, driveshaft torque, and driveshaft speed. Omega PX309 pressure transducers are screwed directly into the gauge ports of the dynamometer driving pump and are able to measure 210bar. They output a voltage proportional to pressure which is read by the target PC's data acquisition (DAQ) system. A Honeywell Lebow 1105 torque sensor placed between the dynamometer driving pump and the driveshaft from the vehicle measures torque at this point. It is a strain gauge based torque sensor with a slip ring. The electronics that modify the torque sensor's signal for use by the DAQ system must be calibrated regularly. Calibration instructions appear in Appendix C. The torque sensor unit also contains a speed sensor. It outputs 60 pulses per revolution, so the frequency of pulses increases with speed.

Most of the sensing hardware was reused from a pump test stand developed to quantify the performance of pump motors in the HHPV. A detailed explanation of the electronics necessary to allow the sensors to communicate with the computer appears in Chapter 2 of reference[15].

2.8 DAQ system

Three Measurement Computing DAQ cards are installed in the target computer. A PCI-QUAD04 quadrature in card reads pulses from the speed sensor. A PCI-DAS1602/12 analog in card reads analog signals from the pressure transducers and torque sensor. A PCI-DAC6702 analog out card sends analog voltage commands to control the pump displacement and proportional valve. A block for each of the DAQ cards is inside of the controller's Simulink block diagram. It allows access to the DAQ cards' signals.

2.9 Safety features

Operation of the dynamometer presents many safety hazards including high pressure, high temperature, external oil leaks, exhaust, noise, diesel fuel, potential flooding, stored energy, tripping hazards, ergonomics, electrical hazards, and rotating machinery.

The dynamometer may be operated from a separate room that adjoins the test cell, mitigating many of the above safety issues. The operator carries the control box into the adjacent room and views the apparatus through a window. The computer for automatic control is on an ergonomic cart and can be adjusted for either a standing or seated operator.

When the operator pushes the emergency stop button on the control box, the dynamometer and vehicle lose power and go into a safe mode. The vehicle turns off, and its main power is cut. The dynamometer also loses power, so the dynamometer driving pump's displacement becomes zero. The dump valve (item 10 in Figure 2.3) opens and empties the accumulator's stored energy. The operator does need to push the power unit's off button to shut this off separately.

The dynamometer is designed so that it cannot turn the vehicle's driveshaft backwards, which could damage the engine. If the driveshaft began to turn backwards, the charge pump would stop providing charge pressure. Thus, the dynamometer driving pump displacement would go to zero, and the dynamometer would stop producing torque.

Exhaust exits the room through a sealed pipe. A large blower replaces the room's entire volume of air once per minute, removing the small amount of exhaust gas that may leak. A fiberglass sleeve encircles the exhaust tube to prevent anyone who touches it from burning themselves.

All rotating components, including the driveshaft, are shielded with expanded metal guards.

Electrical wires and tubes that run across a walk path are taped to the floor and marked with hazard tape. All electrical connectors are mistake proof. No two connectors are the same type, so it is impossible to improperly connect wires.

All high pressure fluid connectors are rated for at least 200bar. Flexible hose is used rather than rigid tubing to connect to pumps to minimize damage and leakage due to vibration. All fluid connectors are tested to be free of leaks, as leaks would create a slipping hazard.

2.10 Operator's station

The operator's station (Figure 2.7) contains the computers for controlling and monitoring the apparatus. It is ergonomic and portable so it can be rolled a safe distance away from the running equipment. The equipment rests on plywood shelves; the keyboard and mouse rest on an adjustable height tray. An enclosure contains the screw terminals which connect the DAQ system to the dynamometer. Another enclosure, reused from reference [15], contains connectors and circuitry to interface with the sensors.

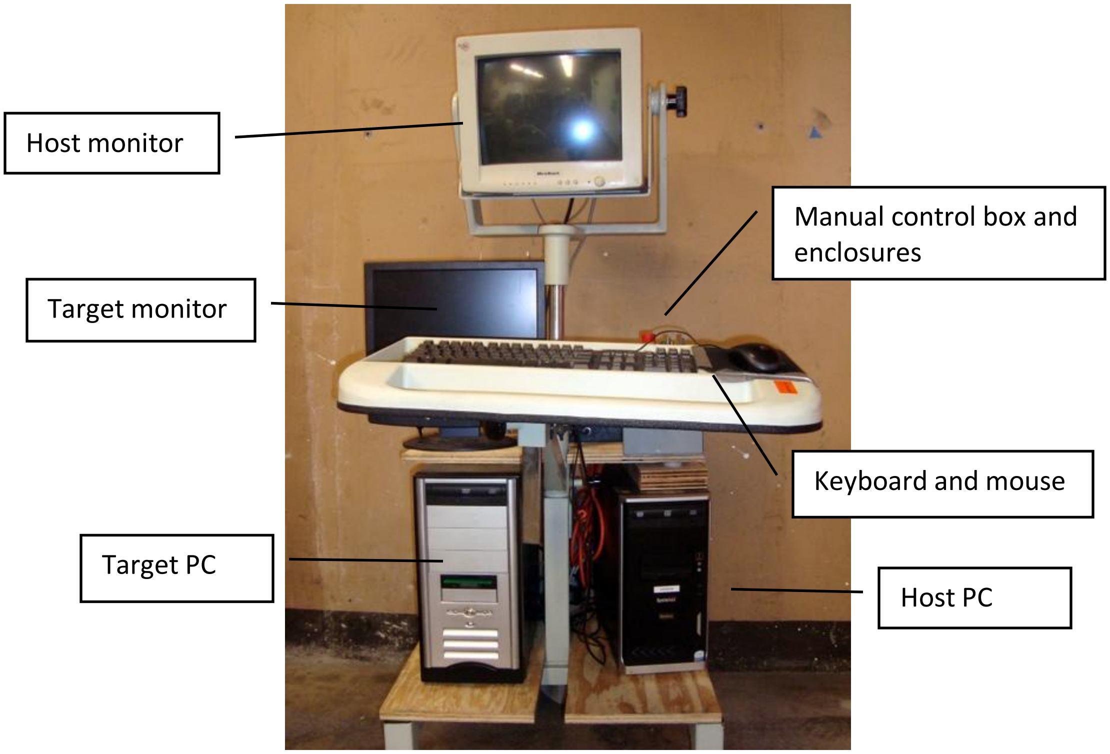

Figure 2.7: Operator's station

2.11 Cost

The cost to the project of new parts was $4200. Modifying an electrical receptacle in the room to power the hydraulic power unit cost an additional $800. A full bill of materials appears in Appendix A. Although many parts were reused for the hydrostatic dynamometer, its total value is still competitive with an AC dynamometer system. An AC dynamometer system that meets the needs of this project would have cost at least $40,000.

Reusing existing components saved the project at least $20,000. The laboratory already owned and reused the dynamometer driving pump, torque sensor, bedplate, proportional valve, cartridge valves, hydraulic power unit, high pressure filter, and some hose. Parker Hannifin Corp., Daman Products Co., Inc., and The University of Minnesota ReUse Program donated components.

Pipe fittings offer a cost savings over JIC fittings but sacrifice leak tightness and durability. Therefore, pipe was used as a low cost fluid connector on low pressure return and suction lines where the risk of leaks was not great. The maximum system pressure is 200bar (3000psi). Although a 340bar (5000psi) system would have increased power density, 200bar components cost less and are more available. Designing the system for the lower pressure saved weeks of time and thousands of dollars.

3 Design

This chapter describes the design process followed to converge on the final product. Critical requirements are summarized in section 3.1. A product design specification appears in section 3.2. Using contract testing services or purchased dynamometers was researched in section 3.3. Eventually, a purpose built hydrostatic dynamometer was chosen using a product selection chart in section 3.4. General component specifications were developed using MATLAB simulations. A MATLAB simulation of the vehicle is described in section 3.5. Two different simulations of the dynamometer were created for distinct purposes. Section 3.6 describes a MATLAB simulation of the dynamometer made for the purpose of component sizing. A dynamic model of the dynamometer was created in Simulink in section 3.7 for the purpose of controller design. Data from these simulations is used to select or justify reusing components in section 3.8.

3.1 Summary of customer requirements

The purpose of the dynamometer is to test a hydraulic hybrid vehicle built on a Polaris Ranger utility vehicle chassis. In order to replicate driving on the road and be feasible for this project, the dynamometer must:

- Be safe to operate.
- Cost about $5000 in new components.
- Follow EPA drive cycles, including portions requiring motoring.
- Be in a convenient location to allow frequent testing.

3.2 Product design specification

Table 3.1 summarizes the design specifications for the dynamometer. The importance of the design criteria is ranked on a scale from 1 to 10.

Table 3.1: Product design specification

Design criterion	Importance	Units	Target value
Cost	9	$	<5000
Safety	10	Safe or unsafe	Safe
Able to motor and absorb together	8	Binary	Yes
Electrical requirements	3	kW	<7
Physical size	3	m^2	<10
Time to assemble	3	Hour	<150
Time to design a controller	2	Hour	<30
Inertia	5	$kg{\cdot}m^2$	<0.01
Number of people familiar with system	1	#	5
Maximum continuous absorbing power	5	kW	>10
Maximum continuous motoring power	5	kW	>5
Maximum peak absorbing power	5	kW	>25
Maximum peak motoring power	5	kW	>20
Maximum simulated ground speed	5	km/h	>100
Maximum simulated wheel torque (motor or absorb)	5	Nm	>500

3.3 Consideration of offsite and purchased dynamometers

Using a dynamometer operated by an outside firm or purchasing a commercially available one ourselves was an appealing option because it would have reduced risk and design time. Ultimately, however, these alternatives did not meet the needs of the project because they were too geographically distant, too costly, or unable to complete the motoring portions of the EPA drive cycles. The various options are described in the following subsections.

3.3.1 Offsite dynamometer

Using an offsite dynamometer was initially considered. This dynamometer would be a chassis dynamometer with the vehicle's wheels spinning on rollers. The vehicle would need to be transported to an offsite location and connected to the dynamometer with minimal modification.

Although the offsite dynamometer would have been the fastest to implement and lowest risk, it was rejected because no dynamometers in close geographic proximity to the vehicle's location in Minneapolis, MN met the needs of the project. Local dynamometers only offered the ability to absorb power, not motor. The nearest available motoring chassis dynamometer was a 90 minute drive away in Mankato, MN and may not have been operational in time for the vehicle's testing. The next available option was in Ann Arbor, MI, a 12 hour drive away. Both of these options were too geographically distant to allow frequent testing and development work.

3.3.2 Purchased dynamometer

An AC or DC dynamometer contains a large electric motor/generator and is capable of both motoring and absorbing power. Initial costs are high; the dynamometer alone costs $40000. Furthermore, the room available for the vehicle testing would have required substantial work to accommodate the electrical demands of the machine.

An eddy current dynamometer uses eddy currents in a large rotating metal disc to generate a load. It would need to be combined with an electric motor to provide motoring power. This project had access to an eddy current dynamometer but not the electric motor which would have also required electrical modifications to the room.

3.4 Selection of general design

After consideration of all options, the three most promising choices were carefully compared. The choices of a custom build hydrostatic dynamometer, an AC dynamometer, and a borrowed eddy current dynamometer combined with an electric motor are compared on several metrics in Table 3.2. The degree to which each of the three choices meets the needs of the project is quantified in Table 3.3, and the hydrostatic dynamometer is deemed the best choice.

Table 3.2: Comparison of general dynamometer designs

	Hydrostatic	AC dynamometer	Borrowed eddy current + electric motor
Cost	$5000.	$40000 plus electrical work.	$12000 plus electrical work.
Safety	Lower than others. Unproven technology. Leaks. Stored energy in accumulator. Possible to drive transmission in wrong direction.	Medium. Some electrical and heat hazards. Possible to drive transmission in wrong direction.	Medium. Some electrical and heat hazards. Unlikely to drive transmission in wrong direction.
Able to motor and absorb together	Yes.	Yes.	No. Requires changing components to change between motoring and absorbing.
Probability of success	Successful in some papers but not commercially available.	Proven.	Eddy current is proven. Motoring portion is unusual.
Electrical requirements	Minimal. Three phase only.	Most. Requires high current plus sending power back to grid.	Three phase. Possibly higher current required than hydrostatic.
Physical size	Smallest.	Medium. May require large drive cabinet.	Largest. Motor and dynamometer are separate.
Time to assemble	Hardest. Entirely custom.	Easiest. Comes semi-assembled.	Medium.
Time to design a controller	Hardest. Multiple variables to control.	Easiest. Manufacturer provides a torque controller.	Easy. Motor drive and dynamometer have their own controllers.
Inertia	Best.	Medium.	Medium.
Number of people familiar with system	Best. Many hydraulics experts available.	Worst.	Medium. These are common in the engine lab but not in our group.
Max. simulated ground speed	Same as others	Same as others	Same as others
Max. simulated wheel torque	Same as others	Same as others	Same as others

Table 3.3 quantifies the information in Table 3.2. Each design criterion is assigned a weight ranging from 1 to 10. The degree to which each of the three design possibilities satisfies the design criteria is rated on a scale ranging from 0 to 5. The ratings are multiplied by the weights and summed for each of the three designs. The best design is the one with the most total points. Maximum torque, speed, and power specifications are excluded because all options are equally capable of meeting these specifications.

Table 3.3: Evaluation of design options

Design criterion	Weight	Hydro-static	AC	Borrowed eddy current + electric motor
Cost	9	5	0	2
Safety	10	1	2	3
Able to motor and absorb together	8	5	5	1
Electrical requirements	3	4	0	2
Physical size	3	5	3	2
Time to assemble	3	1	5	3
Time to design a controller	2	1	5	4
Inertia	5	5	2	2
Number of people familiar with system	1	5	2	3
Total		**157**	**106**	**98**

The hydrostatic dynamometer received the most points and was selected for this project.

3.5 MATLAB simulation of vehicle

A simple MATLAB simulation was created to calculate loads experienced by the vehicle on an arbitrary drive cycle. The output of the simulation is the wheel torque and wheel speed. The code of this simulation appears in Appendix D.

EPA drive cycles provide speed data at a frequency of 1Hz. The simulation differentiates the drive cycle's speed versus time information using a backward difference numerical derivative to calculate the vehicle's acceleration. This method does not generate substantial noise for the EPA drive cycles because the speed data is already smooth. The simulation calculates the force due to acceleration by multiplying acceleration by the vehicle's mass. It calculates the rolling resistance based on the vehicle's speed, weight, and rolling resistance coefficients. It calculates drag force based on the vehicle's speed, frontal area, drag coefficient, and air density. It calculates vehicle power by multiplying the speed by the total resistive force. It calculates the rotational speed of the wheels by dividing the road speed by the tire radius.

The vehicle parameters used in this simulation are summarized in Table 3.4. These parameters are estimated [3]. They can easily be adjusted for fuel economy evaluation if they are more precisely known for the vehicle being tested.

Table 3.4: Vehicle parameters

Mass, m	1000kg
Tire diameter, d	0.619m
Rolling resistance coefficient f_o	0.0095
Rolling resistance coefficient f_s	0.0035
Frontal area, A_f	1.784m^2
Drag coefficient, c_d	0.5
Density of air, ρ	1.29kg/m^3
Acceleration due to gravity, g	9.81m/s^2

Vehicle resistive force, *F*, is given by Equation (3.1) from reference [3].

$$F = m \cdot a + 0.5 \cdot c_d \cdot A_f \cdot \rho \cdot v^2 + m \cdot g \cdot \left(f_0 + \left(3.24 \cdot f_s \cdot \left((v \cdot 2.23693629)/100\right)^{2.5}\right)\right)$$

(3.1)

where *a* is the vehicle's acceleration in m/s and *v* is the vehicle's velocity in m/s.

The first term of Equation (3.1) represents the acceleration force on the vehicle. The second term represents the aerodynamic drag force. The third term represents the tire rolling drag force.

Figure 3.1 shows the road speed verses time trace for the EPA's Urban Dynamometer Driving Schedule (UDDS) using data from reference [4].

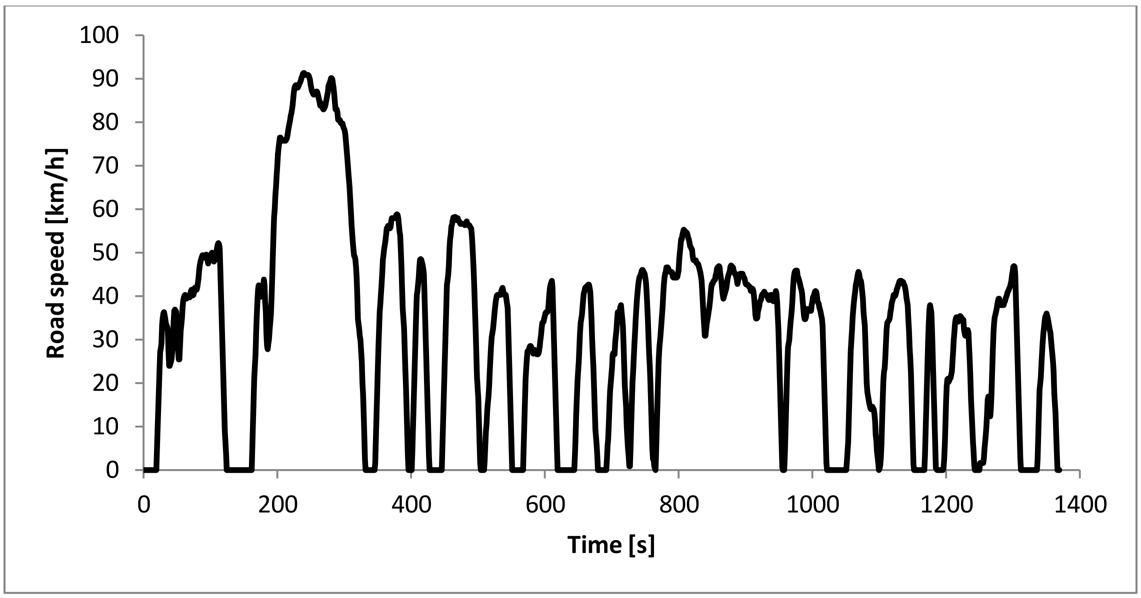

Figure 3.1: EPA UDDS speed

From this, the simulation calculates wheel speed (Figure 3.2) and wheel torque (Figure 3.3). Negative torque corresponds to braking.

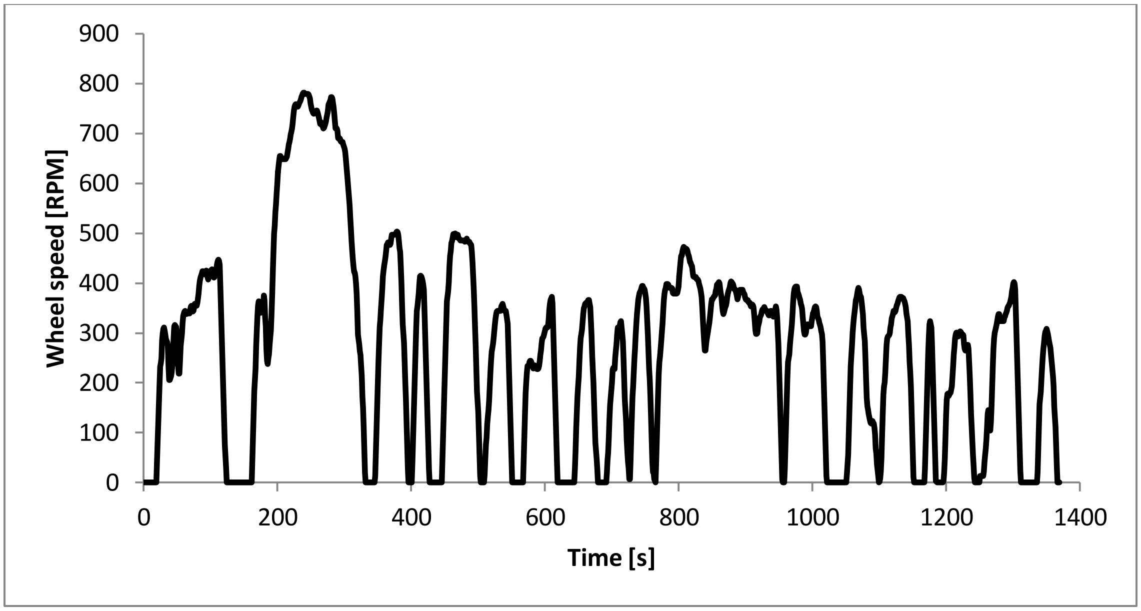

Figure 3.2: EPA UDDS wheel speed

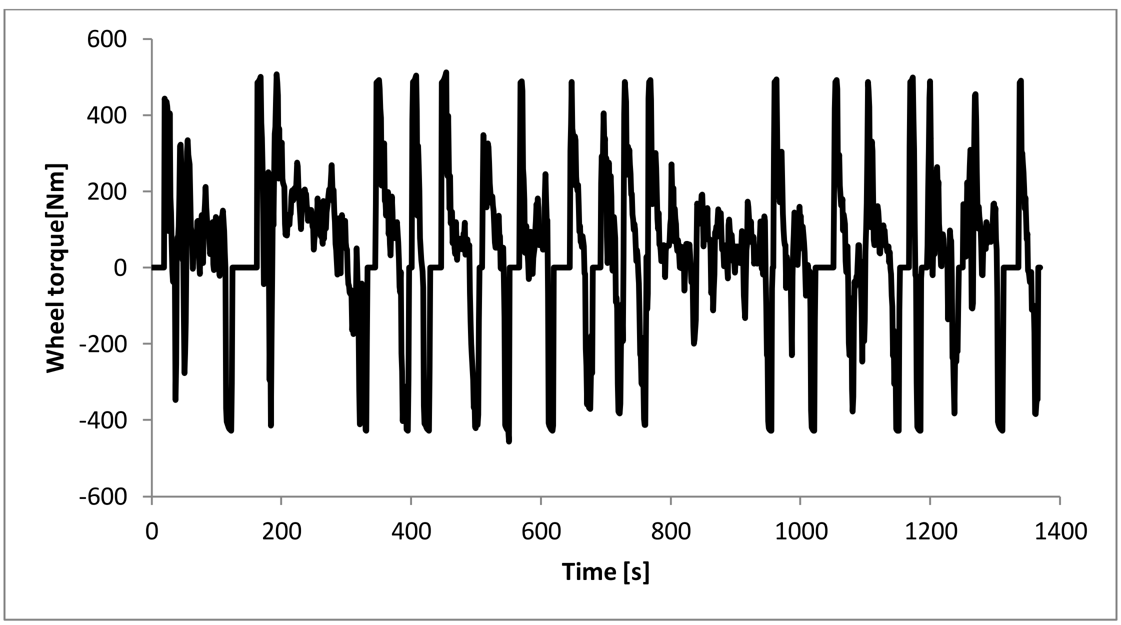

Figure 3.3: EPA UDDS wheel torque

The EPA Highway Fuel Economy Driving Schedule (HWFET) represents highway driving conditions under 60mph [4]. The graphs for the vehicle driving the HWFET are provided in Figure 3.4, Figure 3.5, and Figure 3.6.

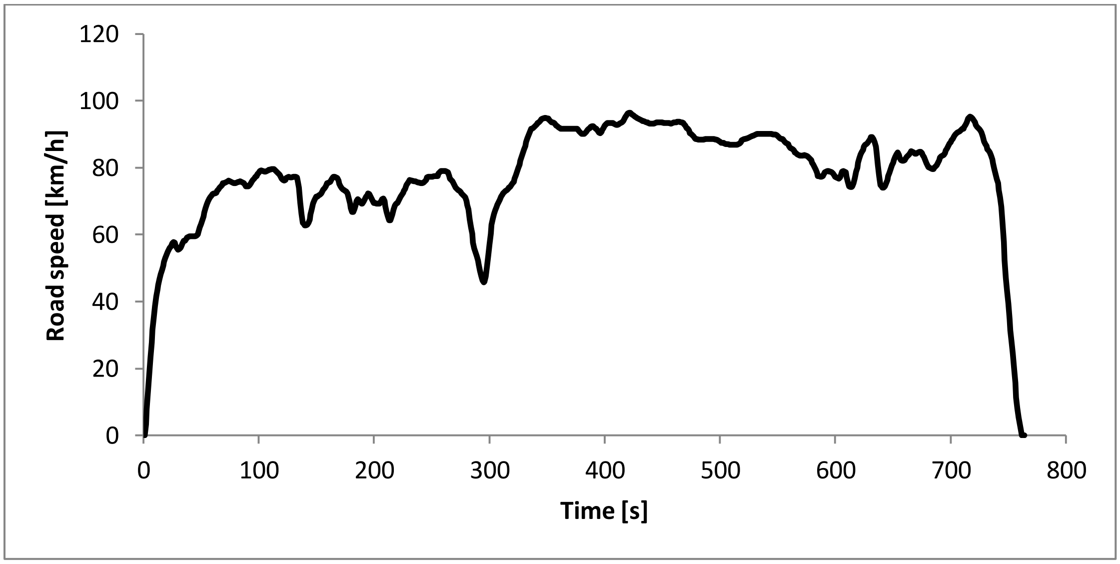

Figure 3.4: EPA HWFET speed

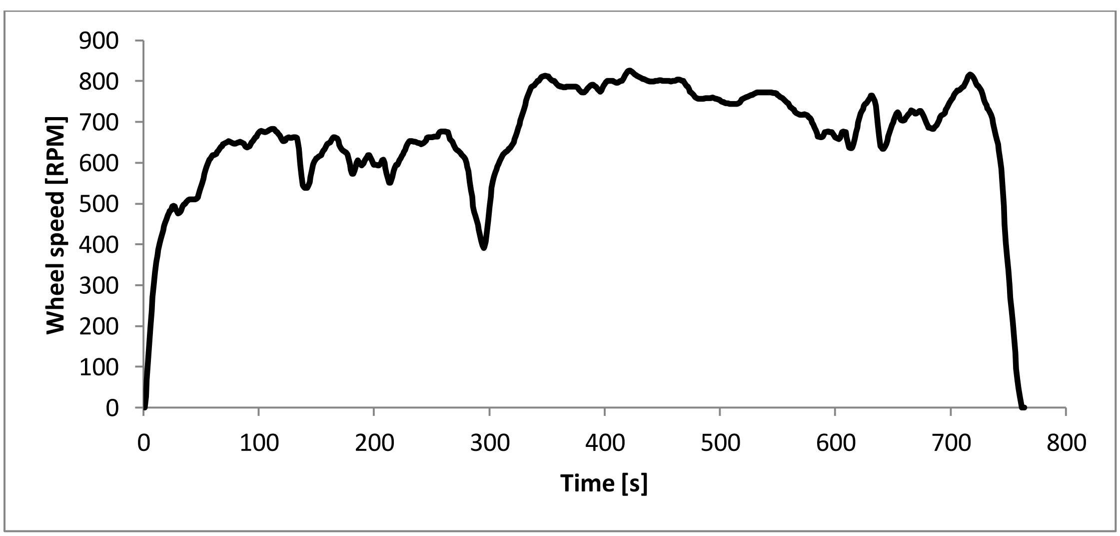

Figure 3.5: EPA HWFET wheel speed

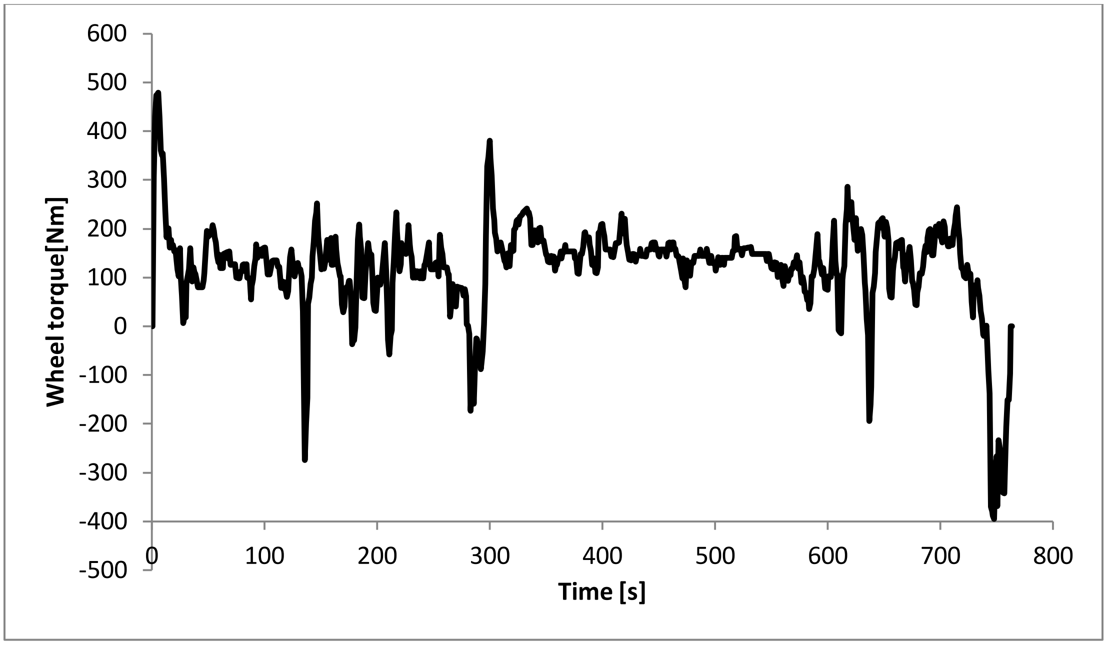

Figure 3.6: EPA HWFET wheel torque

The previous simulations show that the vehicle's typical wheel speed on EPA cycles is about 500rpm. This speed is too slow for a hydraulic pump to operate efficiently if driven directly from the wheel. The differential between the transmission and wheels reduces the transmission's output speed and increases its torque by a ratio of 3.45. Without the differential in place, the typical speed of 1500 rpm can better drive a hydraulic pump. Furthermore, bypassing the differential makes physically connecting a driveshaft easier. We therefore decided to continue design and simulation omitting the differential. Operated in this manner, the maximum speed during either the UDDS or HWFET cycles is 2900rpm. The maximum torque during the cycles is 150Nm.

3.6 MATLAB simulation of dynamometer

Two simulations were created to model the dynamometer. Each simulation serves a distinct purpose. The purpose of the simulation created in MATLAB is to validate component sizing. The purpose of the simulation created in Simulink is to aid the development of the dynamometer's controller. The MATLAB simulation is described in this section. The Simulink simulation is described in section 3.7 beginning on page 47.

The MATLAB simulation of the dynamometer following the EPA UDDS and HWFET cycles was created to size the accumulator, oil cooler, hoses, and valves. The simulation also verified that existing components could be used as the dynamometer driving pump and power unit. It is intended to provide only a rough estimate of dynamometer operating parameters; parasitic losses such as the charge pump and viscous losses are neglected. It is intended to validate the sizing of a package of components but not to optimize the selection. The code for the simulation appears in Appendix D. The inputs of the simulation are dynamometer driving pump speed, torque, and displacement. The simulation outputs are dynamometer driving pump pressure, flow rate, and accumulator fluid volume.

The simulation assumes that the dynamometer is configured in a manner similar to what is shown in Figure 2.3 on page 13. However, it assumes that the accumulator's pressure is reduced by simple throttling for use by the dynamometer driving pump and that this is achieved by some method which may not be the proportional directional valve. It assumes that the accumulator's pressure is constant.

The simulation assumes that the output of the vehicle's transmission directly turns the dynamometer driving pump and that the differential is removed. It calculates hydraulic pressures and flow rates for given vehicle speeds and torques calculated in the separate MATLAB simulation of the vehicle. The pressure at the dynamometer driving pump necessary to produce the desired torque is:

$$\boldsymbol{P = \tau \cdot 10 \cdot 2\pi / x} \qquad (3.2)$$

where P is dynamometer driving pump pressure in bar, τ is torque in Nm at the dynamometer driving pump's shaft, and *x* is the dynamometer driving pump's displacement in cc/rev. The factor of 10 is necessary because *P* is in bar and *x* is in cc/rev. The purpose of Equation (3.2) is to determine if the pressure required at the dynamometer driving pump to produce the torque required by the drive cycle is below the maximum system pressure. *x* is chosen manually and can vary with time if desired.

The proportional directional valve would be used on the physical dynamometer to regulate the pressure this way. The flow rate through the dynamometer driving pump is:

$$\mathbf{Q = x \cdot \boldsymbol{RPM}/1000} \qquad (3.3)$$

where *Q* is the flow rate in LPM and *RPM* is the dynamometer driving pump's shaft speed in RPM. *RPM* is fixed by the drive cycle. *x* is again chosen manually. Determining the flow rate through the dynamometer driving pump with Equation (3.3) is useful for a number of reasons. It is used to calculate the accumulator's state of charge, which ensures the accumulator will be large enough to complete motoring events. This flow rate is also necessary to size the valves, fluid connectors, and oil cooler.

In reality, the dynamometer driving pump's displacement may be varied to control torque, and the proportional directional valve could be left fully open. In this case, the dynamometer's controller would output a displacement command based on measured and desired torque, and monitoring pressure would be optional. The current MATLAB simulation is not intended to model the system operated in this manner since it assumes that the proportional directional valve's throttling controls the pressure to control the torque. However, it is conservative and can still validate component sizing conservatively because the dynamometer operates more efficiently when the proportional directional valve is not throttling.

The sizes of dynamometer components are entered into the simulation prior to execution. If the component sizes do not produce the desired results, the sizes can be changed and the simulation rerun.

The most commonly used results of the simulation are the volume of fluid in the accumulator and pressure at the dynamometer driving pump. The variation of pressure with accumulator state of charge is neglected. This pressure depends on the accumulator's nitrogen precharge pressure. If the precharge pressure is high enough to provide enough torque at the dynamometer driving pump when the accumulator is nearly empty, the system pressure will remain at least that high as long as oil is in the accumulator.

Figure 3.7 shows the volume of fluid in the dynamometer's accumulator during the HWFET with a constant dynamometer driving pump displacement of 50cc/rev and 10L of fluid initially in the accumulator, a reasonable amount for a 38L accumulator. Figure 3.8 shows the pressure at the dynamometer driving pump for the same test. Positive pressures correspond to when the dynamometer is absorbing power; negative pressures correspond to when the dynamometer is motoring.

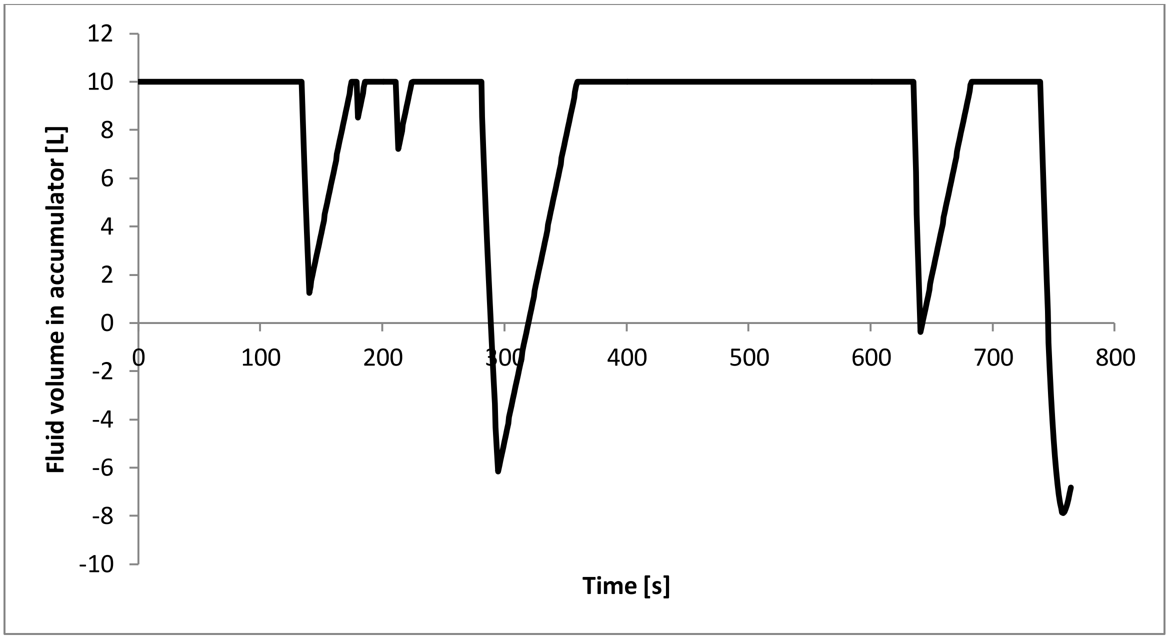

Figure 3.7: Volume of fluid in accumulator versus time drops below 0 in the vicinity of 300s and 750s corresponding to the HWFET

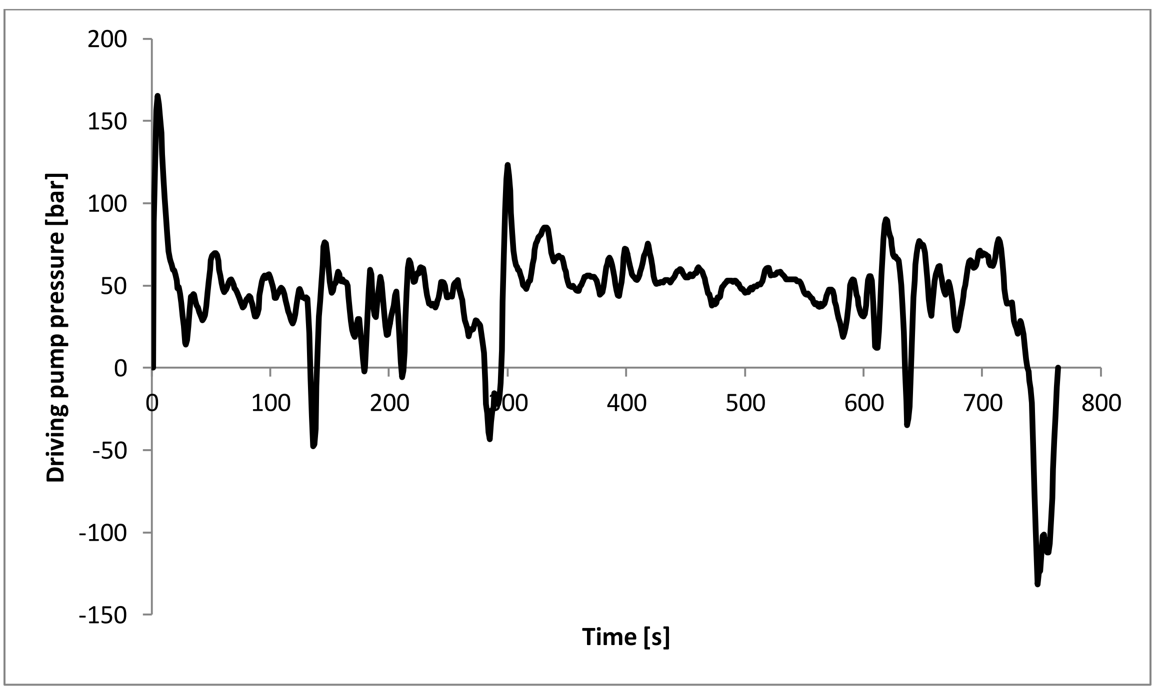

Figure 3.8: Pressure at dynamometer driving pump corresponding to the HWFET

These results indicate that the volume of fluid stored in the accumulator drops below zero at two different times during the drive cycle. This means that the liquid volume stored in the accumulator is insufficient to complete these motoring events. Increasing the accumulator's size could solve this problem. However, a less expensive option is to decrease the dynamometer driving pump's displacement during these events. To maintain the required torque, the pump's pressure increases, but its flow rate decreases. This simulation does not automatically control the dynamometer driving pump displacement, but the user may manually specify displacement as a function of time prior to running the simulation. For example, the user may choose to lower the dynamometer driving pump displacement at times when the required torque is low and speed is high in order to reduce the flow rate and conserve energy stored in the accumulator during motoring.

To verify that the dynamometer could complete the HWFET cycle with a 38L accumulator, the simulation was rerun with a manually varied dynamometer driving pump displacement. To maintain the drive cycle's required torque at a lower displacement, the dynamometer driving pump pressure must increase. Nevertheless, this pressure must always remain below 200bar. Figure 3.9 shows the dynamometer driving pump displacement during the HWFET. Figure 3.10 shows the volume of fluid in the accumulator as a function of time. Since the volume does not drop below 0, the accumulator should be able to store enough energy for all motoring events during the HWFET. Figure 3.11 shows that the pressure remains below 200bar even with the reduced displacement.

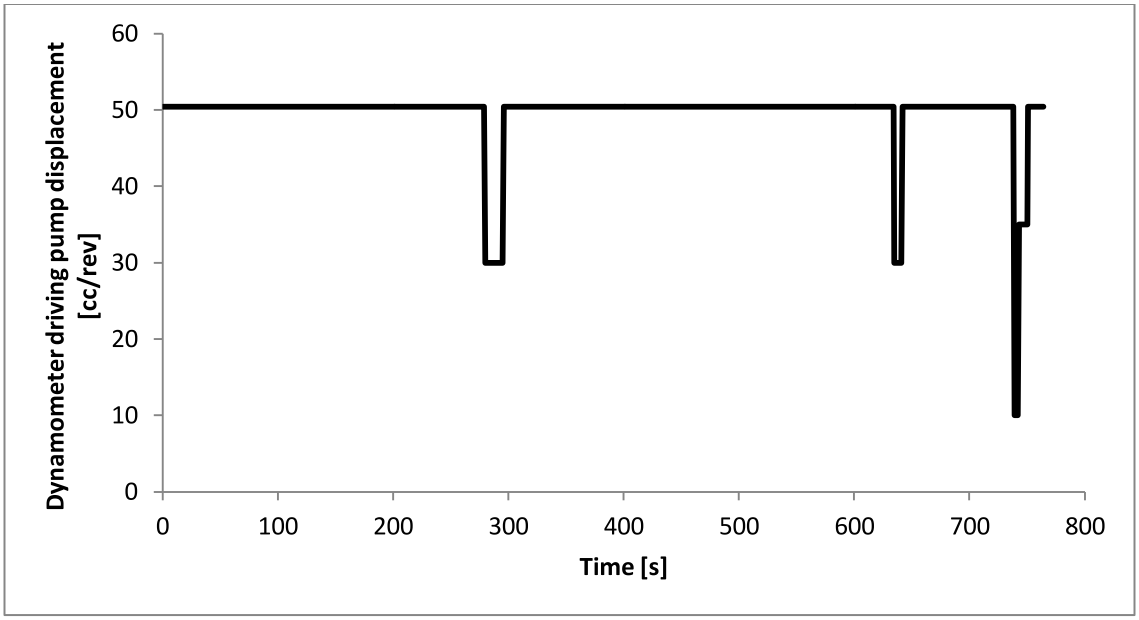

Figure 3.9: Manually varied dynamometer driving pump displacement during HWFET cycle

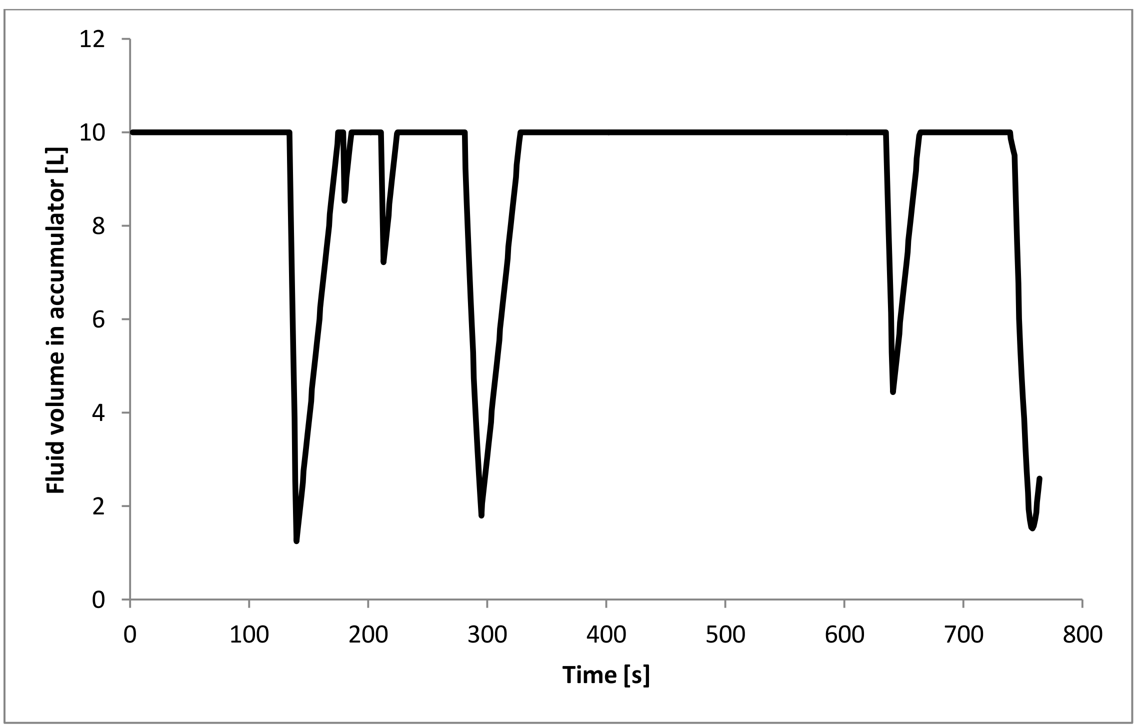

Figure 3.10: Volume of fluid in accumulator remains positive if dynamometer dynamometer driving pump displacement is varied according to Figure 3.9 during the HWFET

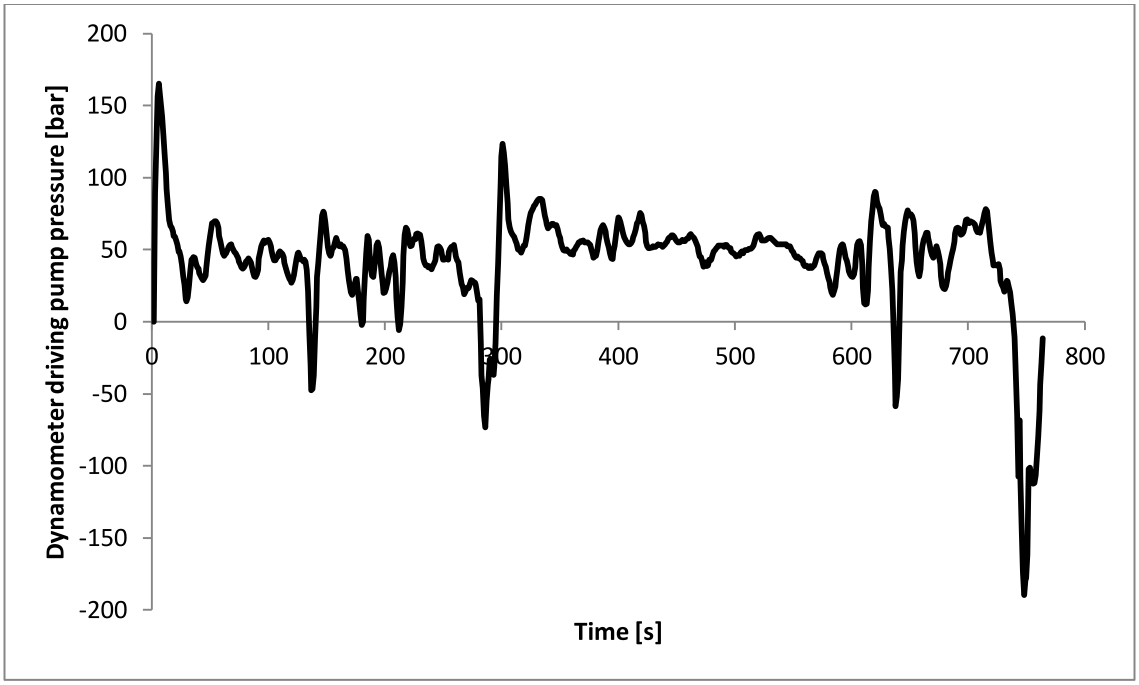

Figure 3.11: Dynamometer driving pump pressure magnitude remains below 200bar even during periods of reduced dynamometer driving pump displacement on HWFET

3.7 Simulink dynamic model of vehicle and dynamometer

A dynamic Simulink model of the dynamometer was constructed to aid development of a dynamometer controller. It is a tool to experiment with controllers more easily than with the actual hardware. It can be combined with the Generation 1 HHPV's model to validate proper function of the system as a whole. An overview of the Simulink vehicle with the new dynamometer model circled appears in Figure 3.12.

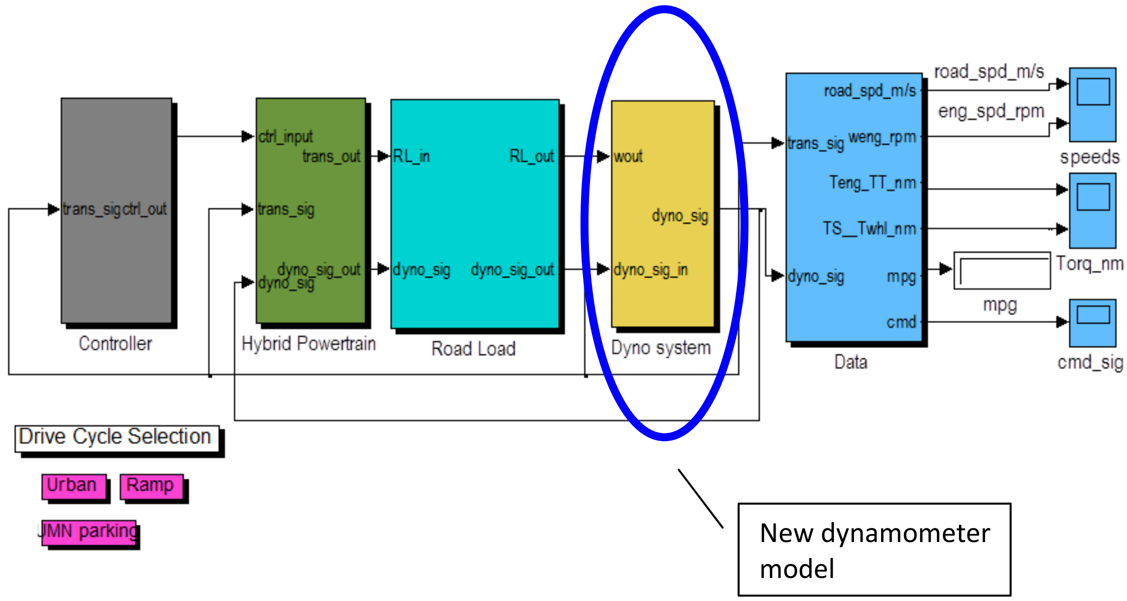

Figure 3.12: Overview of Simulink vehicle and dynamometer model

The dynamometer model can run separately from the vehicle model, which was created prior to the development of the dynamometer. It consists of a mathematical model of the physical dynamometer coupled to its controller. It assumes that the components described in section 3.8 beginning on page 55 are used. An overview of the dynamometer Simulink model appears in Figure 3.13.

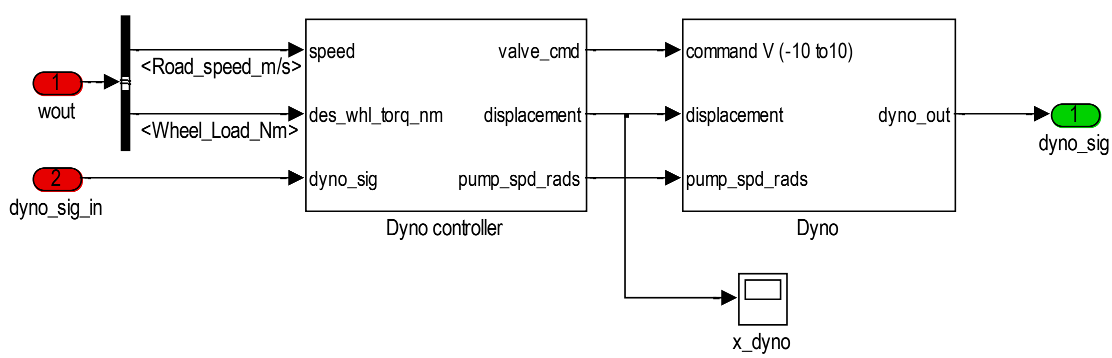

Figure 3.13: Overview of dynamometer Simulink model

Inputs to the simulation are vehicle speed and desired road force on the vehicle. The main output is actual torque at the dynamometer driving pump, but pressures and flows at various locations as well as accumulator fluid volume can also be monitored.

The following sections explain the models of the dynamometer and its controller in more detail.

3.7.1 Physical dynamometer model

The dynamometer model describes the behavior of the main system components. A simplified block diagram of this model appears in Figure 3.14. The simulation assumes that the dynamometer is configured as it is in Figure 2.3 on page 13. The Simulink block diagram for the physical dynamometer model appears in Appendix E. Equations for the models of the dynamometer driving pump, proportional directional valve, power unit, and accumulator are provided in subsections 3.7.1.1 through 3.7.1.4. The charge pump, cooling system, and minor losses are neglected. Inputs to the dynamometer model are proportional valve command voltage, the dynamometer driving pump's displacement command, and the dynamometer driving pump's speed. The proportional valve and pump commands can be controlled directly, but the vehicle controls the speed. The output of this model is torque.

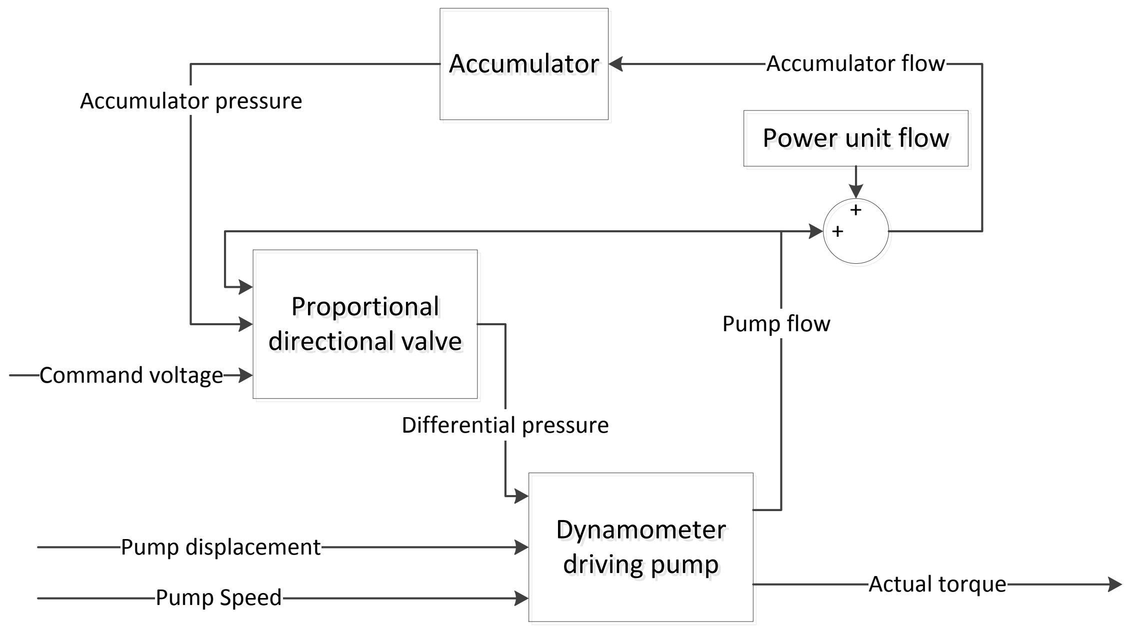

Figure 3.14: Physical dynamometer model block diagram

3.7.1.1 Dynamometer driving pump model

The dynamometer driving pump uses Equations (3.4) and (3.5) to output torque and flow based on pressure, displacement, and speed:

$$\boldsymbol{\tau = P \cdot x \cdot \eta_m} \tag{3.4}$$

where P is dynamometer driving pump's differential pressure in Pa, τ is torque in Nm at the dynamometer driving pump's shaft, x is the dynamometer driving pump's displacement in m^3/rad, and η_m is the pump's experimentally determined mechanical efficiency.

$$\mathbf{Q = x \cdot \omega \cdot \eta_v} \tag{3.5}$$

where Q is the dynamometer driving pump's flow rate in m^3/s, ω is the pump's shaft speed in rad/s, and η_v is the pump's experimentally determined volumetric efficiency.

The dynamometer driving pump was tested extensively in reference [15] to determine η_m and η_v. Torque, speed, pressure, and flow was measured with the pump on a test stand to determine η_m and η_v as a function of *x, P,* and *ω*.

3.7.1.2 Proportional directional valve model

The model for the proportional directional valve follows the equation for a variable orifice. It calculates the pressure drop between the high pressure line and one working port as well as the pressure drop between the tank line and the other working port. The valve's orifice constants are assumed to be proportional to input voltage. The model calculates pressure drops across the valve, *ΔP*, using Equation (3.6)

$$\Delta P = \left(\frac{\mathbf{Q}}{\mathbf{V} \cdot \mathbf{k}}\right)^2 \qquad (3.6)$$

where *V* is the command voltage and *k* is a constant for the valve, estimated based on observation to be $2 \cdot 10^{-7}\ m^3 s^{-1} V^{-1} Pa^{-1/2}$. The quantity *V·k* is the orifice constant. Since during throttling the inlet pressure is assumed to be higher than the outlet pressure, *ΔP* is positive.

Calculated pressures at the valve's working ports saturate at the maximum system pressure to simulate a relief valve opening and at 0bar to simulate an anti-cavitation check valve opening.

3.7.1.3 Hydraulic power unit model

The hydraulic power unit is modeled as a source of constant flow, producing $0.00031 m^3/s$ (19LPM).

3.7.1.4 Accumulator model

The accumulator is modeled using the ideal gas law such that its pressure depends on the volume of liquid contained in it and its precharge pressure. Expansion and compression of the gas is assumed to be isothermal. The accumulator pressure is given by Equation (3.7).

$$\boldsymbol{P} = \boldsymbol{P}_{\mathbf{precharge}} \cdot \frac{\mathbf{V}_{\mathbf{accumulator}}}{\mathbf{V}_{\mathbf{accumulator}} - \mathbf{SOC}} \qquad (3.7)$$

where $P_{precharge}$ is the accumulator's gas precharge pressure in Pa, $V_{\text{accumulator}}$ is the volume of the accumulator in m^3, and SOC is the accumulator's liquid volume in m^3, calculated using Equation (3.8).

$$\mathbf{SOC} = \int \boldsymbol{Q}(\boldsymbol{t})\mathbf{dt} \qquad (3.8)$$

where $Q(t)$ is the flow rate into the accumulator in m^3 as a function of time. $Q(t)$ is the sum of the flow from the dynamometer driving pump and the hydraulic power unit.

3.7.1.5 Simulated open loop dynamometer motoring performance

The model of the dynamometer can be independent of its controller model and can be run as an open loop system. For example, Figure 3.15 shows the results of a simulation of the dynamometer motoring with the proportional valve fully open (command is 10V) and the dynamometer driving pump at full displacement. The speed is equivalent to a vehicle ground speed of 10m/s. Torque drops with time because the dynamometer's accumulator is emptying, lowering the system pressure. This simulation assumes that the dynamometer's load is adequate to allow the dynamometer to reach this torque.

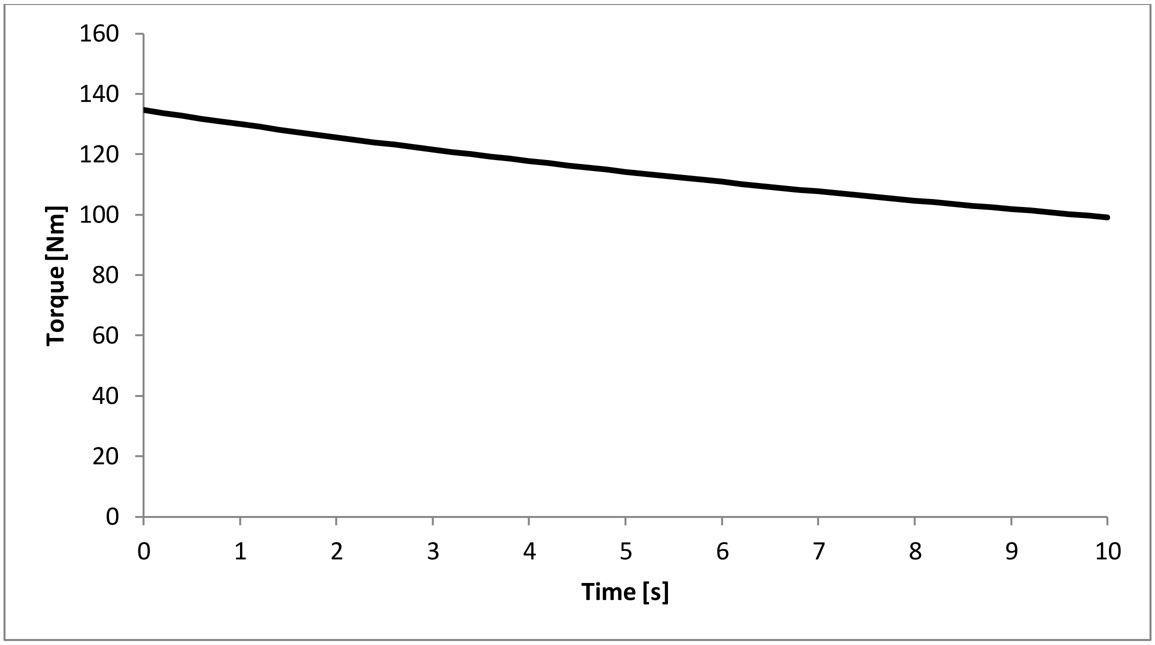

Figure 3.15: Simulated open loop performance of dynamometer motoring

3.7.2 Dynamometer controller model

A general block diagram of the dynamometer and its controller appeared in Figure 3.13. The block diagram of the dynamometer controller in its current state appears in Figure 3.16. The inputs to the controller are dynamometer shaft speed, dynamometer driving pump pressure differential, and actual torque measured at the dynamometer input shaft. Its outputs are proportional directional valve and dynamometer driving pump displacement commands. This controller varies the dynamometer driving pump's displacement to control the output torque. This method of operation is different from that of the MATLAB simulation discussed in section 3.6. Without a delay in this pump, changing the proportional directional valve command is not necessary. The proportional directional valve is left fully open for this simulation since it is assumed that the dynamometer driving pump's displacement changes fast enough. Therefore, the only output of the controller in Figure 3.16 is the dynamometer driving pump displacement. Figure 3.13 shows proportional valve position as a controller output; however, this is not currently used. Figure 3.13 shows a speed signal leaving the controller. Since the dynamometer does not control speed, it is omitted from Figure 3.16 for clarity. The controller calculates the appropriate torque, τ_{ref}, based on vehicle speed and acceleration using Equation (3.1) on page 36 in the same manner as the MATLAB simulation.

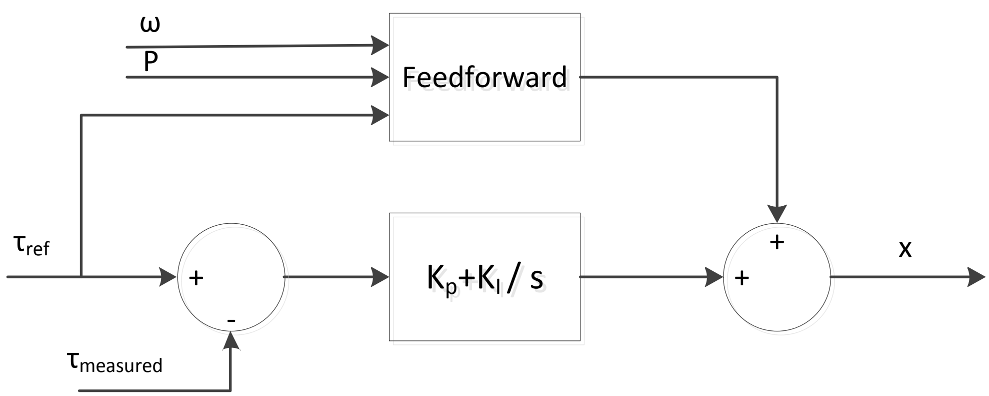

Figure 3.16: Dynamometer controller block diagram

The transfer function for this controller is

$$\mathbf{X_{input}(s) = X_{feedforward}(P, \tau_{ref}, \omega) + \left(K_p + \frac{K_I}{s}\right) \cdot (\tau_{ref} - \tau_{measured})} \tag{3.9}$$

K_p and K_I are the proportional and integral gains, respectively. They can vary depending on the design of the controller.

Since the purpose of the Simulink model is to evaluate controllers, the controller can easily be changed. Results are presented in Figure 3.17 for an example PI with feedforward controller. The input to the system is a $3m/s^2$ speed ramp. This is equivalent to the vehicle accelerating at a constant rate from a stop to 15m/s in 5s. Therefore, the desired torque is roughly constant with time but does increase slightly as the vehicle gains speed due to increasing drag. This example shows that the controller can track torque. However, a response where the actual torque does not overshoot the desired torque would be preferable if it can be achieved without tradeoffs. The overshoot plateaus between 2s and 2.5 s because this particular simulation saturates the pump displacement command at 28cc/rev.

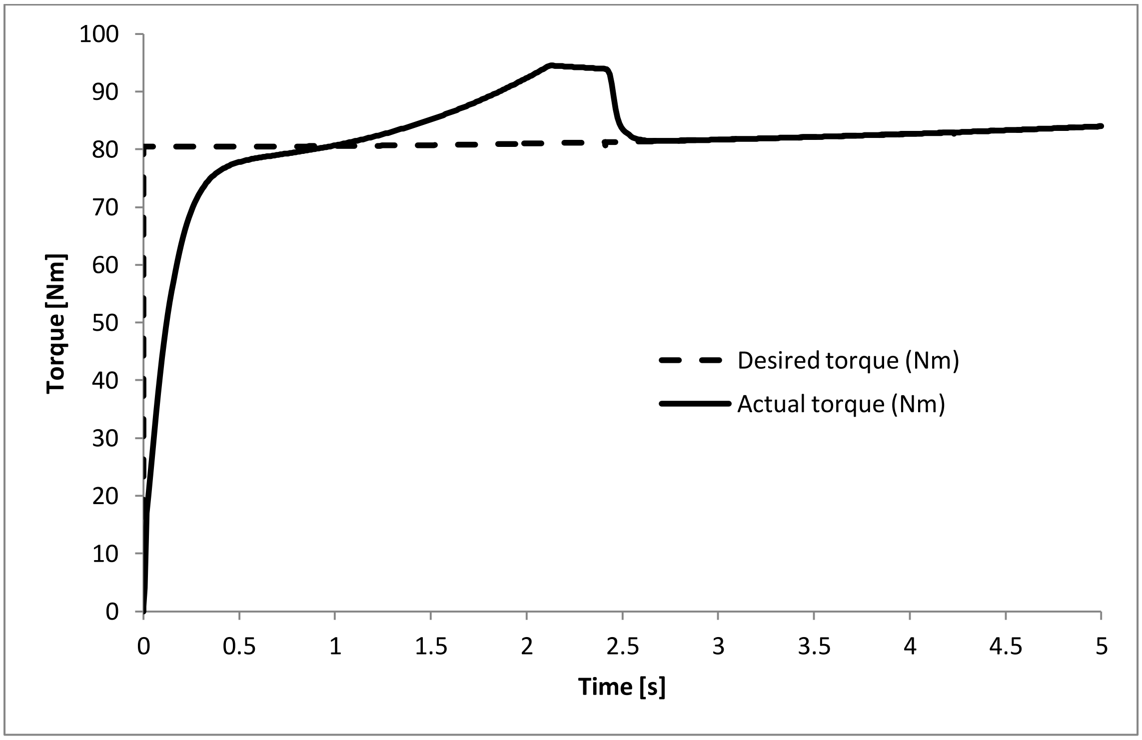

Figure 3.17: Imperfect simulated torque tracking using an example controller

3.8 Component selection

Subsections 3.8.1 through 3.8.3, 3.8.5 through 3.8.8, and 3.8.10 describe components reused from previous projects to reduce cost. The dynamometer driving pump, proportional directional valve, hydraulic power unit, high pressure filter, data acquisition equipment, cold water supply, and concrete bed plate were reused. Subsections 3.8.4, 3.8.9, and 3.8.11 describe the selection of purchased components including the accumulator, oil cooler, and drive shaft.

3.8.1 Dynamometer driving pump

Two Sauer-Danfoss Series 42 28cc/rev axial piston pumps were available and selected for use together as the dynamometer driving pump. Their displacement is infinitely variable from -28 to 28cc/rev. They can be coupled together in a tandem configuration to create a total maximum displacement of 56cc/rev. This tandem configuration offers no advantage to this application over a single 56 cc/rev pump and in fact increases the complexity and cost of plumbing and wiring. These pumps also can only accommodate a charge pump with an uncommon 11 tooth spline, limiting the selection of charge pumps available with an acceptable lead time. They also require a low pressure relief valve on the return line to create about 14bar of backpressure so that their charge flow consumption remains low. We do not understand why the backpressure is necessary.

A larger 100cc/rev pump was available but not selected for use because its displacement was too large for the application. It would have to be operated in an inefficient low displacement regime, limiting the power available for motoring. This pump could be used later if testing a more powerful vehicle is necessary.

3.8.2 Proportional directional valve

An Eaton KFDG5V-7 two stage proportional directional valve was available and selected to provide a faster alternative to control load than by changing the pump displacement. Although its specifications meet or exceed requirements of the application, the valve required substantial effort to become usable. The valve is two stage and pilot operated. The pilot stage is a solenoid operated directional valve that provides pilot pressure to move the main stage spool. It requires a sandwich reducing valve to be installed between the main stage and the pilot stage to reduce pressure for the pilot stage [12, p. 2]. This was purchased and installed. We also needed to purchase the proper amplifier, connectors, and wiring.

Although it is common to use a proportional directional valve to control a motor's speed, using one to control a pump's speed is uncommon. A proportional relief valve was considered for load control during absorbing (pumping). This valve would replace the proportional directional valve to control the outlet pressure of the pump directly. However, we decided to use only the existing proportional directional valve to avoid the cost of purchasing a new valve. Verification of the proportional directional valve's ability to control load of a pump was therefore necessary.

To verify the valve's ability to control a pump's speed, the circuit shown in Figure 3.18 was built. The shaft of a fixed displacement pump was coupled to the shaft of a fixed displacement motor. Torque and speed were measured at this connection. Since the motor was connected to a constant pressure source, it provided a constant torque. The pump was connected to the valve. The pump's displacement was kept high enough to prevent over pressurization, however, using a relief valve in addition would have been safer.

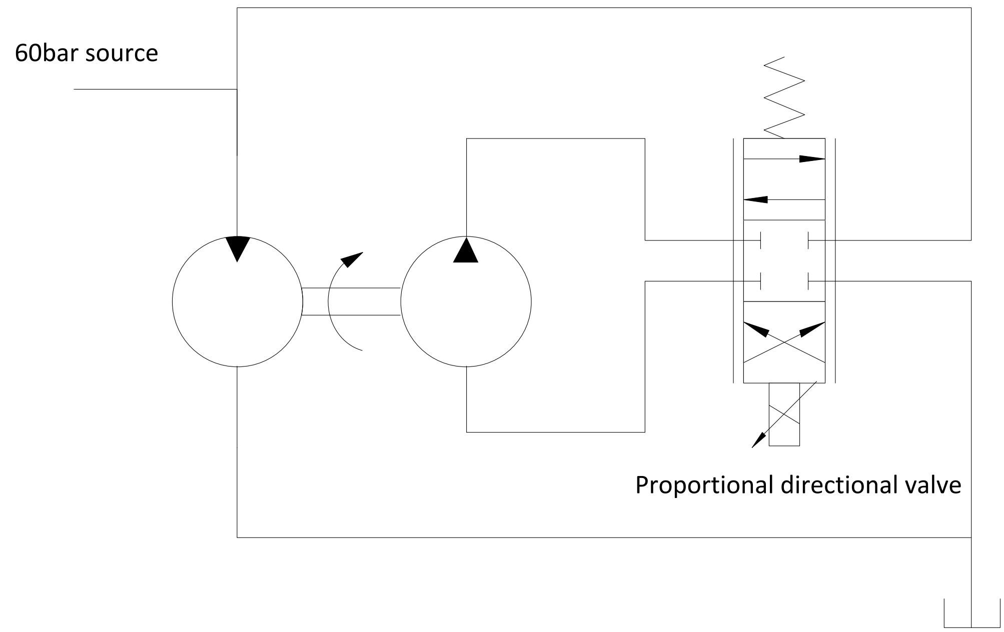

Figure 3.18: Proportional valve verification circuit

Changing the valve's spool position changed the shaft speed. When the spool was near its center position, the pump was locked, and the shafts did not spin. As the spool moved towards one end, shaft speed increased and the magnitude of torque decreased. The spool can travel from one end to the other linearly with a command voltage ranging from -10V to 10V.

For this experiment, the valve's command was varied between -8V and -4V. The system was held at steady state with the valve position fixed at discrete points. Figure 3.19 shows that the valve can vary the speed of the pump between 100rpm and 1300rpm. Figure 3.20 shows the torque of the pump varying for the same operating points. These figures show that the valve can control an overrunning load. Since the mechanical load on the pump is fixed, the small torque range causes a large change in speed; the torque in this experiment varied only between -26Nm and -30Nm. When the valve signal was close to -4V, the pump sputtered and nearly stalled, possibly causing the decrease in torque magnitude for the two data points close to -4V.

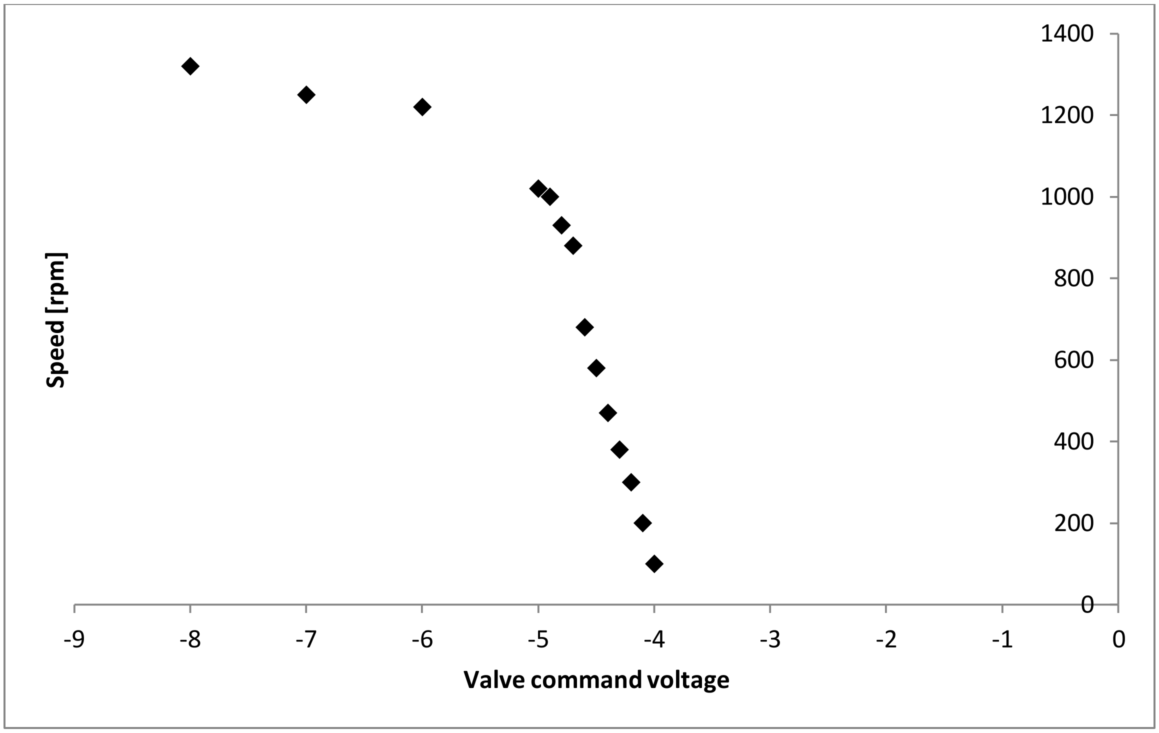

Figure 3.19: Pump speed versus command voltage for proportional valve test

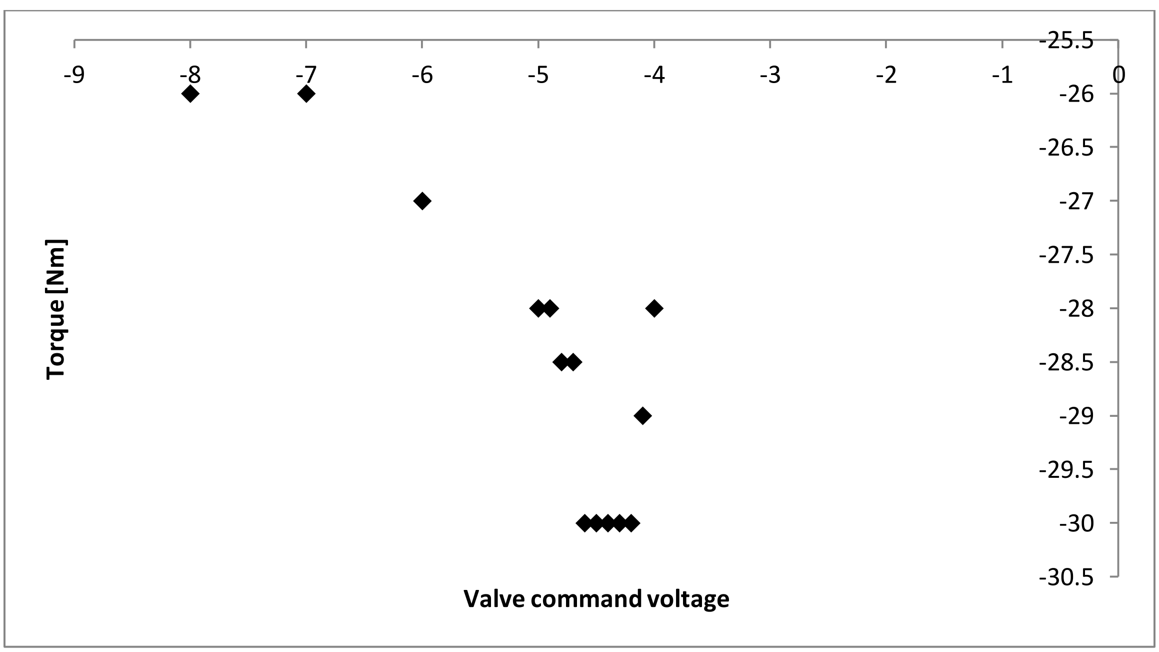

Figure 3.20: Pump torque magnitude versus command voltage for the proportional valve test

3.8.3 Hydraulic power unit

The dynamometer's driving pump requires up to 150LPM when motoring at full displacement during the maximum speed of the EPA drive cycles. The actual flow requirement may be less if the dynamometer driving pump's displacement is reduced. The power unit is only capable of producing 19LPM, so an accumulator was necessary to supplement flow.

The power unit contained a relief valve and filter, but they were unable to accommodate the maximum flow rates of 150LPM and fluid cleanliness requirements of the system, which require a 3 micron or finer filter element. The existing filter was replaced with one with a larger capacity and a finer 3 micron element. The relief valve was duplicated with one designed for a higher flow rate.

Cavitation of the suction line was a concern since the tank could not be raised or pressurized. This required the use of large hose and valves on the suction line. The power unit's tank did not previously have a suction line and required one to be installed. Suction flow was necessary for the charge pump and anti-cavitation check valves.

The maximum pressure of the power unit is 200bar, setting a ceiling for the system. This limit is practical for other components and did not affect the overall design of the system.

3.8.4 Accumulator

Bladder and piston style accumulators are available. The bladder style was selected due to its lower cost and lower seal friction. The MATLAB simulation in section 3.6 indicated that a 38L accumulator should provide adequate energy storage capacity for the dynamometer to complete the EPA's UDDS and HWFET cycles. A Parker BA10B3T01-A1 accumulator was ultimately selected.

3.8.5 Torque and speed sensor

A Honeywell 1105H-101 slip ring torque sensor was available from a previous pump test stand project. It also contains a speed sensor module. This sensor can easily measure the full torque and speed required by the project. It requires regular maintenance according to reference [16]. In particular, the slip ring assembly should be cleaned if the torque signal is noisy, and the brushes should be replaced once they have worn to less than 8mm in length.

Software to interface with this sensor was already made in Simulink with xPC Target. This system is described in reference [15, p. 25]. Changing to a different DAQ system would have required substantial effort to remake this interface software.

3.8.6 Data acquisition equipment

The existing pump test stand used two office style desktop PCs to run xPC Target for controls and data acquisition. This setup is unreliable; communication between host and target PCs often drops, and the DAQ cards are easily damaged. Hardware designed for laboratory use would be preferable, as would a system requiring only a single computer. However, the cost and time required to integrate a new computer system was prohibitive, so the existing system was used.

3.8.7 Cold water supply

A chilled water supply was not available in the room housing the dynamometer. Therefore, the existing cold water supply for the sink needed to be used. The oil cooler needed to be sized to use no more than the flow available from this sink, about 10LPM.

3.8.8 High pressure filter

Instructions for the proportional valve recommend using a 3 micron or finer filter on the high pressure line upstream of the proportional valve. A high pressure filter was available, but its micron rating of 5 did not meet this specification. Replacing this filter with a new high pressure filter was too expensive, so it was instead supplemented with an inexpensive 3 micron return line filter.

3.8.9 Oil cooler

The MATLAB simulation indicated that the dynamometer would absorb 10-15kW of power from the vehicle that would need to be dissipated in the form of heat. The return oil flow rate would be 50LPM on average and could reach a maximum of 150LPM. The temperature of the hydraulic oil should not exceed 70°C. The flow rate of cool water available from the sink is about 10LPM at 10°C.

A Thermasys EK series shell and tube oil cooler was selected because of its low cost and common availability. The limited flow rate of cool water required an oil cooler sized for at least a 5:1 oil to water ratio. Using instructions in the manufacturer's catalog [17], the power removed in the cooler was calculated to be 7kW (9hp), and the oil flow was calculated to be 50LPM (13GPM). Any line above this point on the graph titled "7:1 Oil to Water Ratio – Lower Water Usage" on page 78 of reference [17] corresponds to an oil cooler that will work. The line corresponding to model EKS-714-F was selected to be slightly oversized for this application. Although the dynamometer is capable of absorbing more than 7kW for short times, the oil cooler only needs to remove the average power.

3.8.10 Concrete bed plate

The concrete bed plate provides a stable, heavy surface for mounting components. All components with the exception of the power unit must fit on the existing bed plate's 1.9 by 1.2m surface. Drilling new holes into the concrete material is difficult, so moving or remounting components once they are installed is difficult.

3.8.11 Driveshaft

A driveshaft is necessary to couple the vehicle's transmission output to the dynamometer's torque sensor. Universal joints are necessary on both ends to allow the shafts to be misaligned and off center. Machine Service, Inc. was selected to design and build the driveshaft to order. The data in Table 3.5 was provided to Machine Service, Inc.

Table 3.5: Driveshaft design parameters

Parameter	Value
Dynamometer driving pump inertia	0.004kg-m^2
Maximum torque	330Nm
Typical torque	70Nm
Maximum speed	2900rpm
Typical minimum speed	600rpm
Vehicle engine	3 cylinder, 1.1L
Combined inertia of engine and transmission	Unknown

Based on this information, the manufacturer recommended a rubber isolated tube style driveshaft. It includes damping to eliminate the possibility of torsional vibrations, which would cause the torque to fluctuate twice per revolution and cause damage.

4 Results and performance of dynamometer

Section 4.1 describes limitations of testing the dynamometer due to vehicle reliability issues. Section 4.2 describes a test in which the dynamometer motored with open loop control. Section 4.3 describes the process used to determine the dynamometer's transfer function experimentally and how it will be used for the automatic torque controller. The torque controller varies the dynamometer driving pump's displacement to produce the desired torque. Section 4.4 describes the performance of the automatic torque controller. Section 4.5 describes results from a test when the dynamometer absorbed power from the vehicle. Section 4.6 evaluates the automatic torque controller. Section 4.7 describes the practical environmental, usability, and reliability aspects of operating the dynamometer.

4.1 Vehicle reliability limited testing

Poor reliability of the Generation 1 HHPV hindered full evaluation of the dynamometer operating on a drive cycle with the vehicle. The connection between the vehicle's engine and transmission was particularly unreliable and prevented substantial testing with the vehicle powering the dynamometer and the dynamometer absorbing power. Evaluation of torque tracking in response to vehicle speed changes, measuring the dynamometer's maximum torque and speed, measuring maximum motoring duration, and tracking of an EPA cycle was desired but not possible. Testing by motoring the vehicle using the dynamometer was possible. The dynamometer could be coupled to the transmission with the vehicle's engine disconnected from its transmission. These tests could provide data on the motoring performance of the dynamometer as well as the energy recovery capabilities of the vehicle's hydraulic system.

4.2 Dynamometer open loop control demonstrates motoring torque of 85Nm

For this test, the dynamometer motored the transmission of the vehicle. The vehicle's engine was disconnected from its transmission, and its pumps were stroked to full displacement in order to provide a resistance to the dynamometer. The torque and speed of the dynamometer are shown in Figure 4.1 for a portion of this test. At about 805s into the test, the dynamometer driving pump displacement was manually increased from 0.4 to 1.0 for the remainder of the time shown in the plot. This caused its motoring torque to increase from 30Nm to 85Nm which accelerated the transmission from 1100rpm to 2400rpm. Near this time, the motoring power peaked at 17kW. As the dynamometer's accumulator emptied, the torque dropped. The increase of torque beginning near 807s may be a result of a changing load on the vehicle or the changing efficiency of the dynamometer driving pump.

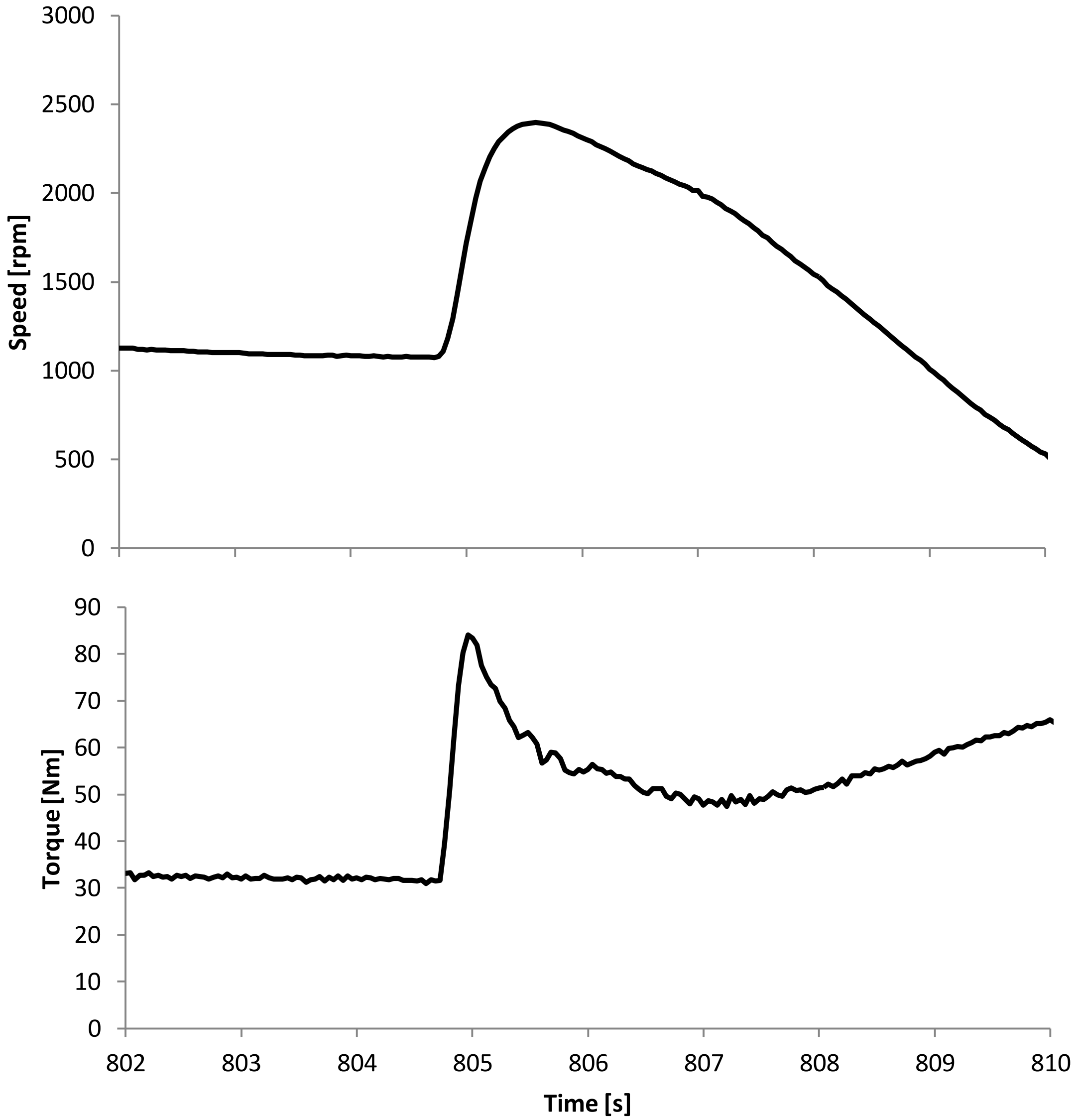

Figure 4.1: Dynamometer open loop torque and speed

The dynamometer is theoretically capable of producing 170Nm. The reason that it did not produce its maximum torque is likely because the transmission did not provide enough resistance without its engine coupled to it. The dynamometer did not attempt to produce torque at low speeds due to reasons explained in section 5.3.

In addition to the capabilities of the dynamometer, this test demonstrated the energy recovery capabilities of the vehicle's hydraulic system. The pressure of the vehicle's accumulator rose from 80bar to 100bar during the course of the test shown in Figure 4.1.

4.3 Dynamometer transfer function obtained through unloaded motoring

In order to design a torque controller properly, the transfer function for the dynamometer system needed to be identified. To do this, the dynamometer was decoupled from any load and motored. A simple PI controller was used to control the speed. The dynamometer tracked a sinusoidal signal with increasing frequency (chirp). The minimum speed of the chirp signal was 560rpm, and the maximum was 960rpm. At high frequencies, tracking of the chirp signal deteriorated. A Bode plot comparing the actual speed to the dynamometer driving pump displacement appears in Figure 4.2.

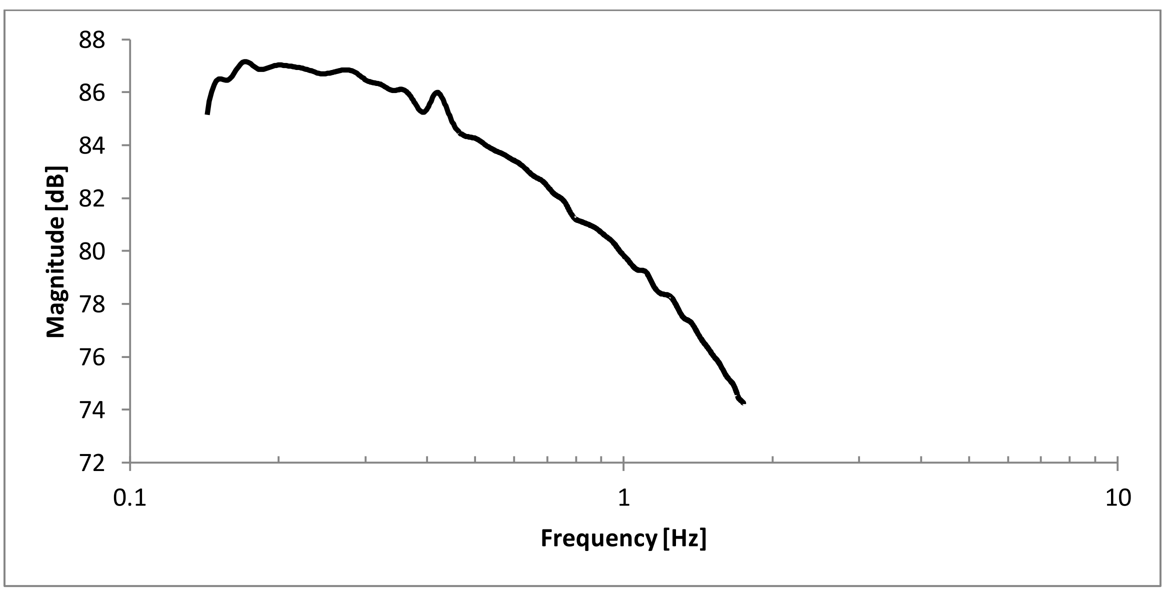

Figure 4.2: Dynamometer system ID Bode plot

The magnitude on the y-axis in dB is given by Equation (4.1).

$$dB = 20 \cdot \log_{10}\left(\frac{RPM}{x}\right) \tag{4.1}$$

where *RPM* is the magnitude of the dynamometer driving pump's shaft speed in RPM and *x* is the magnitude of the dynamometer driving pump's displacement scaled to a range of -1 to 1. This equation was not evaluated at displacements near 0 because the dynamometer driving pump could not produce enough torque to spin.

The Bode plot was used to determine the transfer function of the dynamometer with driving pump displacement (on a scale of 0 to 1) as an input and speed (in rpm) as an output. The transfer function appears as Equation (4.2).

$$\boldsymbol{H(s) = \frac{714500}{s^2 + 12.36 \cdot s + 30.17}} \tag{4.2}$$

This transfer function is valid for pressures around 180bar. Although it is a transfer function for speed, it is useful for the torque controller as well because the dynamometer driving pump's swashplate dynamics are the same regardless of whether speed or torque is being controlled.

4.4 Dynamometer closed loop automatic torque control

Using the transfer function from section 4.3, a torque controller was developed for the dynamometer. For this experiment, the operator entered a desired torque manually into the computer. The torque controller operated by the Simulink program coupled to the xPC Target real time control system adjusted the dynamometer driving pump displacement so that the actual and desired torque match. For later experiments, the desired torque could be calculated automatically as a function of speed and the vehicle parameters outlined in section 3.5 beginning on page 35.

The displacement of the pumps in the vehicle's transmission was fixed throughout the test. The vehicle's engine was decoupled from the transmission, and the dynamometer motored. Figure 4.3 shows that the torque controller initially maintained the dynamometer's torque at 25Nm. The reference torque then increased as a step to 30Nm. The actual torque followed the step and reached the desired torque of 30Nm 0.2s after the step. Overshoot is within the noise of the torque sensor. The bottom portion of Figure 4.3 shows the speed increasing in response to the torque increase. Steady state error is not apparent.

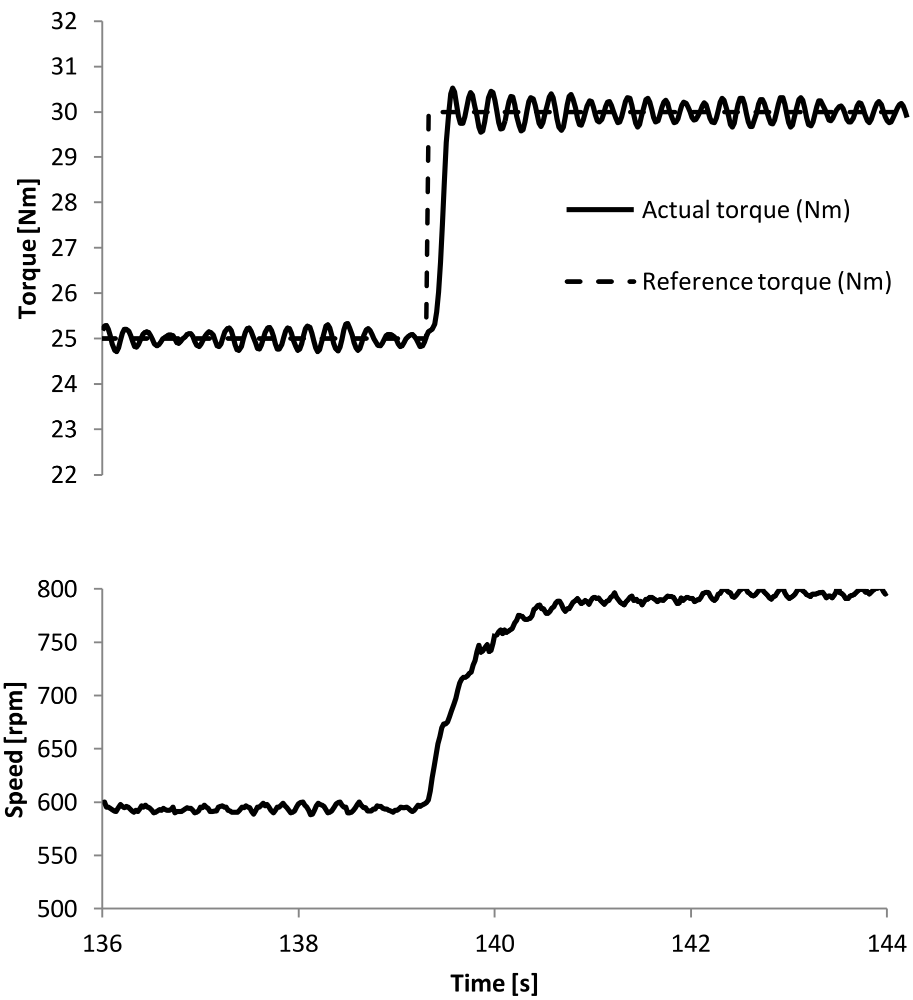

Figure 4.3: Closed loop torque control

4.5 Absorbing performance

The dynamometer was able to demonstrate absorbing power from the vehicle for a brief period of time before a freewheeling clutch between the vehicle's engine and its transmission failed. The dynamometer driving pump displacement was controlled manually. The vehicle's engine speed and pump displacements were set manually. Results from a portion of this test appear in Figure 4.4. Negative torque corresponds to the dynamometer absorbing power. The torque absorbed by the dynamometer peaked at -42Nm. The speed at that time was 900rpm, corresponding to 4kW of absorption. The dynamometer is designed to be capable of absorbing more power and torque; the vehicle was the limiting factor for this test. Since both machines could change the torque and speed, there is no direct indication of which one is responsible for the changes in torque and speed observed in Figure 4.4. For example, either an increase in vehicle throttle or dynamometer driving pump displacement could cause an increase in torque magnitude.

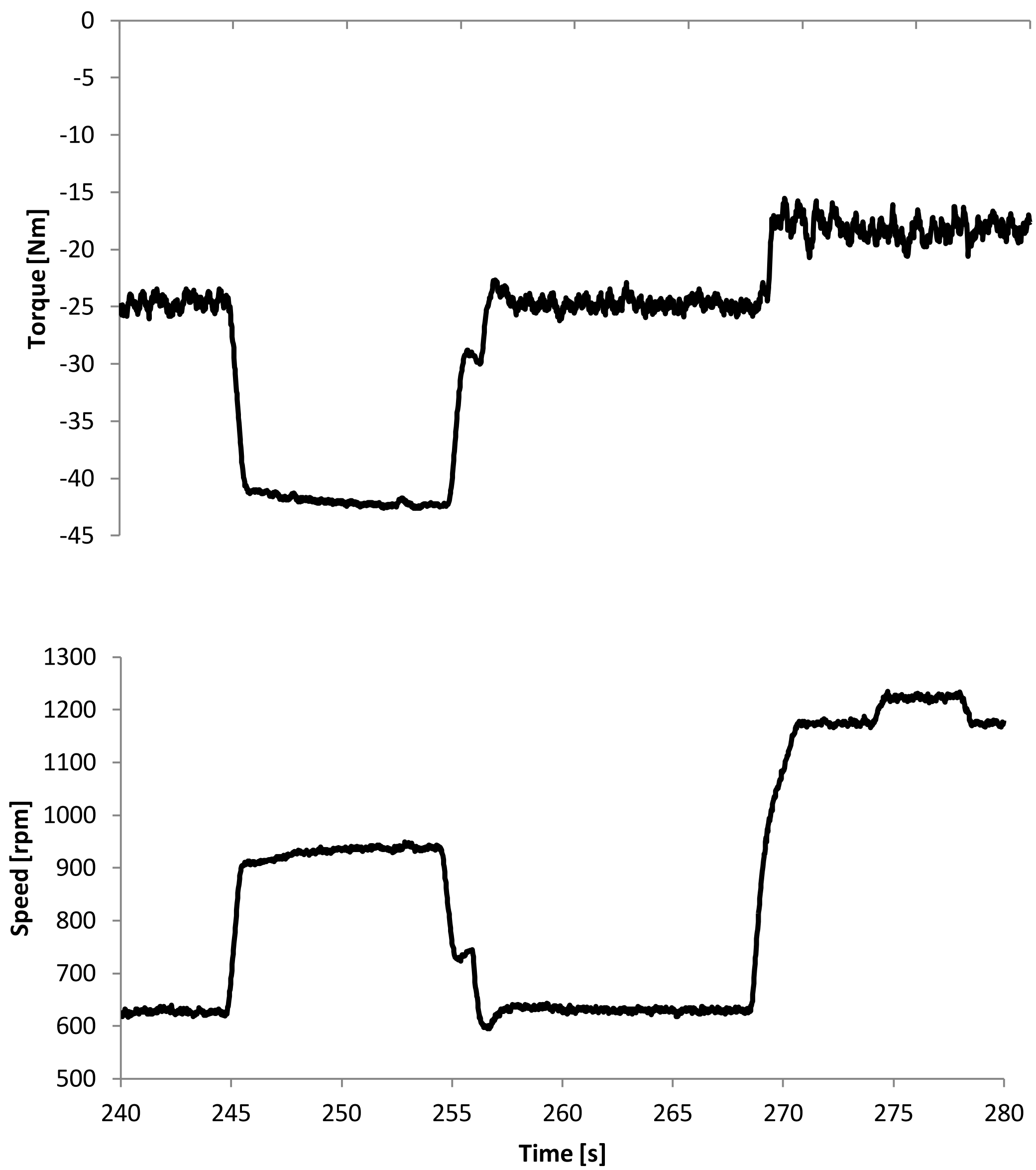

Figure 4.4: Dynamometer absorbing performance

4.6 Evaluation of controller

A description of the operation of the controller appears in section 2.6 on page 24. Although the vehicle currently prohibits the dynamometer from completing an EPA drive cycle, the speed and accuracy of closed loop torque tracking have been proven. In section 4.4 beginning on page 68, the dynamometer responded to a 5Nm torque step within 0.2s. Since the EPA drive cycles provide speed data points every 1s, this response time is likely to be fast enough to track the drive cycles without using the dynamometer's proportional directional valve to improve the dynamic response of torque tracking. The controller does not currently use the proportional directional valve.

The selection of hardware contributes to the controller's fast performance. The inertia of a hydrostatic dynamometer is low compared to other styles of dynamometers. Low inertia allows the dynamometer to change speeds quickly. Furthermore, the dynamometer driving pump has the fastest displacement control solenoid available from its manufacturer for this series of pumps.

4.7 Usability and environmental assessment

The dynamometer generally meets the usability, safety, and environmental requirements set prior to the start of this project. It has proven to be easy enough to learn how to use that a graduate student already familiar with MATLAB became a proficient operator within a day.

The dynamometer has begun to show that it is reliable and requires little maintenance. In about four months of operation it has had only one day of unscheduled downtime due to a blown fuse in its hydraulic power unit. After the control box was disconnected and reconnected, the hydraulic power unit started unexpectedly. The fuse blew in the process of troubleshooting the unexpected startup. The reason for the unexpected startup is not fully understood but seems to depend on the order in which the controls are connected. Since the rest of the dynamometer's hydraulic system was shut off, it did not build pressure. Scheduled maintenance has included replacement of the low pressure filter element.

A larger room would improve the operator's ability to move about, make repairs on site, and move the vehicle in and out. The current space available is 5 by 6m; double that area would be preferable. Cord drops from the ceiling would be preferable to running wires across the floor to eliminate tripping risks. The ventilation system is adequate to remove the exhaust from the vehicle. Noise is a concern as the test is audible from a neighboring office. However, most of the noise comes from the vehicle rather than the dynamometer. The dynamometer has no unintentional leaks, but the vehicle does leak gear oil from its transmission and requires a drip pan underneath.

5 Conclusion

This chapter summarizes the accomplishments of this project and the importance of this work. It also includes recommendations to improve and expand the capabilities of the dynamometer.

5.1 Review

Chapter 1 introduced the vehicle that the dynamometer is designed to test. It listed the requirements and goals for the project. The EPA test procedure was summarized along with other dynamometer projects with similar objectives.

Chapter 2 provided a detailed description of the dynamometer and described its hardware implementation and how it works. A hydraulic schematic of the dynamometer was presented, and each component was explained. The physical configuration of the machine was described. Manual and automatic methods of controlling the dynamometer were described. Finally, safety features and ergonomics were explained.

Chapter 3 described the design process followed to converge on the final product. Using contract testing services or commercially available dynamometers was considered and ultimately rejected. A purpose built hydrostatic dynamometer was selected after comparing three styles of dynamometers. General component specifications were developed using a MATLAB simulation, and a dynamic model was created in Simulink. Reuse of existing components to reduce cost imposed design constraints.

Chapter 4 provided experimental results to evaluate the performance of the dynamometer. The dynamometer torque controller was developed with the dynamometer disconnected from the vehicle. Some testing and verification was performed with the dynamometer and vehicle connected.

5.2 Contributions

The primary accomplishment of this thesis work is the construction of a hydrostatic dynamometer to test the Generation 1 HHPV. A special feature of this machine is its ability to motor as well as absorb power from the vehicle. Experimentally measured power, torque, speed, and response times indicate that the dynamometer will likely be able to complete EPA drive cycles when coupled to the Generation 1 HHPV. This dynamometer will validate the fuel economy of the vehicle. The dynamometer will also allow convenient and repeatable retesting which is necessary for the comparison and refinement of controls algorithms. This will allow several students to include experimental results in their theses.

The success of this dynamometer also demonstrates the feasibility of using hydraulics for dynamometers in general. Hydraulics offer the opportunity to reduce dynamometer cost and improve dynamic performance. Since many components were reused for this dynamometer, its cost of parts was low. Manufacturing hydrostatic dynamometers on a larger scale would reduce the high labor costs. Although all components would need to be purchased for a mass manufactured hydrostatic dynamometer, in many cases, simpler inexpensive components could be used.

5.3 Future work

The source of charge pressure for the dynamometer driving pump is a fixed displacement charge pump directly connected to the shaft. The dynamometer cannot produce torque at low speeds below about 100rpm because the charge pump spins too slowly to produce enough flow to maintain charge pressure. This behavior is a byproduct of a safety feature designed to prevent the dynamometer from turning backwards. If the possibility of the dynamometer turning backwards is acceptable, a reducing valve could be added to reduce the high pressure to maintain charge pressure at all times. However, this method is inefficient and would diminish the power available for motoring.

Shortcomings of the Generation 1 HHPV's reliability have prevented full validation of the capabilities of the dynamometer. Vehicle reliability and durability must improve to enable full dynamometer performance data to be gained. Reliability of the connection between the vehicle's engine and transmission has been particularly poor; without this connection, the absorbing capabilities of the dynamometer could not be thoroughly tested. In addition, the shaft connection to the sun gear of the planetary gear train utilized within the transmission of the vehicle began slipping once during testing. The transmission needed to be disassembled to correct the slipping shaft. If it slips again, the design should be modified.

Once the Generation 1 HHPV is running reliably, the dynamometer's software may need to be tuned to improve drive cycle tracking and accuracy. Refining the torque controller based on observations of performance will allow better tracking of the EPA drive cycles. The coefficients in Table 3.4 on page 36 used to calculate road load in Equation (3.1) on page 36 are estimates and should be verified before fuel economy data for this particular vehicle is published. The coefficients can be determined experimentally based on the deceleration of the vehicle coasting on a test track. However, different coefficients could be entered into the dynamometer to simulate the Generation 1 HHPV's powertrain in a different vehicle.

The proportional directional valve shown in Figure 2.3 on page 13 is not currently used because the dynamometer driving pump's displacement control appears to be fast enough. However, if faster dynamics are desired, controls for this valve will need to be developed.

5.4 Possibility of testing other machines

The dynamometer is not limited to testing only the Generation 1 HHPV. It could test another machine or vehicle with approximately the same power. Prime movers such as motors or engines could be tested. At maximum torque and speed, it could theoretically absorb 50kW for short periods. Typical sustained operation could be around 15kW as long as oil temperature is monitored carefully. Sustained low speed operation below 500rpm is not recommended since the dynamometer driving pump may be unable to produce its full torque. A prime mover must start the dynamometer before it can reliably absorb torque.

If necessary for testing a different machine or drive cycle , increasing the displacement of the dynamometer driving pump could increase the power capacity of the dynamometer. A 100cc/rev model is available which would roughly double the available power. The size of the accumulator, oil cooler, and plumbing would likely need to increase as well to accommodate the increased flow rate. A larger accumulator alone would increase the possible motoring duration.

A CCEFP member company has expressed interest in testing its customer's riding lawnmowers on this dynamometer. A zero-turn mower may be able to be tested through its wheel with the other wheel locked hydraulically. The best way to test a vehicle would be in a similar manner to the Generation 1 HHPV with the differential removed. A vehicle could be tested through one wheel if the other driving wheel is locked and the speed is sufficiently high for reliable operation of the dynamometer's driving and charge pumps.

The construction of this dynamometer offers the opportunity to validate the performance and efficiency of a variety of machines, including the Generation 1 HHPV for which it was designed.

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Appendix A: Bill of Materials

Total cost column includes shipping charges if applicable.

Hydraulic valves and power							
Item	**Requirements**	**Manufacturer**	**Model number**	**Supplier**	**Quantity**	**Unit Cost**	**Total cost**
Proportional directional valve		Eaton	KFDG5V-7-33CH60N-EX-VM-U1-H1-10	Closet in ME 458	1	$ -	$ -
Reducing valve	for proportional directional valve	Eaton	DGMX2-3-PP-BW-S-40	eBay	1	$ 45	$ 55
Pump(s)		Sauer Danfoss	Series 42			$ -	$ -
Valve amp		Vickers	EEA-PAM-561-A-31	eBay	1	$ 300	$ 310
Subplate			AD07SPSO12S	Daman	1	$ -	$ -
Accumulator	10 gallon	Parker	BA10B3T01-A1	Parker	1	$ -	$ -
Relief valve	3 for circuit and 1 for charge flow	Eaton	RV5-10-S-0-35		4	$ -	$ -
check valve, anticavitation	<10psi pressure drop @ 10gpm	Eaton	CV1-16-P-0-5		1	$ -	$ -
check valve, anticavitation		Parker	C-2020-S	eBay	1	$ 40	$ 54
Charge pump	~15 cc	Hydreco	162025A2	Surplus Center	1	$ 106	$ 119
Accumulator dump valve	normally open	Eaton	SV13-16-O-0-00		1	$ -	$ -
Accumulator dump orifice	Needle valve, set screw, size 8 connections	Sun	NFCC-KDN		1	$ -	$ -
Nitrogen gas	For precharging accumulator		CX60180	Ustores	2	$ 4	$ 9
Crossover plate	To replace and	Daman	AD07COP	Air-	1	$ 149	$ 159

	bypass proportional directional valve			Hydraulic Systems			
Fluid connectors							
Item	**Requirements**	**Manufacturer**	**Model number**	**Supplier**	**Quantity**	**Unit Cost**	**Total cost**
Hose and fittings		Aeroquip	PO# 0000491841	Air Hydraulic Systems	1	$ 831	$ 851
Pipe fittings			PO# 0000491797	Grainger	1	$ 46	$ 46
ORB adapter		Aeroquip	4VTA8	Grainger	1	$ 16	$ 16
NPT/JIC adapter		Aeroquip	2F291	Grainger	1	$ 4	$ 4
Cooling and filtration							
Item	**Requirements**	**Manufacturer**	**Model number**	**Supplier**	**Quantity**	**Unit Cost**	**Total cost**
Oil cooler	10000-15000W (34000-51000 Btu/hr) cooling capacity, 70C max oil temp, 140 LPM max oil flow, 50 LPM average	Thermasys	EKS-714-F	SunSource	1	$ 403	$ 423
Low pressure filter	Should be non bypass, ideally before inlet to proportional valve. Must meet ISO 18/16/13	Parker	09621509	MSC	1	$ 58	$ 58
Low pressure filter element		Parker	09621665	MSC	1	$ 35	$ 35
High pressure filter		Eaton	HF2P4SA1JNB2H05		1	$ -	$ -
Pressure gauge	1/8" connection	Ashcroft	56479116	MSC	1	$ 13	$ 13
Hose for water	1"ID		79815304	MSC	10	$ 2	$ 15
Solenoid valve for water	3/4" NC		4711K841	McMaster	1	$ 101	$ 106
Adapter tube for exhaust pipe	4"x.188", 3"L		14014	Discount Steel	1	$ 13	$ 23

Suction strainer		Mueller Steam Specialty	1RNC8	Grainger	1	included in pipe fittings	
Replacement screen for suction strainer			1RNT5	Grainger	1	not purchased	
Filter element for power unit (not used)		Sperry Vickers	573082				$ -
Fiberglass sleeve	Fit over exhaust tube to prevent burns		31950884	MSC	10	$ 2	$ 18
Measurement, sensors, and electronics							
	Requirements	**Manufacturer**	**Model number**	**Supplier**	**Quantity**	**Unit Cost**	**Total cost**
LabVIEW or MATLAB							$ -
Computer							$ -
Computer cart				ReUSE	1	$ -	$ 9
Mouse pad				Ustores	1	$ 3	$ 3
Mouse	USB		LOG910001600	Ustores	1	$ 10	$ 10
Torque sensor		Honeywell	1105H-101				$ -
Speed sensor		Honeywell	Included with torque sensor				$ -
Pressure gauges							$ -
Thermometer	Allows remote viewing of temperature		5834619	MSC	1	$ 25	$ 25
Brushes for torque sensor	Special (15psi) spring stiffness	Honeywell	064-LW10983-11	Precision Measurement Products	8	$ 22	$ 184
Flash drive	For transporting data		IVR37608	Ustores	1	$ 6	$ 6
Crossover cable	To extend cable from vehicle		BLKA3X12607YLWM	Ustores	1	$ 3	$ 3
Crossover cable coupler			7282	Monoprice, Inc	1	$ 1	$ 2

Mechanical couplings							
	Requirements	**Manufacturer**	**Model number**	**Supplier**	**Quantity**	**Unit Cost**	**Total cost**
Shaft coupling for torque sensor	L190	Lovejoy				$ -	$ -
Spider	L190	Lovejoy	6408K82	McMaster	1	$ 25	$ 25
Drive shaft		Machine Service Inc		Machine Service, Mark Riddle (313-475-5939)	1	$ 989	$ 1,019

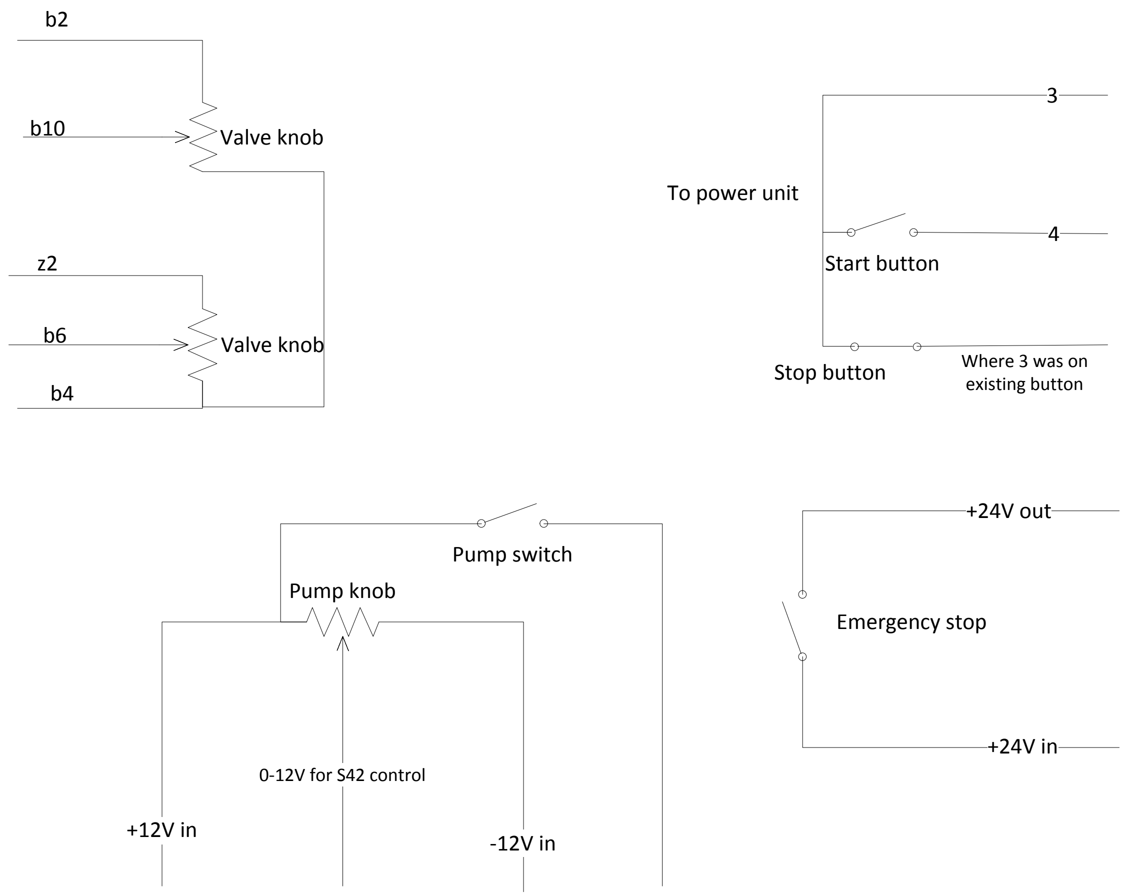
Mounting hardware							
	Requirements	**Manufacturer**	**Model number**	**Supplier**	**Quantity**	**Unit Cost**	**Total cost**
Chassis plate		use existing			1	$ -	$ -
Foot bracket	SAE B foot mount	Hydra-mount	FM-42-B2	Surplus Center	1	$ 40	$ 51
Accumulator base bracket	Holds 10 gallon accumulator	Parker	09633090	MSC	1	$ 112	$ 112
Strut	10 ft 1-5/8"		48024	Fastenal	1	$ 25	$ 31
Strut channel nut	fits 1-5/8" channel			Fastenal	12	$ 1	$ 11
Strut right angle	supports wood		54055330	MSC	4	$ 1	$ 4
Strut right angle	Connects struts together		54055371	MSC	4	$ 5	$ 21
Rubber sheet for damping	1/16th thick		06405302	MSC	2	$ 4	$ 7
Concrete anchors	sleeve type		50306	Fastenal	1	$ 21	$ 21
Expanded metal shield	30"x48"		14334	Discount Steel	1	$ 38	$ 58
Caster wheels for power unit			00995712	MSC	4	$ 4	$ 17

Electrical							
	Requirements	**Manufacturer**	**Model number**	**Supplier**	**Quantity**	**Unit Cost**	**Total cost**
3 phase extension cord	NEMA L15-30 locking extension cord, 10 foot			Custom AV Rack	1	$ 79	$ 91

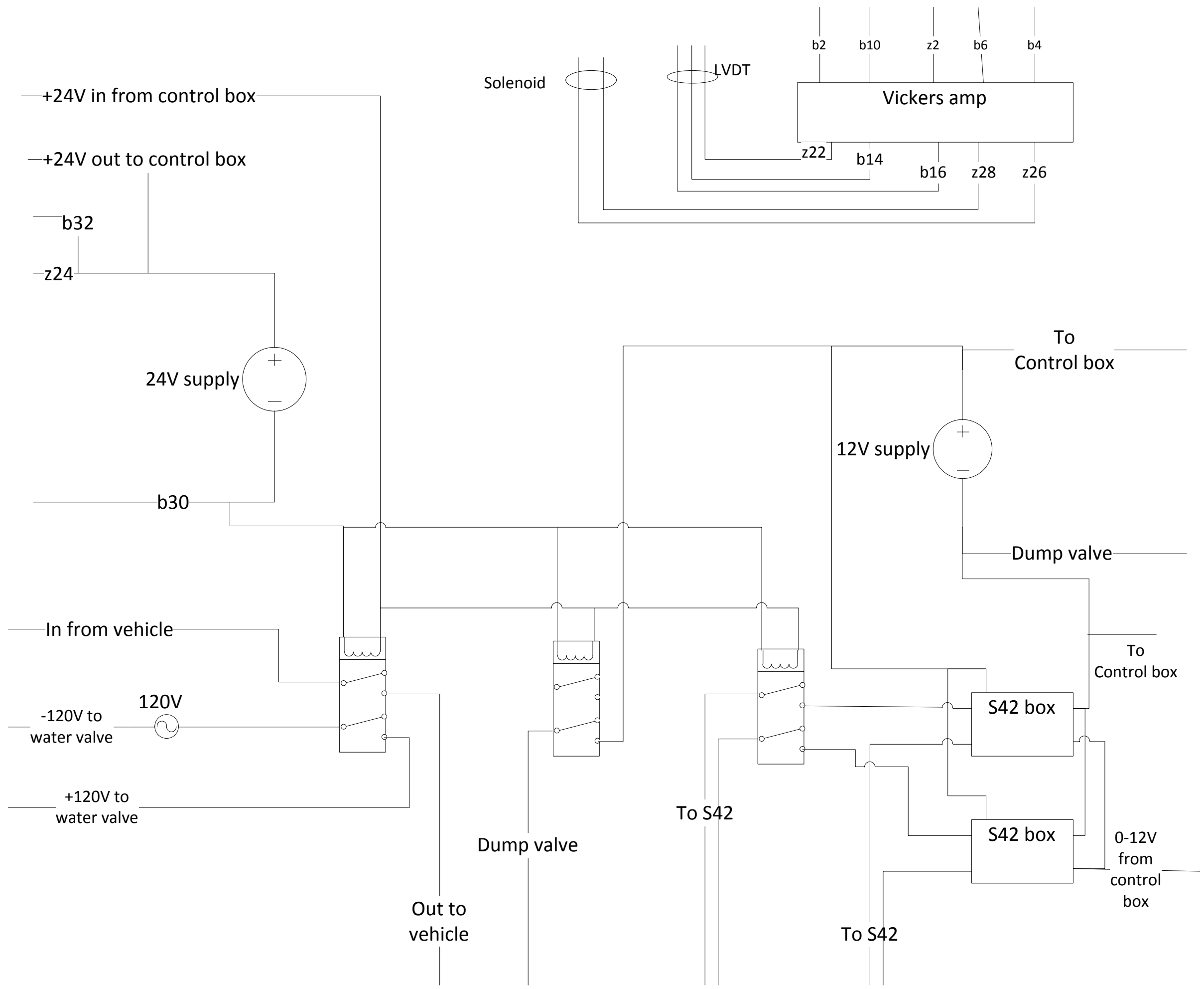
Emergency stop for vehicle, pump displacement, dump valve, water valve	normally open						$ -
Pump start switch	normally open		74K3606	Newark	1	$ 12	$ 12
Pump stop switch	normally closed		74K3605	Newark	1	$ 14	$ 14
Enclosure for buttons			87F2530	Newark	1	$ 8	$ 18
Enclosure for power supplies							$ -
Solder board	for relays						$ -
Relays	0.1" pin spacing, 24V coil		94B9941	Newark	3	$ 2	$ 7
Wire harness tubing			01578467	MSC	1	$ 19	$ 19
Cable sleeve			03080074	MSC	1	$ 23	$ 23
Pump potentiometer	1 kohm		10C0280	Newark	1	$ 1	$ 1
Valve potentiometer	5 kohm		43K0671	Newark	2	$ 1	$ 3
Knobs for potentiometer			10M7580	Newark	3	$ 0	$ 1
Terminal for valve LVDT connection	small enough to fit		34B626	Newark	3	$ 0	$ 0
Enclosure for screw terminal			64051170	MSC	1	$ 13	$ 13
Spare fuse for power unit	6/10A		4CX09	Grainger	3	$ 6	$ 19
Spare fuse	30A		1A698	Grainger	2	$ 2	$ 4
Hazard marking tape	yellow striped		04582821	MSC	1	$ 7	$ 7
					Total		**$ 4,168**

Appendix B: Electrical Schematics

Control Box

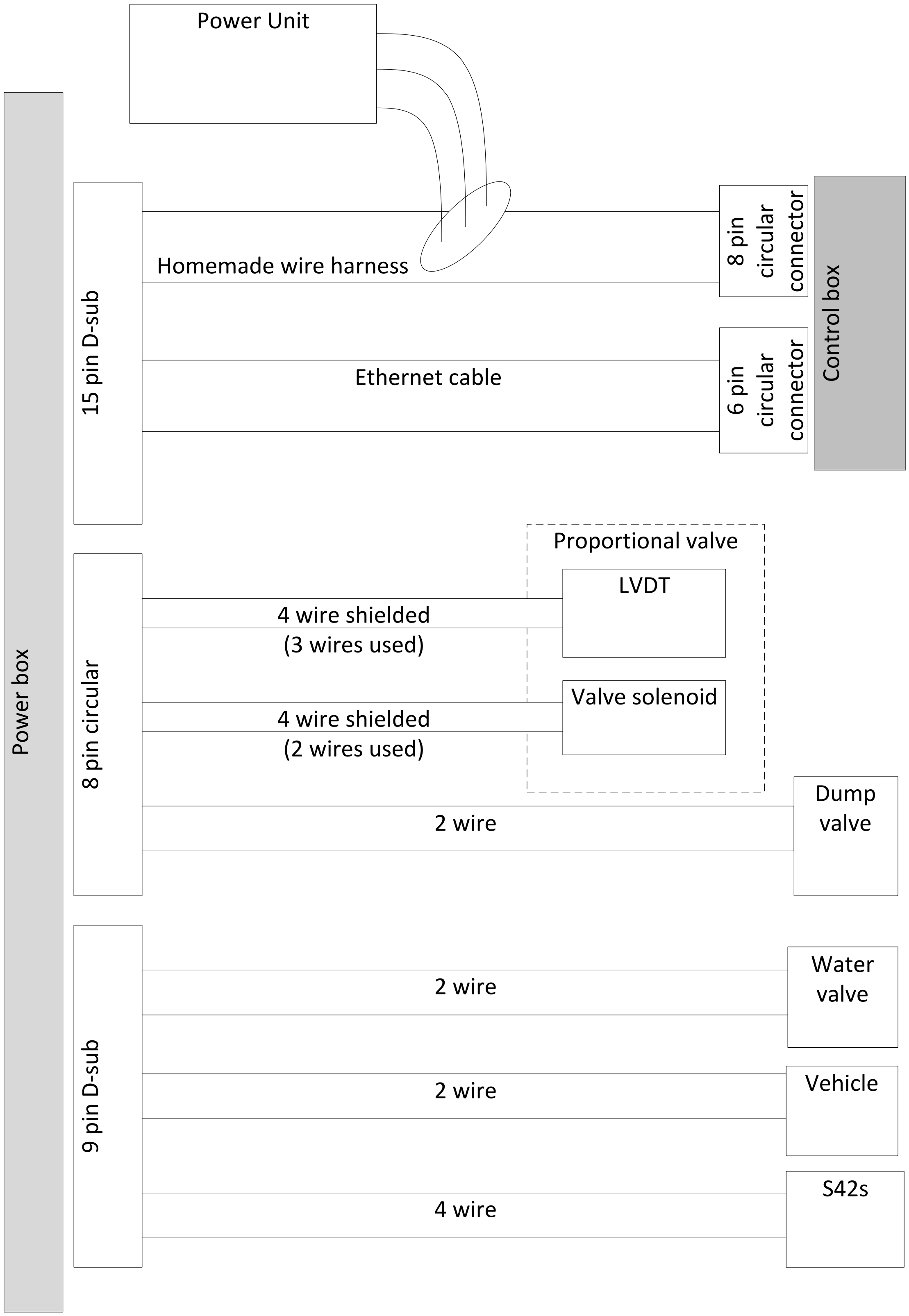


Power Box

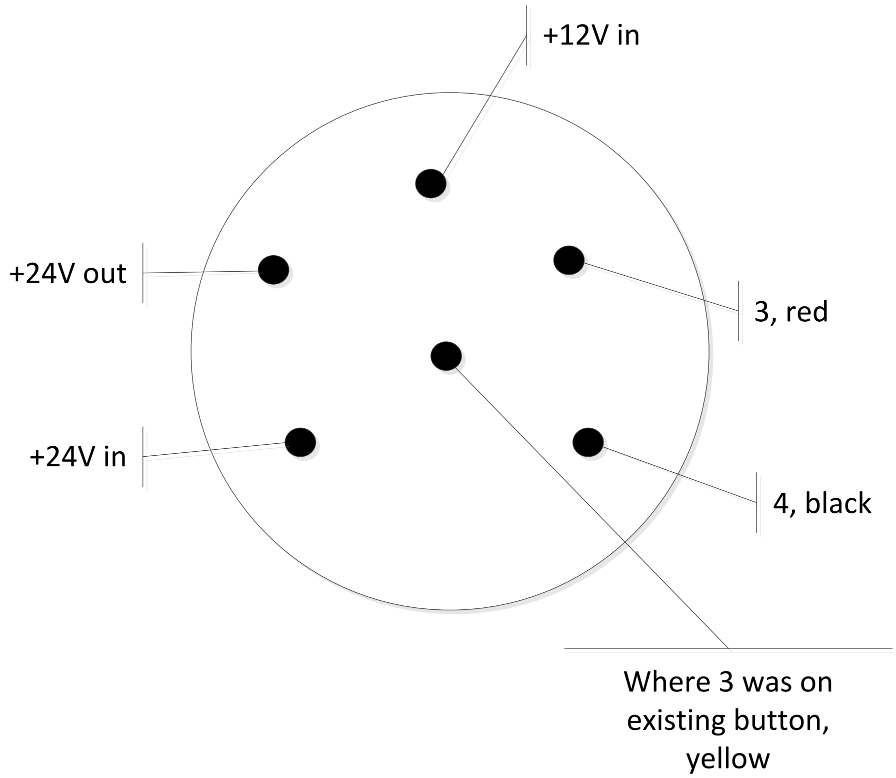


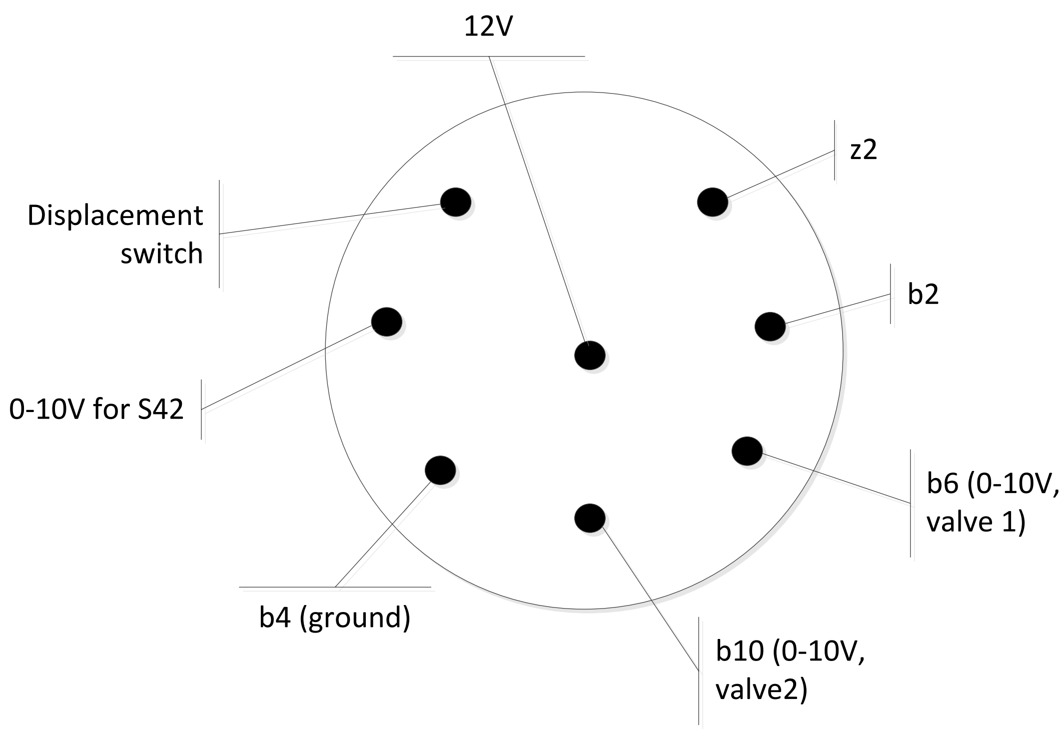
b2
b10
z2
b6
b4
Solenoid
LVDT
Vickers amp
z22
b14
b16
z28
z26
+24V in from control box
+24V out to control box
b32
z24
24V supply
b30
To
Control box
12V supply
Dump valve
In from vehicle
To
Control box
120V
-120V to
water valve
S42 box
+120V to
water valve
To S42
Dump valve
S42 box
0-12V
from
control
box
Out to
vehicle
To S42

Overview of wire harnesses

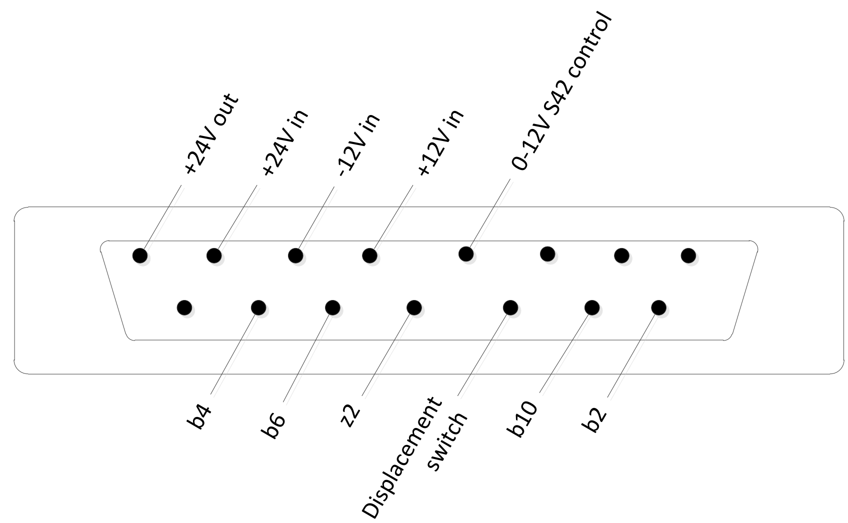


Control box connectors, viewed from front



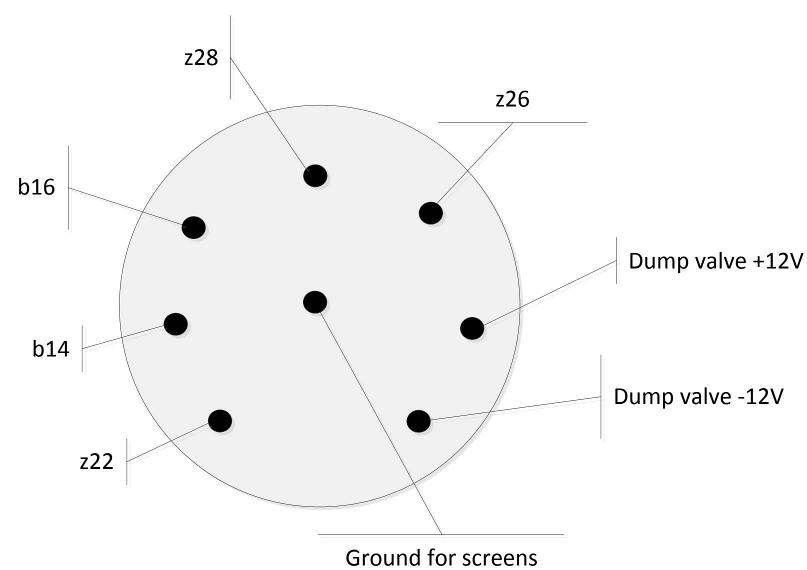


Power box, viewed from front outside of box (names are same as on control box)



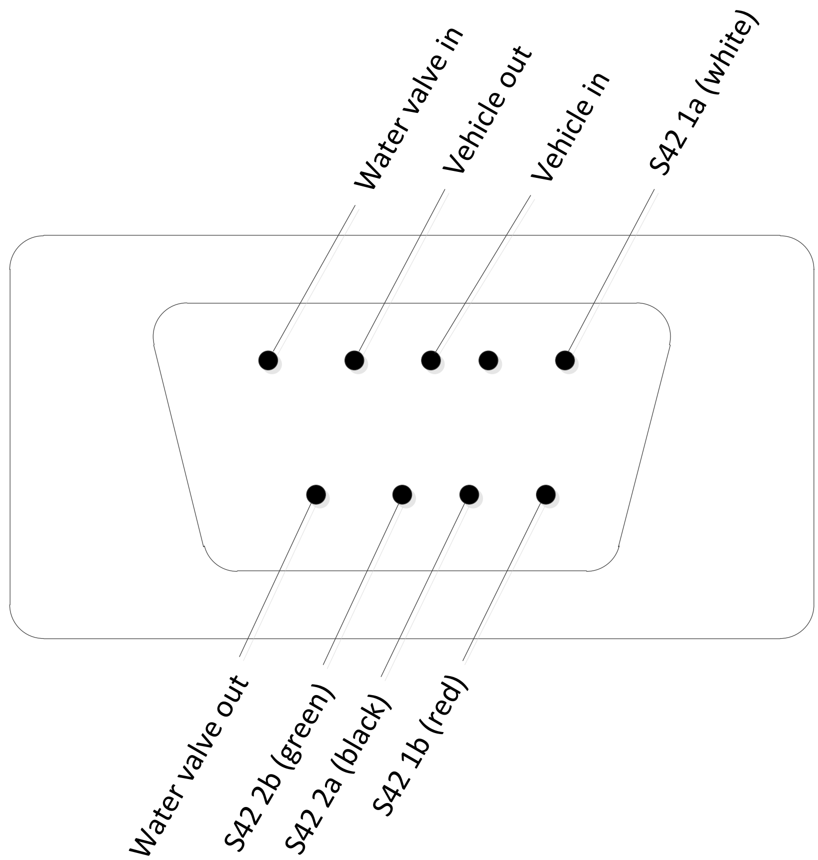
b4: brown
z2: orange/white
b10: blue

b6: blue/white
z11: green
b2: orange



z22: black
b14: green
b16: white
LVDT

z28: black
Z26: red
Proportional valve soleniod


Water valve in
Vehicle out
Vehicle in
S42 1a (white)
Water valve out
S42 2b (green)
S42 2a (black)
S42 1b (red)

Appendix C: Torque Sensor Calibration Instructions

1. Manually rotate the driveshaft to zero torque.
2. Open grey Honeywell in-line amplifier box.
3. The blue potentiometers are zeros. Turn the screws on them such that the torque measurement on the screen shows zero.
4. Flip the top switch on the black aluminum connector box to up.
5. Turn the grey (span fine-control) such that the torque measurement on screen shows 390.55Nm.
6. Flip the switch back down.

Appendix D: MATLAB Code

Vehicle simulation

```
% Mike Olson
% January 2007
%
% Edited Stephen Sedler
% February 2010
% Account for timestep length =/= 1
%
% function m-file wheelpwr.m
% The purpose of this function is to import a drive cycle and vehicle
parameters and then
% calculate the power requirements needed at the wheels as well as other
parameters.

function [cycle] =
drivecycle(drivecycle,N,mass,dragc,fronta,airdens,fo,fs,g,dia,slope)
        cycle.time=cumsum(drivecycle(:,3));
        cycle.roadspeed = drivecycle(:,2);
%Computes average velocity (m/s)
        cycle.acc = [0;diff(drivecycle(:,2))]./drivecycle([1:end],3);
%Computes acceleration at each pt. (m/s^2)
        cycle.accF = mass*cycle.acc;
%Calculates acceleration force (N)
        cycle.drag = 0.5*(dragc*fronta*airdens*cycle.roadspeed.^2);
%Calculates drag force (N)
        %If the vehicle is rolling then compute the rolling resistance
        cycle.roll =
(mass)*(g)*(fo+(3.24*fs*((cycle.roadspeed*2.23693629)/100).^2.5));
%rolling resistance coefficient
        cycle.roll(find(cycle.roadspeed==0))=0;
%Resistance force to rolling (N)
        cycle.slope = (mass*g)*(tan(slope));
%Slope Force
        cycle.force = cycle.accF+cycle.drag+cycle.roll+cycle.slope;
%Ft: Total force seen by engine (N)
        cycle.power = cycle.roadspeed.*cycle.force;
%Power required at the wheels (W)
        cycle.w = (cycle.roadspeed/(dia/2));
%Rotational speed of wheels
        cycle.distance=cumsum(cycle.roadspeed);
%Figures out the total distance traveled
        cycle.torque=cycle.force*dia/2;
```

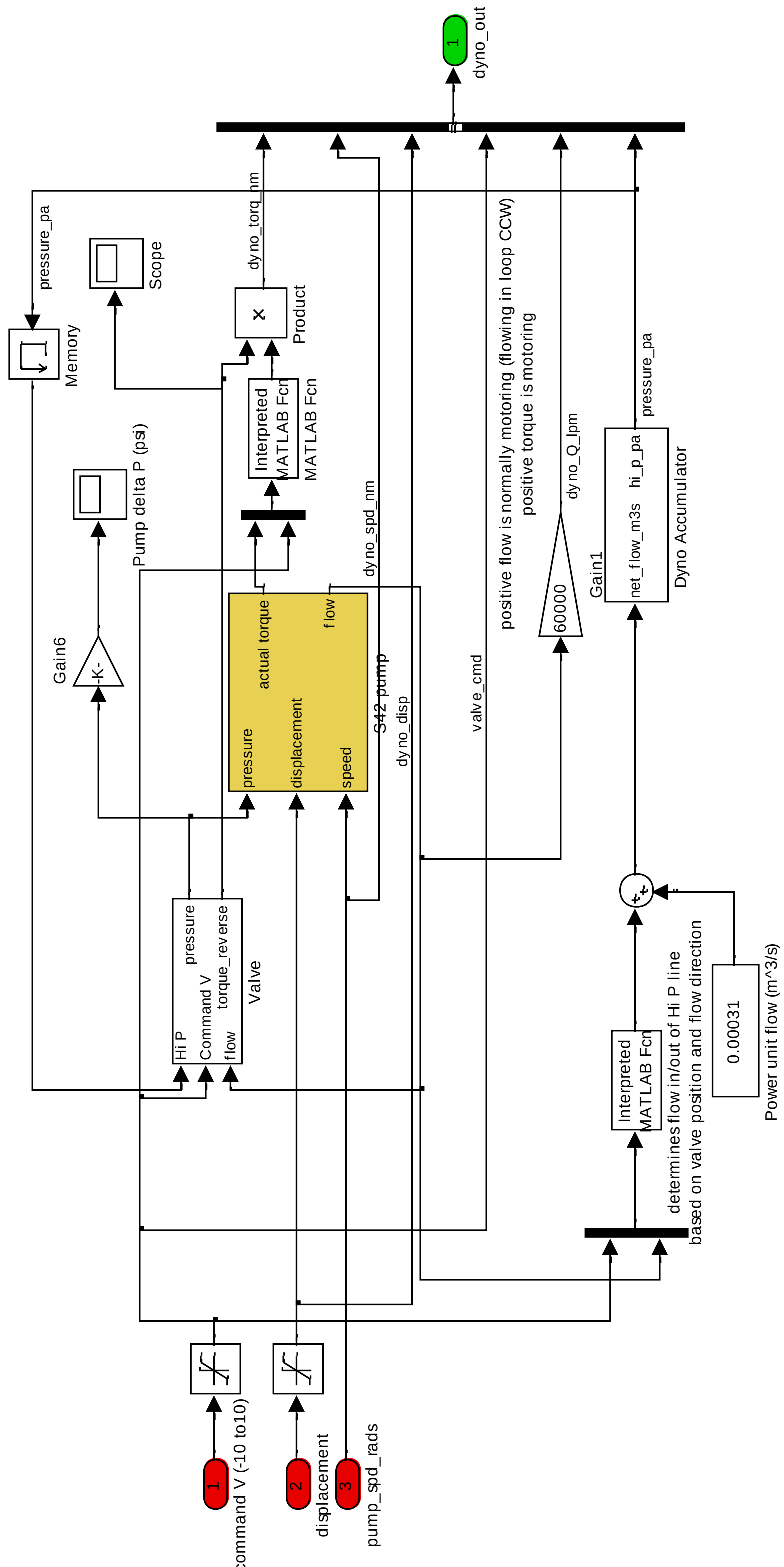
Dynamometer simulation

```
%calculates flow through a tandem center proportional valve

pump_disp=28*2*.9*ones(1873,1); %cc/rev
% pump_disp(280:295)=30;
```

```
%manually changing pump displacement:
% pump_disp(635:641)=30;
% pump_disp(739:743)=10;
% pump_disp(743:750)=35;
pump_disp(300:317)=15;
pump_disp(318:326)=25;
pump_disp(327:330)=35;

pressure_psi=3000; %psi
pressure_pa=pressure_psi*6895;
power_unit_flow=15; %[L/min] = 4GPM

%uncomment for EPA
torque=smooth(EPA.torque)/3.45; %torque at wheels divided by differential
gear ratio
power=EPA.power;
time=EPA.time;
pump_speed=EPA.w*3.45/2/pi*60; %rpm


% %uncomment for highway
% torque=smooth(hway.torque)/3.45; %torque at wheels divided by
differential gear ratio
% power=hway.power;
% time=hway.time;
% pump_speed=hway.w*3.45/2/pi*60; %rpm




pressure_pump=torque*20*pi./pump_disp; %pressure at loading pump [bar]
pressure_pump_psi=pressure_pump*14.5;

Q_rail=zeros(1,length(torque)); %assume no flow when absorbing
for i=1:length(torque)
    if torque(i)<0
        Q_rail(i)=pump_disp(i)*pump_speed(i)/1000; %[L/min] assume all
flow through pump comes from rail
%         Q_rail(i)=power(i)/(pressure_pump(i)*1e5);
    end
end

accum_flow=power_unit_flow-Q_rail;
accum_volume(1)=10; %[L] initialize volume of oil in accumulator; assume
this is maximum capacity of accumulator
%integrates flow to calculate accumulator volume:
for i=1:length(accum_flow)-1
    accum_volume(i+1)=accum_volume(i)+accum_flow(i)/60*(time(i+1)-
time(i));
    if accum_volume(i+1)>accum_volume(1)
        accum_volume(i+1)=accum_volume(1);
    end
end

% close all
```

```
% plot(time,accum_flow)
% title('flow')

ax(1)=subplot(3,1,1);plot(time,accum_volume*.264)
xlabel('time (s)')
ylabel('fluid in accumulator (gal)')
% Q_rail_LPM=Q_rail*1000*60;
% plot(EPA.time,Q_rail_LPM);
% title('flow')

ax(2)=subplot(3,1,2);plot(time,pressure_pump_psi)
title('pressure')
ylabel('pump pressure (psi)')

ax(3)=subplot(3,1,3);plot(time,pump_disp)
title('Pump displacement')
ylabel('pump disp (cc/rev)')
linkaxes([ax(3) ax(2) ax(1)],'x');


% figure
% plot(EPA.time,EPA.torque/3.45)
% title('torque')
% figure
% plot(EPA.time,EPA.power)
% title('power')

%plot(EPA.time,pressure_pump_psi);
% flow=flow*1000*60;
% plot(EPA.time,flow)


% pump_speed=EPA.w*3.45/2/pi*60; %rpm
% flow=EPA.power/pressure_pa; %m^3/s
%(abs(EPA.torque)/3.45)*60/2/pi*pump_disp*pump_vol_eff/1000; %L/min,
positive for flow into high pressure line, negative for flow out of high
pressure line (i.e. motoring)
% pressure_pump=EPA.power./(EPA.w*3.45/2/pi*pump_disp/1e6);
% pressure_pump=pressure_pump/6895;
```

Appendix E: Simulink Dynamometer Physical Model



Appendix F: Dynamometer Operation Instructions

Prepare vehicle

- Ensure gear oil levels approximately cover the bottom gear in each of the three transmission casings.
- Fill and prime fuel if necessary.
- Ensure drip pans are in place under vehicle.
- Ensure fan is operational.
- Ensure exhaust pipe is secure.
- Ensure back end of vehicle is supported by two jack stands.
- Ensure wheels are chocked.
- Ensure driveshaft is secure and set screws are tight.
- Ensure expanded metal shield around driveshaft is in place.

Prepare workspace

- Ensure make up air shutters are open.
- Unlock and open big door to ME 471 to give an escape route in case of fire.
- Clear loose debris from Ranger bed and test cell.

Prepare dynamometer

- Ensure startup reducing valve is disconnected. This valve can be used for startup when the vehicle is disconnected but will allow the dynamometer to turn backwards.
- Check oil temperature periodically during testing and allow to cool if temperature exceeds about 200°F.
- Open cooling water gate valve.

Set up dyno controls

For manual controls

- Plug cables into both connectors in the manual control box.

For automatic controls

- Plug the 8 pin connector into the grey box. Leave the 6 pin connector plugged into the manual control box.
- Start host PC and open MATLAB.
- Start target PC
- Type "xpcnetboot" in the MATLAB prompt on the host PC while the dial on the target PC is spinning.
- Open desired controller in Simulink on host PC.
- Run dyno_initialize script.
- Build the Simulink model on the target PC (press ctrl+B).
- Click the play button to run the model.
- Start in open loop. After vehicle is started, you may switch to closed loop.

Set up vehicle controls

- Plug in Toughbook cable to connect to vehicle.
- Open dSpace ControlDesk.
- Open HHPV_dashboard folder.
- Open dashboard_layout.lay.
- Open plots_layout.lay for plots (optional).
- Start Animation mode (F5) to start the program.

Final preparation

- Turn on power strip connected to dyno power box.
- Turn on main room air circulation fan.
- Turn on exhaust suction fan.

Shutdown instructions

- Press emergency stop buttons on dyno and vehicle. Turn off hydraulic power unit.
- Turn off circulation and exhaust fans.
- Close water gate valve at the end of the day.
- Turn off power strip for dyno power box.
- Close make up air shutters if it is cold outside.
- Close main door to ME 471 at end of day.