DESIGN, IMPLEMENTATION AND TESTING OF A
HYDROGEN-ASSISTED COMBUSTION SYSTEM
FOR A LIGHT-DUTY DIESEL VEHICLE

A Thesis in
Mechanical Engineering

by
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Two experiments were conducted to investigate the effects and feasibility of the addition of gaseous hydrogen to the intake air of two similar, production compression-ignition (CI) engines fueled primarily with a biodiesel fuel blend. In the first experiment, a 1.3L, 52 kW CI engine coupled to an eddy-current dynamometer was tested at eight steady-state operating modes of fixed speed and load. The operating points for steady-state dynamometer testing were chosen based on those most prevalent in a computer-simulated urban drive cycle. The drive cycle was simulated in the Matlab-based Powertrain Systems Analytical Toolkit (PSAT) program using the specifications of the engine to be tested along with those of a 2005 Chevrolet Equinox that has been modified for further testing of a hydrogen-assisted diesel combustion system. During the tests, data were collected on engine operating parameters, fuel consumption, and the concentration of total oxides of nitrogen (NO\textsubscript{X}) in the exhaust stream. Each steady-state point was baseline tested, without the addition of hydrogen to the intake air. Each mode was then tested with hydrogen being fumigated into the intake air at flow rates equivalent to both five and ten percent of the total fuel energy flow into the engine.

For the second experiment, the drivetrain of a 2005 Chevrolet Equinox was removed and replaced with a 1.3L, 67kW CI engine, an improved version of the engine used in the first experiment. The engine was mated to a five-speed automatic transmission utilizing a modified torque converter calibrated for the CI engine and vehicle mass. A gaseous hydrogen storage system was installed in the vehicle along with fueling, fuel delivery,
leak detection, and hydrogen fuel injection systems developed as part of this experiment. The gaseous fuel injector used to deliver hydrogen to the engine intake was calibrated to correlate hydrogen flow rate to duty cycle, and a control strategy was developed to inject the hydrogen as a function of real-time diesel fuel energy flow into the engine. The fuel injection system was controlled by code embedded in a master vehicle controller (MVC) via the vehicle’s controller area network (CAN) bus. After months of extensive road and track testing of the vehicle operating in the hydrogen-assisted combustion mode, chassis dynamometer testing was used to run repeatable drive cycles while logging data. On the dynamometer, the vehicle was operated through a portion of the urban drive cycle simulated for the first experiment, while engine data were logged along with diesel and hydrogen fuel flows, and temperature and NO\textsubscript{X} concentration of the exhaust stream. The drive cycle was repeated, with different hydrogen quantities commanded, and the same data were logged. NO\textsubscript{X} emissions and engine data recorded during the baseline testing were compared with those results obtained while injecting hydrogen during the drive cycles.

The results of the experiments indicated a slight rise in exhaust temperatures, little change in NO\textsubscript{X} concentration, and minimal changes in efficiency as the engine was operated with five and ten percent hydrogen during steady state and cycle testing. An increase in thermal NO\textsubscript{X} was expected due to elevated exhaust temperature, along with slightly lower efficiency due to the increased premixed combustion of the pilot injection and hydrogen mixture before top dead center. This effect was expected to be exacerbated by the decrease in the amount of working fluid in the cylinder, as air was displaced by
aspirated hydrogen. The onboard dual-fuel hydrogen-biodiesel system was shown to be feasible through operation, both during dynamometer and road testing.
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CHAPTER 1

INTRODUCTION AND MOTIVATION

1.1 Motivation

The rising price and inevitable scarcity of petroleum fuels are driving the investigation of alternative motor fuels and methods for implementing them. Concurrently, there is a worldwide demand for reduced vehicle pollutant and greenhouse-gas emissions. While a long-term goal for transportation vehicles is to become free from their dependence on petroleum fuel, alternative fuel substitution is being investigated as a pathway to petroleum fuel independence.

Hydrogen and its infiltration into the future automotive market is a trend that is becoming increasingly more apparent. While fuel cells are often touted as the future of clean, renewable energy, hydrogen has many other uses in advanced transportation. Research has been investigating the benefits of burning hydrogen as a second fuel in dual-fuel combustion processes. The role of hydrogen in an internal combustion engine can be multiple, with emissions reduction and displacement of petroleum fuel being major motivators.

The ever-increasing demand for fuel-efficient vehicles has increased interest in diesel technology, due to the diesel cycle’s inherent high efficiency. Evolution of the diesel
engine has brought about modern machines that are superior in many ways to the gasoline engines that dominate the passenger-vehicle market in the United States. High-pressure common-rail systems, multiple injections per cycle, and increased fuel quality all have helped the modern diesel engine to emit less soot and harmful emissions, run quietly and smoothly, and operate near peak practical efficiency. Many alternative fuels have been experimented with in diesel engines, and hydrogen is one of those fuels.

During the transition from conventional fuels to alternative fuels including hydrogen, many challenges must be addressed. Hydrogen in particular must overcome onboard storage, fueling and delivery systems as hurdles on the way to becoming commonplace as a vehicle fuel. The acceptance of hydrogen as a safe, practical fuel by the public is also a challenge to hydrogen’s implementation.

1.2 This Thesis

In this thesis, the implications of fumigating relatively small amounts of gaseous hydrogen into the intake air of a light-duty, common-rail, direct-injection turbocharged diesel engine fueled primarily by a B20 biodiesel fuel blend (a mixture of 20% biodiesel and 80% ultra-low-sulfur diesel fuel) are explored through experimentation. The purpose of the investigation is two-fold. The first goal of the project is to determine whether hydrogen, fumigated into the intake air of an unmodified diesel engine, can be used to displace some of the diesel fuel consumed by the engine. The effects on the performance of the engine will be explored, along with the practicality of integrating a dual-fuel
hydrogen-injection system into a production vehicle powered by a light-duty diesel engine. The second goal of the project is to determine the effects of the partial diesel fuel substitution on the engine’s exhaust emissions. NOX, a pollutant of primary concern in diesel engines, will be monitored using an onboard vehicle sensor designed for integration in production diesel and lean-burn vehicle engines (Inagaki et al. 1998). Considerations are made for future work on this project including closed-loop emissions monitoring and control using the hydrogen-injection and NOX-sensing systems along with a urea selective catalytic reduction (SCR) catalyst system and additional exhaust sensor instrumentation.
2.1 Hydrogen in Transportation

Two factors motivating the study of advanced transportation technology are environmental responsibility and sustained mobility. While these goals at times may seem at odds, they both underline the need for reduced vehicle emissions and renewable fuels. Collaboration of many in industry has yielded agreement that the use of hydrogen as a transportation fuel can simultaneously address these needs (Eichlseder and Freymann 2003). It is clear that a hydrogen economy has yet to be realized; however, investigation into the use of hydrogen as a fuel in the present is an important step to transitioning from fossil fuels to a hydrogen infrastructure in the future (Lohse-Busch et al. 2006). While hydrogen can be utilized in a fuel cell to produce electricity, it can also be burned in an internal combustion engine (ICE) to yield mechanical power. Either as a standalone fuel or a secondary fuel in a dual-fuel engine, hydrogen can be beneficial when used as an ICE fuel.

2.1.1 Hydrogen as a Primary ICE Fuel

Hydrogen has long been part of our fuel for transportation vehicles, but unfortunately for us it is packaged most conveniently attached to carbon chains in hydrocarbon fuels.
When hydrogen alone is used as a fuel in internal-combustion engines there are many advantages, but also many challenges. Because hydrogen is so physically different than traditional liquid fuels, many considerations must be made when designing or converting an engine to run primarily on gaseous hydrogen.

2.1.1.1 Hydrogen ICE Implementation

For a hydrogen-fueled engine to produce power comparable to a gasoline or diesel-fueled engine of the same displacement, the same amount of fuel energy must be introduced to the combustion chamber, as this is of large influence on the mean effective pressure and therefore power output. In mixture-aspirating engines, where the fuel and air charge is premixed before entering the cylinder, the low volumetric energy density of hydrogen causes a challenge. When premixed, the hydrogen displaces about 30% of the air in the cylinder during stoichiometric operation, and this corresponds to a 30% reduction in indicated mean effective pressure over an equivalent gasoline or diesel engine, or an air-aspirating hydrogen engine with direct injection (Eichlseder and Freymann 2003). While it is true that direct injection of hydrogen is advantageous for maintaining power output, mixture-aspirating engines utilizing port injection are much more readily converted for hydrogen fueling. A spark-ignition (SI), four-valve-per-cylinder, port-fuel-injected engine was modified to burn hydrogen in a project conducted by Ford. The engine of their P2000 hydrogen-engine powered test vehicle, the first production-viable OEM hydrogen-powered ICE vehicle in North America, lacked acceptable acceleration;
however it was expected that boosting of the engine would improve specific power (Szwabowski et al. 2002). Engines that are further modified or constructed specifically to be fueled with hydrogen to utilize direct injection must have fuel injectors capable of injecting hydrogen at the rate necessary for both idle and full power. The test bed used by BMW for their direct-hydrogen-injection study used gaseous fuel injectors with variable supply pressure to meet the conflicting needs of both low and high fueling rates (Eichlseder and Freymann 2003). Three production vehicles with modified SI gasoline engines were modified to run on port-injected hydrogen under cooperation of government and industry as part of the U.S. DOE’s Advanced Vehicle Testing Activity (AVTA). These vehicles were used to test and validate fueling and operation, and logged over 15,000 miles between the three without any major operational problems (Francfort and Karner 2006).

### 2.1.1.2 Hydrogen ICE Emissions

Four significant emissions of ICEs are hydrocarbons (HC), carbon monoxide (CO), NO$_X$, and carbon dioxide (CO$_2$). An ICE burning hydrogen fuel alone removes itself partially from the carbon dioxide debate, since hydrogen contains no carbon. While the majority of hydrogen production currently employs carbon containing fuels, hydrogen produced by renewable energy sources such as solar and wind energy, or nuclear energy is capable of being produced and converted into energy with near zero overall carbon emissions. During hydrogen-fueled engine testing, no significant or traceable carbon dioxide emissions were produced by hydrogen combustion, thus avoiding further contributions to
atmospheric carbon dioxide. Slight carbon emissions are possible due to trace amounts of engine oil on the cylinder wall, however the products of complete lean hydrogen combustion were found to consist of mainly water (Eichlseder and Freymann 2003). NO\textsubscript{X}, however remains a problem for hydrogen-fueled ICE engines. The Ford P2000 study found that when the fuel-air equivalence ratio was increased past 0.5, NO\textsubscript{X} soared, and then fell again somewhat as the equivalence ratio approached 1.0 (Szwabowski et al. 2002). The BMW study of direct hydrogen injection produced similar results, with ‘zero equivalent emissions’ up to an equivalence ratio of 0.45, past which NO\textsubscript{X} increased dramatically; however, it was noted that NO\textsubscript{X} could be lowered by delaying the start of injection (SOI), which cannot be applied to port-fuel-injection engines as an emissions control (Eichlseder and Freymann 2003). Testing performed at Argonne National Laboratory on a supercharged, port-injection, hydrogen-fueled SI V8 engine in a pickup truck maintained low NO\textsubscript{X} emissions by throttled, lean-burn operation with a constant equivalence ratio of 0.45. The supercharger was necessary to counter the power loss due to lean-burn operation, and it was concluded that the hydrogen-fueled engine had very low emissions overall when operated in the lean regime, outside of which NO\textsubscript{X} was the only significant pollutant (Lohse-Busch et al. 2006). The majority of NO\textsubscript{X} formation in a diesel engine can be contributed to the thermal NO mechanism, in which the excess oxygen encountered in lean operation reacts with nitrogen present in the cylinder at high temperatures. The reaction to form NO\textsubscript{X} is endothermic, thus the increased combustion temperature associated with increased F/A equivalence ratio below stoichiometric explains the increase in NO\textsubscript{X} as F/A equivalence ratio increases. The NO\textsubscript{X} peak tapers
off as the mixture approaches stoichiometric, since the amount of oxygen available to react with nitrogen to produce NO\textsubscript{X} is limited (Heywood 1988).

2.1.1.3 Hydrogen ICE Efficiency

Equivalence ratio not only was found to affect emissions, but it was found to have an impact on efficiency as well. While lean operation produces less power than stoichiometric, the lean combustion tends to yield more efficient operation. The Argonne National Laboratory study with the hydrogen-ICE-powered pickup truck found that the leaner the mixture, the higher the efficiency of the engine operation at fixed points. Contrary to what was expected, urban test cycles performed on a dynamometer showed lower overall fuel efficiency during very lean operation test cycles. The lack of torque forced the engine to reach higher speeds to reach the cycle power requirements, to compensate for the loss of torque from very lean operation (Lohse-Busch et al. 2006). The BMW study did not operate test cycles. However, during steady-state testing it was found that late injection of the hydrogen through their direct-injection system would increase fuel economy; the concentration of unburned hydrogen in the exhaust also increased with retarded SOI (Eichlseder and Freymann. 2003). The possibility of stable lean operation due to the higher flame speed and lean operation allow for a fuel efficiency increase over similar gasoline-powered engines (Lohse-Busch et al. 2006). A fuel economy comparison between the hydrogen-fueled Ford P2000 and its gasoline-fueled equivalent showed an improvement of 4.9 miles per gallon gasoline equivalent
(MPGGE) in the hydrogen-powered vehicle over the 27.5 mpg achieved with the gasoline-fueled vehicle in a city cycle (Szwabowski et al. 2002).

### 2.1.2 Hydrogen Assisted Combustion

Hydrogen can offset some of the demand for hydrocarbon fuel by being combusted along with gasoline, diesel, or natural gas in an internal-combustion engine. These systems, known as bifuel or dual-fuel systems, can either use very small amounts of hydrogen to alter combustion or else use a large amount of hydrogen as the principle source of energy in the combustion chamber. The operation of these systems and engines has been investigated for several types of hydrogen-assisted combustion (HAC).

#### 2.1.2.1 Hydrogen Assisted Spark Ignition Engines

Two main types of spark-ignition engines utilizing hydrogen as a secondary fuel are compressed natural gas (CNG) and gasoline engines. HCNG is a blend of compressed natural gas and hydrogen, where the amount of hydrogen can be varied, depending on the system that will burn the fuel. One factor that makes HCNG a simple kind of HAC is that the two fuels are mixed before being stored in the vehicle, eliminating the need for separate fueling systems (Francfort and Karner 2006). Typically, HCNG is employed not only to reduce the amount of CNG fuel used, but also to reduce emissions. NO\textsubscript{X} emissions are reduced compared to a CNG only vehicle by operating the engine in the
lean combustion mode; however, the engine cannot be run below what is known at its lean limit, or the point where substantial amounts of the fuel are not burned (Burke et al. 2005). The addition of hydrogen to the CNG enhances this lean limit, allowing for leaner operation and thus lowered emissions. Power is reduced proportionally to how lean the engine is operated, so to run in the lean range and maintain power, either a larger engine, forced induction, or a combination of the two may be used (Burke et al. 2005). Lean operation may also be enabled in advanced gasoline engines by the addition of hydrogen to the mixture; however, ignition timing is important for optimization (Ma et al. 2003).

2.1.2.2 Hydrogen Assisted Compression Ignition Engines

Particulate matter (PM), oxides of nitrogen (NO\textsubscript{X}), carbon monoxide (CO), and hydrocarbons (HC) are all regulated vehicle emissions. Carbon dioxide (CO\textsubscript{2}) is under consideration to be limited in the future, and all of these emissions have been found to be harmful to the environment; thus, their reduction is desirable. Diesel-hydrogen dual-fuel combustion has been reported to yield decreases in all of these emissions, when compared to operation of the compression-ignition engine burning diesel fuel alone.

In one experiment, small amounts of hydrogen peroxide (H\textsubscript{2}O\textsubscript{2}) were fumigated into the intake of the engine using an electronic injector. The resultant reduced ignition delay was believed to cause a decrease in NO\textsubscript{X} in the test engine proportional to the amount of H\textsubscript{2}O\textsubscript{2} fumigated (Gjirja et al. 2000). In another experiment by Tomita et al., (2001), hydrogen was mixed with the intake air of a direct-injection diesel engine. SOI timing of
the diesel fuel was varied across a wide range, holding the overall equivalence ratio equal with and without the addition of hydrogen. Their tests showed very low NO\textsubscript{X} emissions when SOI was advanced to or beyond 40° before top-dead-center (BTDC), with NO\textsubscript{X} emissions rising with later injection timing. This was reasoned to have occurred because of the thorough mixing of the hydrogen/air mixture and the diesel fuel before ignition. It was also observed that CO\textsubscript{2} decreased proportionally to the amount of hydrogen substituted for diesel fuel, due to less carbon available in the reactants. The hydrogen was believed to cause an increased ignition delay due to the large mole fraction of hydrogen in the air displacing oxygen. Efficiency of the engine was affected with the addition of hydrogen, such that typically the brake thermal efficiency of the engine was slightly reduced (Tomita et al. 2001). A study of lean premixed combustion by Jacobs et al. (2005) in a conventional diesel engine fueled with diesel fuel showed that NO\textsubscript{X} was reduced as timing was retarded. This was attributed to the reduced flame temperatures caused by retarded injection (Jacobs et al. 2005).

An investigation into the effects of gaseous fuels burned with diesel fuel in a direct-injection engine conducted by Varde and Varde (1984) used hydrogen to supplement the diesel fuel up to 15% of the total fuel energy in a naturally-aspirated diesel engine. At light loads, the addition of hydrogen reduced soot formation up to 50% over the conventional diesel-fuel mode due to the high equivalence ratio. In their study, NO\textsubscript{X} increased as the ratio of H:C was increased by partial fueling with hydrogen, with greater hydrogen amounts increasing NO\textsubscript{X} proportionally (Varde and Varde 1984). Tsolakis and Megaritus (2004) used a fuel reformer to produce hydrogen-rich gas to be introduced into
the combustion process by way of reformed exhaust-gas recirculation (REGR). Their findings showed that this method of achieving partially premixed charge compression ignition (PCCI) yielded the potential for reduced NO\textsubscript{X} and PM emissions, and enhanced efficiency.

Homogeneous charge compression ignition (HCCI) has been attempted with hydrogen, and in the regime where it is feasible, an efficiency gain over spark-ignited hydrogen combustion engines was observed (Stenlåås et al. 2004). Hydrogen has also been experimented with in the biofuels sector; it was observed in an experiment by Senthil Kumar et al. (2002) that hydrogen addition to the intake air of a primarily vegetable-oil fueled diesel engine can reduce smoke and increase thermal efficiency, addressing two inherent problems with using vegetable oils as fuel. In their study hydrogen was added as a mass fraction of the total fuel, and increasing the ‘mass share’ of hydrogen had the following effects: decreased CO, increased NO, increased ignition delay, decreased HC, and both increased and decreased efficiency, depending on the amount of hydrogen and type of base fuel used (Senthil Kumar et al. 2002).

2.1.3 Onboard Hydrogen Storage and Fueling

For hydrogen to be a useful vehicle fuel, it must be readily stored or produced from some other fuel stored on board. According to Aceves and Berry (2004), three main ways to store hydrogen are: as a compressed gas, as a cryogenic liquid, or in a metal hydride.
Hydrogen has much less energy per unit volume than typical fuels like gasoline and diesel fuel. Figure 2-1 shows the relative energy storage potential per liter of fuel, based on the fuel’s lower heating value (LHV).

![Figure 2-1: Volumetric energy density based on LHV of each fuel (adapted from Aceves and Berry 2004).](image)

The current state-of-the-art for compressed hydrogen storage is between 35 and 69 MPa (5-10 kPSI). The compressed gas must reside in a pressure vessel, and using the current state-of-the-art for hydrogen pressure vessels, a large tank usually is placed under the vehicle or in the rear. This method typically yields unsatisfactory range for the vehicle, limits clearance or storage space, and is expensive. Metal hydride hydrogen storage
requires very heavy structures. This additional mass can limit the value added by using hydrogen as a fuel in the first place, such as decreased fuel economy, increased emissions, and loss of performance. The low-pressure cryogenic hydrogen storage method suffers from fuel loss during periods of inactivity due to the fuel boiling off. One proposed solution to some of these problems is to combine pressure-vessel storage of hydrogen with cryogenic insulation of the vessel, allowing lower evaporative losses and the ability to operate on compressed fuel or liquid fuel when longer range is desired or necessary (Aceves and Berry 2004).

While compressed fuel is one method of storage, and is the most common in current prototype vehicles, it has disadvantages which are discussed in a trade off study by Berry (1996). The costs to compress the hydrogen to useful working pressures are significant. The transportation, distribution, and utilization of hydrogen as a transportation fuel also pose many questions. Cost analyses have been thoroughly conducted, and it is important to note that the expected costs of a hydrogen infrastructure will likely be seen as less intimidating as the cost of fossil fuels rise, increasing the need for a hydrogen alternative (Berry 1996).

**2.1.4 Hydrogen Vehicle Safety**

Several additional safety items must be considered on vehicles that use hydrogen as a motor fuel. Jacob (2005) notes that “A catastrophic failure in any hydrogen project could irreparably damage the entire transition strategy.” He notes that hydrogen is safer than
many fuels, in that it has a higher autoignition temperature in air, is less toxic to humans, and can disperse faster than other fuels; in fact, it disperses faster than gasoline due to its 12-times-higher gas-phase diffusion coefficient. Hydrogen, however, is easily ignited through a wide range of concentrations in air, from 4.1% to 74% by volume, and therefore concentration must be monitored where ignition could cause damage (Jacob 2005).

Because of the small molecular size of hydrogen, special precautions must be made against leakage in the pressurized storage and delivery systems. Szwabowski et al. (2002) note that the construction of the Ford P2000 H₂ ICE vehicle’s fuel system features several safety precautions including in-tank isolation solenoids with internal check valves and thermal pressure relief device vent port (PRD). The PRD on the Ford vented through the roof to disperse hydrogen as quickly as possible from the vehicle area. Their storage system senses tank pressure and temperature via a temperature probe and pressure transducer. The hydrogen fuel, after leaving the storage tanks, is regulated to down to 75 PSI, and a downstream pressure relief valve set at 125 PSI would prevent rail pressures from becoming dangerously high. Fueling-safety considerations included ground connections between vehicle and fueling station, and quarter-turn lockdown valves to isolate the tank during fueling. Stainless-steel tubing was used for the fuel delivery system on the vehicle. In case of a leak, hydrogen sensors were installed in the engine compartment, passenger compartment, and trunk, and they were set to be triggered at 15%, 25%, and 40% of the lower flammability limit of hydrogen in air (Szwabowski et al. 2002).
A study carried out by Maeda et al. (2006) of the Japan Automobile Research Institute tested ignition of hydrogen inside of a vehicle, and rated the safety of the ignition for the vehicle and persons around the outside of the vehicle. Leaked hydrogen in the wheel well of the vehicle increased hydrogen concentrations in the front engine compartment to a maximum of 23.7% by volume. The mixture was then ignited, and the damage was studied to the vehicle and to a peripheral area to determine injury potential to people in that area. The results of the ignition analysis showed “almost no impact” on people in the vicinity of the vehicle or to the vehicle itself (Maeda et al. 2006).

2.1.5 Hydrogen Injectors

For complete or partial hydrogen-air mixture-aspirating engines, a controlled means of introducing hydrogen to the intake air is required. Typically, electronic injectors are used for SI engines to deliver fuel, and they can be calibrated to deliver a known fuel quantity (Stone and Ball 2004). For use with hydrogen, the injector often must be larger to accommodate the higher flow rate due to hydrogen’s low volumetric energy density. Natural-gas fuel injectors of the sonic flow, pulse-width modulated type that can be obtained commercially have proven to be sufficient for hydrogen injection by testing performed by Heffel and Nabeck (1998). The study investigated leakage and found no significant leakage of hydrogen from the unmodified natural-gas injectors. Also, they confirmed that linear calibration was possible (Heffel and Nabeck 1998). To maintain a linear relationship between duty cycle and flow, the injectors must be operated in the
linear range, which is at duty cycles with a period longer than the response time of the injector; this can be confirmed during calibration and testing (Barkhimer and Wong 1995).

2.1.6 Onboard NO\textsubscript{X} Emissions Measurement Systems

Because NO\textsubscript{X} was found to increase in some cases of hydrogen assisted combustion, it is beneficial to be able to sense the concentration of NO\textsubscript{X} onboard the vehicle, where that information can be used to adjust the fueling calibration. The NGK Spark Plug Company LTD has developed a NO\textsubscript{X} meter intended for vehicle integration, and operation on board. Inagaki et al. (1998) discuss NGK’s meter, which utilizes a thick-film ZrO\textsubscript{2} pumping cell. The diagram of this cell is shown in Figure 2-2.
With ambient exhaust flowing into the first cell, voltage applied to the electrodes in the cell causes oxygen ions to be pumped through the cavity, maintaining a nearly constant stoichiometric value in the cavity, independent of the stoichiometry of the exhaust gas. The partial pressure of the oxygen in the first cell is indicated by the current flowing between the electrodes. This current correlates to the concentration of oxygen in the exhaust. The gas from the first cell, devoid of oxygen, flows through the narrow passage between the two cells. In this second cell, the NO\textsubscript{X} is dissociated into N\textsubscript{2} and O\textsubscript{2}, and the measured current between electrodes corresponds to the amount of NO\textsubscript{X} gas dissociated. This single element is thus able to measure NO\textsubscript{X} and O\textsubscript{2} concentrations in the exhaust (Inagaki et al. 1998). This sensor has been integrated with a controller area network (CAN) transceiver and control package using the CANOPEN protocol by Engine Control
and Monitoring of Los Altos, California. ECM publishes a NO\textsubscript{x} accuracy of +/- 30 PPM or 3\%, whichever is greater (ECM 2006).
CHAPTER 3
ENGINE DYNAMOMETER TESTING

A 1.3L, 53kW turbocharged direct-injection diesel engine was baseline tested on an engine dynamometer. The methods used to test the effects of hydrogen aspiration on the diesel engine, along with the results of the testing, are introduced in this chapter.

3.1 Experimental Setup

The computer modeling used to calculate the eight test modes, test-stand setup, instrumentation and control, and the hydrogen-aspiration method for steady-state testing are discussed in this section.

3.1.1 Computer Modeling

Rather than choosing arbitrary fixed speed/load points at which to test hydrogen injection on the diesel engine, or use the AVL 8-mode test procedure for heavy-duty engines, eight unique fixed-speed-and-load modes were calculated to correspond with those most frequented in a simulated drive cycle (Dieselnet 2008). To simulate the speed and load of an engine powering a vehicle through a prescribed sequence of acceleration and deceleration, the following parameters were defined: A cycle of predetermined speed versus time, the specifications of the vehicle platform and driveline to be powered
through the drive cycle, and the performance map of the engine that will power the vehicle.

3.1.1.1 Drive Cycle

The drive cycle chosen for the simulation was the Urban Dynamometer Driving Schedule (UDDS) cycle, a 1372-second test procedure designed to reflect typical speeds and accelerations in city traffic conditions. The UDDS cycle, shown in Figure 3-1, has many stops including vehicle idling time to indicate stopped traffic and red lights. The average speed over the cycle is 19.4 miles/hour, with 56.6 miles per hour being the fastest speed reached
3.1.1.2 Vehicle Platform and Drivetrain

A modified 2005 Chevrolet Equinox was chosen as the simulated vehicle platform. This vehicle was chosen due to its use in the Challenge X program at Penn State University, and its role as the test vehicle used in the second experiment of this research project (Chapter 4). Challenge X is a program sponsored by the General Motors Corporation and the United States Department of Energy, along with many corporate sponsors. The goals

Figure 3-1: Urban Dynamometer Driving Schedule (UDDS) cycle.
of the Challenge X competition are for teams of engineering students from North American universities to improve the fuel economy and lower vehicle emissions of the Chevrolet Equinox, a ‘Crossover SUV’, while maintaining a safe, marketable vehicle. As part of the competition, along with the research being conducted for this thesis, a 2005 Chevrolet Equinox was converted to a Hybrid Electric Vehicle (HEV) powered by a compression-ignition engine utilizing a custom biodiesel-hydrogen dual-fuel system with the latter of these modifications being relevant to this thesis. All testing for this experiment was conducted using the engine only to power the vehicle; the electric drivetrain was disabled during testing. The vehicle, as prepared for and used in this experiment, is shown in Figure 3-2.
Many physical properties of the vehicle chassis and driveline affect the instantaneous road-load power required to keep the vehicle in sync with the drive-cycle speed trace. These parameters were programmed into the simulation software to reflect those of the Equinox as closely as possible. The transmission, connected to the engine along with the torque converter, and the final drive ratio are items of principal importance, as they determine the relationship between the vehicle’s ground speed and engine speed.

### 3.1.1.3 Engine

The engine parameters programmed into the model mirrored those of the engine to be used for dynamometer testing. The model is relatively simple, and only calculates the engine speed and load at each point throughout the drive cycle; the only input required is a map of maximum engine torque versus engine speed. Figure 3-3 shows the peak torque versus engine speed curve used for the model.
3.1.1.4 Simulation

A model of the vehicle was programmed into PSAT (Powertrain Systems Analytical Toolkit), a software program developed by the Argonne National Laboratory used to model and analyze the powertrains of advanced vehicles (Argonne 2007). While PSAT has the capability to model advanced vehicles with energy storage and hybrid systems, it was used in this instance to simply predict the engine’s speed and load throughout the UDDS drive cycle. The instantaneous road-load power requirement, $P_v$ in Eq. 3.1, is typically broken up into four separate components: Rolling Resistance ($R_r$), Grade Resistance ($R_g$), Aerodynamic Resistance ($R_a$) and Inertia Forces ($\gamma_m$) due to acceleration.
\[ P_v = \nu (R_r + R_g + R_a + \gamma_m a) \]  

The sum of these power requirements, scaled by a driveline efficiency factor, is the power that must be supplied by the engine. The engine torque can be calculated from power when the engine speed is known. Engine speed as a function of road speed is determined by a combination of tire size and transaxle final drive ratio, which are fixed, along with transmission gear ratio and torque converter speed ratio, which are both variable. Once the necessary vehicle parameters are programmed into PSAT and a drive cycle is selected, the simulation was run. The variable automatic transmission gear ratio is controlled by the controller block in PSAT. The controller block uses an efficiency algorithm based on the efficiency map of the engine, and often this is the lowest engine speed possible to produce the necessary power. The torque converter speed ratio is determined by a model of the torque converter’s fluid coupling. Once the characteristics of the torque converter are programmed into PSAT, the intricacies of the model are transparent to the user. The results of the simulation provided traces of engine speed and torque, which are plotted along with power in Figure 3-4.
Figure 3-4: Simulated engine power, speed, and torque for the 1.3L diesel engine.
3.1.1.5 Selection of Test Points

To select operating points for testing, the map of engine speed and torque was divided into 13 bins for torque and nine bins for speed. The resulting matrix allowed for each possible operating point to be placed into one of 117 groups. The points at which the engine operated were tallied throughout the cycle at 0.1 second intervals. The eight groups that were operated in the most frequently were selected for testing. The centroid of each speed and load group was used to represent the group, and was chosen as one of the eight test modes. Figure 3-5 shows that idling and deceleration are frequent in the cycle; these are represented by low speed and load. Each of the eight points, with their respective frequency, is shown in Table 3-1. These are the eight points that were chosen to be tested as significant operating points for the Penn State Challenge X Equinox as simulated through the UDDS cycle.
Figure 3-5: Histogram of operating point frequency.

Table 3-1: Distribution of operating points for the diesel engine.

<table>
<thead>
<tr>
<th>Point</th>
<th>Percent of Cycle</th>
<th>Torque (NM)</th>
<th>Speed (RPM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>26.0</td>
<td>15</td>
<td>1000</td>
</tr>
<tr>
<td>2</td>
<td>7.5</td>
<td>60</td>
<td>1750</td>
</tr>
<tr>
<td>3</td>
<td>6.2</td>
<td>45</td>
<td>1750</td>
</tr>
<tr>
<td>4</td>
<td>5.7</td>
<td>75</td>
<td>1750</td>
</tr>
<tr>
<td>5</td>
<td>4.7</td>
<td>30</td>
<td>1250</td>
</tr>
<tr>
<td>6</td>
<td>4.6</td>
<td>45</td>
<td>1250</td>
</tr>
<tr>
<td>7</td>
<td>4.1</td>
<td>15</td>
<td>1250</td>
</tr>
<tr>
<td>8</td>
<td>3.6</td>
<td>30</td>
<td>1750</td>
</tr>
</tbody>
</table>
3.1.2 Test Stand Setup

A GM 53kW, 1.3L, turbocharged, direct-injection diesel engine originating from the European passenger car market was set up on an engine dynamometer test stand to be used as a test apparatus to determine the effects of hydrogen induction on a production diesel engine. All engine control was performed by the engine manufacturer’s engine control unit (ECU). A cutaway image of the engine tested is shown in Figure 3-6. The mounting and setup, and ancillary equipment of the engine and dynamometer are discussed in this section, along with the control hardware and software developed.

![Cutaway image of the GM 1.3L diesel engine (Challenge X 2006).](image-url)
### 3.1.2.1 Engine Mounting and Setup

The engine used was a small direct-injection diesel that is used in small European vehicles. Table 3-2 lists key engine characteristics. The engine was designed for mounting in a vehicle engine bay, so adaptation for stationary installation was necessary. An aluminum adapter plate was machined to mount the bellhousing of the engine to mounting feet bolted to the test bed through isolation mounts. The opposite side of the engine was adapted at its mount point, bolting to additional mounting feet through a vibration isolator. Once the engine was securely mounted to the test bed, it was coupled to an eddy-current dynamometer as shown in Figure 3-7. A custom damped driveshaft was manufactured and bolted between the production dual-mass flywheel by way of an adapter plate and the dynamometer rotor.

Table 3-2: GM diesel engine characteristics.

<table>
<thead>
<tr>
<th>Characteristics</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>52 kW</td>
</tr>
<tr>
<td>Torque</td>
<td>180 Nm</td>
</tr>
<tr>
<td>Cylinders</td>
<td>4</td>
</tr>
<tr>
<td>Displacement</td>
<td>1250 cc</td>
</tr>
<tr>
<td>Bore</td>
<td>69.6 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>82 mm</td>
</tr>
<tr>
<td>Comp. Ratio</td>
<td>18:1</td>
</tr>
<tr>
<td>Valves</td>
<td>4 /Cyl.</td>
</tr>
<tr>
<td>Rail Pressure</td>
<td>1400 Bar</td>
</tr>
</tbody>
</table>
3.1.2.2 Engine Ancillaries

Because the engine was to be operated while stationary, an oversized aluminum radiator was plumbed to the engine and mounted at the end of the test stand with a large electric cooling fan and shroud. For intake and exhaust plumbing, 1.5 inch diameter tubing was used. The intake air was filtered through an appropriately sized cone-type filter without the use of an air box, as the production airbox was unavailable. An aluminum intercooler was custom built for the application, and was cooled by two electric fans. A custom
wiring harness and calibrated ECU were provided for dynamometer testing by GM Europe. This allowed the engine to be run without a key; the immobilizer unit, used as a theft-prevention system on production vehicles, was circumvented by permanently mounting the transponder chip normally found in the ignition key to the sensor attached to the harness. The engine alternator and air-conditioning compressor were both disconnected, ensuring that they could apply no transient load. Because the alternator was disconnected, a large DC power supply along with a large storage battery powered the ECU, low pressure fuel pump and fans. Diesel fuel was supplied from a marine-type fuel tank mounted on an electronic balance, and pressurized using an electronic fuel pump. The fuel was filtered through a combination diesel fuel filter and water separator.

3.1.2.3 Dynamometer Setup

A 450 HP eddy-current dynamometer was used to load the engine and absorb the power output by the diesel engine. The dynamometer was cooled by water stored in a large inground tank and pumped through the dynamometer by an electric pump. The dynamometer torque was adjusted by applying current across the stator windings by means of an adjustable DC power supply. The power supply was configured to a constant-current mode to minimize the current drift and subsequent load drift encountered in constant-voltage mode due to transient coil heating as the cooling water temperature rose during operation.
3.1.3 Engine Instrumentation and Control

Data were collected on engine operating parameters, dynamometer parameters, and fuel mass and flow. The hardware and software methods used to collect, display, and log this data are discussed in this section.

3.1.3.1 Instrumentation and Control Hardware

To perform testing with the engine, and to analyze the data collected during testing, it was determined that engine speed, torque, intake air temperature, coolant temperature, manifold pressure, mass air flow, exhaust temperature, diesel fuel mass, and hydrogen fuel flow would need to be monitored and logged. In order to maintain a safe, effective test-stand system, the engine dynamometer cooling water temperature, and hydrogen concentration in the lab were monitored as well. A National Instruments PXI Chassis computer, shown in Figure 3-8, was used as the test-stand control, monitoring, and data logging system for all of the sensors having electronic outputs. Only hydrogen flow rate and hydrogen concentration in the lab air were unable to be routed into the control and monitoring computer, and were thus monitored and recorded manually.
Much of the instrumentation necessary to collect engine parameters was already in place on the engine for use with the on-board diagnostic and closed-loop control systems programmed into the ECU. Because the transfer functions of the transducers and sensors were not known, communication with the ECU was performed periodically to request the values of the engine speed, intake air temperature, coolant temperature, manifold boost pressure, and mass air flow as read by the ECU.

Additional transducers and sensors were integrated into the test stand to read engine speed, exhaust temperature, intake manifold air temperature, dynamometer cooling water temperature, engine torque, and fuel mass. Although engine speed is read from the ECU,
an additional sensor was used to monitor speed in case of ECU communications failure or monitoring station crash, ensuring the engine speed could be determined redundantly. A Hall-effect sensor was fitted in proximity to a timing gear on the dynamometer shaft. The frequency signal was split and fed into an LED display independent of the PXI chassis computer, while the other signal was fed into a frequency input card on the SCXI portion of the PXI Chassis. Type K thermocouples were installed into the exhaust downstream of the oxidation catalyst, the air intake between the intercooler outlet and the intake manifold, and the dynamometer cooling water outlet. These thermocouples were connected to a dedicated thermocouple voltage input card installed in the SCXI portion of the PXI Chassis. To determine engine torque, a strain-gage-type load cell was installed on the dynamometer’s torque arm and connected to an I/O card in the SCXI portion of the PXI chassis which was calibrated to provide the excitation voltage for the Wheatstone bridge of the strain gage. The output voltage of the strain gage was wired to an input channel on the SCXI I/O card. By measuring the force at the end of the dynamometer load arm, engine torque was calculated in the software by multiplying the force measured by the length of the dynamometer torque arm. Fuel mass was measured at a resolution of +/- 1 gram by an electronic balance connected via RS-232 communications to a serial port on the PXI chassis computer.

The engine torque command to the ECU was originally controlled by a pair of potentiometers mounted in the housing of the OEM accelerator. The ECU output five volts DC to two potentiometers wired as voltage dividers. The resulting voltage outputs of the voltage divider circuits varied between one half and two volts on one channel and
one and four volts on the other channel. At any given time, the voltage of one accelerator signal channel was to be half of the other, creating redundancy in the system. If an error was detected, meaning either of the signal voltages was out of range or incorrectly proportioned with respect to the other, the ECU would lock the engine at a speed of 1000 rpm until the power to the ECU was cycled and the correct signals were restored. To eliminate the physical accelerator pedal from the test-stand setup, two channels of an analog voltage output card of the PXI chassis computer were wired to the ECU inputs for accelerator pedal voltage channels. The voltages of these two channels were output independently by the PXI chassis controller and analog output card and configured to the correct 2:1 relationship required by the ECU.

To sample NO\textsubscript{X}, an on-board vehicle NO\textsubscript{X}/O\textsubscript{2} sensor manufactured by ECM and shown in Figure 3-9 was integrated into the test stand, with the physical sensor installed in the exhaust stream just downstream of the diesel oxidation catalyst. The operation of the sensor has been described in Chapter 2. The NO\textsubscript{X} sensor communicated its readings by sending out messages containing the percent oxygen and the concentration of NO\textsubscript{X} in the exhaust stream over a CAN network. To get the data messages from the CAN network into LabView, a National Instruments CAN-to-USB converter was wired between the terminus of the CAN network and a USB port on the PXI chassis.
3.1.3.2 Control, Monitoring, and Data-Logging Software

Monitoring, control, and data-logging of the test-stand systems were performed by a program written in Labview for execution in the PXI chassis computer. To perform data acquisition of the parameters available from the ECU, a subroutine was coded in LabView and executed by the PXI chassis computer, communicating through a serial port via RS-232. The program sent hex strings corresponding to parameter request commands over RS-232 to an RS-232 to ISO-14230 converter module. This converter shifted the RS-232 signal voltage to the higher voltage necessary for ISO-14230 communications.
over the ECU’s K-Line. The program first would initialize the K-Line to establish communications, then request engine speed, intake air temperature, coolant temperature, manifold boost pressure, and mass air flow one at a time per the ISO-14230 communications protocol used by the Magnetti Marelli ECU for diagnostic communications. After each request, the program would wait for the corresponding hex string to be received at the serial port. After reading the string from the serial port, it was decoded per ISO-14230, and converted to engineering units in decimal form to be displayed on the dynamometer control stand graphical user interface (GUI) and logged into memory by the data logger program. After each request was processed, the next parameter was requested and read until each of the requested parameters was collected, at which point the process was repeated. The rate of data collection was limited by the response time of the ECU: about 200 ms per request. This limited the data collection rate of the five ECU-based engine parameters to a maximum of one hertz.

The other parameters including engine speed, exhaust temperature, intake manifold air temperature, dynamometer cooling water temperature, engine torque, and fuel mass were read from the PXI chassis data acquisition (DAQ) cards and serial port as configured in Labview. The program collected these signals from the DAQ cards at a rate of 1 Hz to match with the ECU-parameter-sampling rate. The raw data were input into functions corresponding to each sensor’s transfer function, where the parameter was output in decimal engineering units. These data values were also displayed on the GUI of the control stand, and entered into the data log file. The program was written to allow data logging to be turned on and off, and to allow the user to adjust the frequency at which
data were sampled. The engine torque request was controlled by the accelerator pedal position signal input of the ECU. The accelerator pedal position could be varied between 0 and 100 percent by use of a slider or direct numerical entry into the LabView GUI. This control was set up in LabView to output the correct signals through the PXI hardware.

3.1.4 Test Stand Hydrogen Aspiration System

A system was set up to deliver, control, and monitor the flow of gaseous hydrogen into the intake of the diesel engine. Lab bottles of gaseous hydrogen were secured near the test stand, and were fitted with a combination pressure gauge and regulator. The output of the regulator was connected to a Matheson rotameter-type flow meter sized for the range of hydrogen flow rates expected to be used in the experiment. This was determined by calculating the maximum hydrogen flow at the lower flammability limit (LFL) of hydrogen in air, 4.1% by volume, as a function of the highest mass air flow expected as determined through modeling and subsequent baseline testing. A ten-turn metering valve fitted at the input side of the flow meter was used to control the hydrogen flow rate. The outlet of the flow meter was plumbed to a ¼-turn valve for emergency shut-off of the hydrogen should a leak be detected or should it become otherwise necessary. 3/8” stainless-steel tubing ran above the user interface through a larger volume chamber acting as a settling tank, to the air intake of the engine. A settling tank was used in the line to eliminate noise in the flow meter caused by possible pressure fluctuations in the engine
intake. Hydrogen was aspirated into the engine intake air just downstream of the mass air flowmeter (MAF).

The hydrogen was added to the intake air before the turbocharger for two reasons. First, the flow of the air and hydrogen through the turbocharger was intended to promote mixing in hopes that the hydrogen would be homogeneously distributed in the air by the time it reached the cylinder. The second reason was to ensure that the point of entry of the hydrogen into the intake air would remain near a steady pressure at or below ambient atmospheric pressure. For steady-state testing, positive pressure at the hydrogen point of entry could be compensated for by raising pressure in the delivery system until the desired flow rate was dialed in; however, during transient testing, which would be encountered in chassis dynamometer testing and real drive cycles, the varying pressure due to turbocharger boost would complicate control of the hydrogen injection system, as discussed in Chapter 4.

Two ways of consistently gauging the amount of hydrogen between operating points were considered: Percent of hydrogen in the intake air by volume, and percent of total fuel energy from hydrogen. Measuring hydrogen on an energy basis was decided upon, due to the large variation of equivalence ratio of diesel engines as a function of load (Heywood 1988).
3.1.5 Steady State Dynamometer Testing

To determine any effects that hydrogen aspiration would have on engine performance and emissions, baseline testing with no hydrogen was conducted at the eight engine operating modes determined to be most significant in the modeled Equinox UDDS drive cycle. The engine was first run at low speed and load, typically 1000 RPM at 15 NM torque, until normal operating temperature was reached. The dynamometer load and accelerator pedal position (APP) signal were then adjusted until the desired engine speed and load were reached simultaneously. Engine exhaust temperature was monitored, and data logging started once the exhaust temperature stabilized. The parameters to be analyzed (engine speed, torque, exhaust temperature, NO\textsubscript{X} concentration, percent oxygen, mass air flow, and fuel mass) were recorded. Each of the remaining modes were tested in the same manner, and partially analyzed before hydrogen induction testing was begun.

Using the mass air flow recorded at each point during baseline testing, and brake specific fuel consumption (BSFC) calculated, the volumetric flow rates of hydrogen equivalent to five and ten percent of the total fuel energy into the engine were calculated and verified not to exceed the LFL of hydrogen in air. The hydrogen flow rates calculated and used are shown in Table 3-3.
The mixture of hydrogen in air was kept below the LFL to prevent potential ignition in the considerable combined volume of the intake and charge air lines, turbocharger, and intercooler. After warm up, each test point was initially dialed in using the method described for the baseline test points. Next, hydrogen flow was adjusted to the rate determined for the operating point and percent energy to be tested. The accelerator pedal position signal was lowered to bring the engine speed, raised by the addition of the hydrogen into the engine, back to the desired operating point. Exhaust temperature was again allowed to stabilize before data logging commenced. Each operating point was tested in a similar manner for both five and ten percent hydrogen energy. During the testing, no unexpected problems with the engine operation were encountered, other than the need to provide additional capacity to the engine cooling system for the prolonged testing at the relatively high-load modes.

Table 3-3: Hydrogen flow rates for 5% and 10% of total fuel energy.

<table>
<thead>
<tr>
<th>Operating Point</th>
<th>Engine Speed (RPM)</th>
<th>Load (NM)</th>
<th>H2 Flow, 5% (SLPM)</th>
<th>H2 Flow, 10% (SLPM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1000</td>
<td>15</td>
<td>2.1</td>
<td>4.2</td>
</tr>
<tr>
<td>2</td>
<td>1750</td>
<td>60</td>
<td>9.4</td>
<td>18.8</td>
</tr>
<tr>
<td>3</td>
<td>1750</td>
<td>45</td>
<td>7.3</td>
<td>14.6</td>
</tr>
<tr>
<td>4</td>
<td>1750</td>
<td>75</td>
<td>11.3</td>
<td>22.6</td>
</tr>
<tr>
<td>5</td>
<td>1250</td>
<td>30</td>
<td>3.7</td>
<td>7.4</td>
</tr>
<tr>
<td>6</td>
<td>1250</td>
<td>45</td>
<td>5.4</td>
<td>10.8</td>
</tr>
<tr>
<td>7</td>
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<td>2.5</td>
<td>5</td>
</tr>
<tr>
<td>8</td>
<td>1750</td>
<td>30</td>
<td>5.6</td>
<td>11.2</td>
</tr>
</tbody>
</table>
3.2 Experimental Results of Dynamometer Testing

The data gathered during the steady-state tests with and without hydrogen aspiration were analyzed. The behavior of NO\textsubscript{X} emissions and efficiency were studied in relation to the addition of the increasing levels of hydrogen. It is important to note that the original engine controller was used, and no changes were made to the injection timing, nor was injection timing measured. Heywood (1988) notes that a common strategy to control NO\textsubscript{X} in diesel engines is to retard injection timing at a slight cost to BSFC as higher loads increase NO\textsubscript{X} output. This strategy of retarding injection timing with increased load is implemented in the diesel-engine ECU, and can be seen in manufacturer test data. The engine control system has no closed-loop control to calculate equivalence ratio, so the calibration expects a certain equivalence ratio with respect to accelerator pedal position (APP) and engine speed. When hydrogen is aspirated, APP must be lessened to maintain the same power output, indicating lower load to the ECU, and thus a point with lesser NO\textsubscript{X} output. The accelerator pedal position signal recorded during each trial was compared to the manufacturer data, and the small change in APP does not correspond to a shift in injection timing for any of the modes tested, thus eliminating that as a variable in NO\textsubscript{X} production. It is expected, though, that larger relative amounts of energy supplied with hydrogen would cause the injection timing to advance, which could lead to higher NO\textsubscript{X}, without factoring in any irregularities caused by the hydrogen combustion.
3.2.1 NO\textsubscript{X} Emissions

The effect of hydrogen aspiration into the intake air on NO\textsubscript{X} production in the diesel engine was analyzed by examining data collected from the NO\textsubscript{X} sensor fitted in the engine’s exhaust. The sensor reported total NO\textsubscript{X} in Parts Per Million (PPM). Figure 3-10 shows the NO\textsubscript{X} output in PPM for each of the trials. The magnitude of the error bars is the root sum squared (RSS) of the sensor accuracy as a percent of the measurement, and the percent uncertainty error due to the scatter of the data, evaluated at a 95% confidence interval (Wheeler and Ganji 1996).

![Figure 3-10: Total NO\textsubscript{X} emissions in parts per million.](image)

To compare NO\textsubscript{X} output on the brake-specific basis of grams per kilowatt hour, three initial assumptions were made. First, because the sensor was not able to differentiate NO from NO\textsubscript{2}, it was initially assumed that all of the NO\textsubscript{X} was NO. While this assumption is
certainly not accurate, it allows an initial comparison to be made on a brake-specific basis, upon which further error analysis can be conducted. Secondly, the exhaust was treated as typical diesel exhaust with an assumed makeup of 76% N₂, 10% O₂, 7% CO₂, and 7% H₂O with a molecular weight of 28.8 (Chan et al. 1992). The third and last assumption treated both the total exhaust and NOₓ species as ideal gases, with volumes of 22.4 liters per mole at STP. Using these assumptions and taking the exhaust temperature into account, the volumetric concentration of NOₓ was converted to a brake-specific output of grams of NO per kilowatt hour, shown in Figure 3-11.

Figure 3-11: Brake Specific NOₓ Output

The figure shows the specific NOₓ output in grams per kilowatt hour, assuming all NOₓ is NO. The uncertainty of the species distribution is accounted for in the error analysis,
and is evident from the height of the positive error bars. The negative magnitude of the error bars is the RSS of the sensor accuracy as a percent of the measurement, and the percent uncertainty error due to the scatter of the data, evaluated at a 95% confidence interval. The magnitude of the positive error bar is lengthened by the addition of the uncertainty of the NO/NO$_2$ distribution. Experiments performed by Lilik (2008) using a NO$_X$ analyzer with the capability to differentiate NO from NO$_2$ show that aspiration of hydrogen into the intake air of a diesel engine caused conversion of NO into NO$_2$, to the point where NO$_2$ dominated NO emissions. The suspected mechanism is the initial oxidation of the aspirated H$_2$ and O$_2$ into H and HO$_2$. This resultant HO$_2$ and NO then react to form NO$_2$ and OH (Lilik 2008). While NO$_X$ sensor error overshadows the decrease of NO$_X$ in PPM, the trend suggests reduced NO$_X$ concentration with increasing hydrogen aspiration for six of the eight modes.

3.2.2 Efficiency Results

The efficiency of the engine was calculated for each trial, taking into account the total fuel energy into the engine and mechanical energy output by the engine. The fuel energy into the engine was calculated using the lower heating value of each of the fuels for the mass of fuel consumed. The diesel fuel mass was measured by a balance during operation, and the hydrogen mass was calculated from the measured flow rate and test time. Energy output was calculated simply from the power output of the engine by the test time. Figure 3-12 shows the efficiencies compared for increasing hydrogen percent for each operating point. The magnitude of each of the error bars is the RSS of the scale
accuracy, flow meter accuracy when applicable, power measurement accuracy, and time measurement accuracy, each as a percent of the measurement.

![Bar chart showing engine efficiency during steady-state testing with error bars for different mode numbers and hydrogen percentages.]  

Figure 3-12: Measured engine efficiency during steady-state testing.

No clear trend is observed to occur with increasing amounts of aspirated hydrogen, leading to the assumption that the small amounts of hydrogen added had little effect on the operating efficiency of the combustion.

### 3.2.3 Exhaust Temperature

The test points where hydrogen was injected typically show higher exhaust temperatures than the baseline testing at each of the eight modes. The temperatures recorded during testing are shown in Figure 3-13. The error bars in Figure 3-13 are not easily seen, due to
little scatter in the data and the high precision of the thermocouples used. The magnitude of the error bars is the root sum squared (RSS) of the thermocouple accuracy as a percent of the measurement, and the percent uncertainty error due to the scatter of the data, evaluated at a 95% confidence interval.

The slightly different trend shown in mode seven is likely due to increased error in the data taken during baseline testing. The error was caused by data being taken before the temperature had settled, thus the higher standard deviation of the data. The overall trend shows that the addition of hydrogen typically elevated the exhaust temperatures about ten degrees C over baseline temperature measurements. It is interesting to note that in modes

Figure 3-13: Exhaust temperatures for each test mode.
one, two and four the temperature increased with increasing concentrations of hydrogen, however in modes three, five, six, and eight the highest temperature was recorded with five percent hydrogen; in the latter modes the ten percent hydrogen exhaust temperatures were only slightly higher than the baseline exhaust temperatures.

3.3 Chapter Summary

A computer-simulated drive cycle was used to determine relevant engine operating modes for a small turbocharged diesel engine. The diesel engine was set up on a dynamometer test stand, and instrumented to allow data to be recorded as the test modes were run. A hydrogen fueling system was built, and the modes were operated with 5 and 10 percent hydrogen energy with the engine primarily fueled with a B20 biodiesel fuel blend. Data comparing NO\textsubscript{X} emissions, efficiency, and exhaust temperature were analyzed and the following was observed:

- A trend of NO\textsubscript{X} reduction was correlated to the increased hydrogen aspiration, but the reduction was not significant enough to lie outside of the range of measurement error. One possible cause of the reduced NO\textsubscript{X} could be an increase of pre-mixed combustion, lessening the NO\textsubscript{X} formation in the diffusion flame during the later phases of combustion.

- Engine efficiency showed no clear trend as the amount of hydrogen aspirated into the engine was increased. A decrease in efficiency might be attributed to more of the fuel being ignited before top dead center due to the displacement of part of the main diesel injection by aspirated hydrogen.
Exhaust temperatures generally rose above the baseline exhaust temperatures when hydrogen was added to the intake air. There are too many factors involved to reasonably speculate why the data shows slightly increased exhaust temperatures.

The steady state dynamometer testing proved reliable engine operation while fueled in part by hydrogen, allowing the second part of the experiment (vehicle testing) to be conducted without concern for engine operational problems.
CHAPTER 4
CHASSIS DYNAMOMETER TESTING

To further investigate the effects of hydrogen aspiration into the intake air of a light-duty diesel engine, drive-cycle testing of a diesel vehicle utilizing a hydrogen injection scheme was conducted on a chassis dynamometer. The details of the experimental setup along with the results of the testing are discussed in this chapter.

4.1 Experimental Setup

The set up of the experiment including test vehicle preparation and dynamometer setup, instrumentation, and control are discussed in this section.

4.1.1 Test Vehicle Preparation

A 2005 Chevrolet Equinox was prepared with a dual-fuel hydrogen/biodiesel system. The installation and adaptation of the diesel engine for use in the Equinox, installation of a hydrogen fueling, storage and delivery system and the development of a hydrogen injection scheme and hardware are discussed here. Figure 4-1 shows the packaging of the dual-fuel system and drivetrain components in the Equinox. Note that the hybrid electric vehicle components not relevant to this study are shown in grey.
4.1.1.1 Engine Adaptation and Installation

The diesel engine used in this part of the experiment differs somewhat from the engine initially modeled and used for steady-state dynamometer testing. The engine fitted into the Equinox was a 1.3L turbocharged, direct-injection diesel engine of the same physical specification featuring an updated variable-geometry turbocharger and intake air throttle. The enhanced turbocharger boosted the power output of the engine to 66 kW, and the peak torque to 200 Nm. The compression ratio and all physical internal parameters remained unchanged. The intake throttle is used only to smooth engine shutdown, and does not affect operation (Zima 2007).

Figure 4-1: Equinox dual-fuel system and modified drivetrain packaging.
The first step of installing the engine in the Equinox was to adapt it to fit with the five speed automatic transmission, with which the vehicle was originally equipped. Beginning from the geometry of the crankshaft of the engine, a custom flywheel was designed, and CNC machined from aluminum to replace the original dual-mass flywheel intended for use with a manual transmission. The flywheel was designed and modeled in SolidWorks and analyzed in CosmosWorks, an FEA program, to verify the design for strength. The flywheel was fitted with the diesel engine’s original ring gear, so that the engine starter could be used without modification, and the timing wheel necessary for crank angle position sensor operation, as shown in Figure 4-2.

Figure 4-2: Custom flywheel fitted with original engine timing wheel and ring gear.
The flywheel was designed to be bolted to a torque converter, which was modified to allow the small engine to perform adequate launches and hill starts of the large vehicle. The flywheel, mounted on the engine’s crankshaft with the modified torque converter attached, is shown in Figure 4-3.

The torque converter was adapted by ProTorque, a company specializing in modifying torque converters. The torque converter was taken apart, and the core was replaced with one that would yield a higher k-factor (Rivera 2007). The modified converter is shown in Figure 4-4. The stator was modified to raise stall in order to meet the curve shown in Figure 4-5.
Figure 4-4: Inside of converter modified by ProTorque (ProTorque 2007).

Figure 4-5: Desired torque converter curve for improved launch characteristic.
The engine and transmission were next installed in the Equinox, as shown in Figure 4-6. Modification was necessary to the passenger side of the engine compartment, where an aluminum mount was fabricated to mate the engine end mount to the mount point on the vehicle body. Diesel fuel, intake air and charge air lines were fabricated and installed in the vehicle along with a custom air-to-air intercooler sized for the engine. The cooling system and heater core of the vehicle were plumbed to the engine, and all necessary wiring was connected between the ECU and all engine sensors. An exhaust was fabricated out of stainless steel, and a test pipe, shown in Figure 4-7, was fit into the exhaust containing a thermocouple and the sensor of the ECM NOxCAN system. Controls wiring was installed to allow the engine to be started by the master vehicle controller, and the alternator, starter and battery were connected.

Figure 4-6: Engine, transmission, accessories and wiring installed in the Equinox engine bay.
4.1.1.2 Hydrogen Storage, Fueling and Delivery System

Hydrogen fuel was stored on board the Equinox for delivery to the injection system. A 34L, 5000 PSI, composite tank manufactured by Quantum Technologies was used as the storage vessel, and secured in the rear of the vehicle behind the rear bench seat as shown in Figure 4-8. The capacity of the hydrogen storage tank allows hydrogen to supplement the diesel engine with 5% energy for approximately 14 gallons of diesel fuel burned, or seven gallons of diesel fuel at 10% hydrogen energy, before depleting the hydrogen tank.
The tank featured three ports; a refueling port with a one-way check valve, a PRD outlet port for thermal protection, and a solenoid-switched fuel-outlet port internally regulated to 200 PSI. The tank was equipped with internal pressure and temperature sensors.

To deliver the hydrogen from the tank, 3/8” stainless steel lines and compression fittings, both made of type-316 stainless steel, were installed. The refueling port was connected to a WEH standard hydrogen fill port installed behind the vehicle’s unoccupied fuel filler door. The PRD, used to relieve tank pressure in case of a thermal event, was vented to the exterior at the rear of the vehicle, exiting through the floor near the corner of the bumper. The fuel outlet port was connected to a regulator to reduce hydrogen pressure to
20 PSI for the long delivery line, running under the vehicle body to the injector in the engine bay. Figure 4-9 shows the layout of the hydrogen delivery system.

Figure 4-9: Hydrogen storage and delivery system.

The delivery line was equipped with a ¼-turn valve located on the rear passenger side quarter panel to allow manual shut off of hydrogen fuel as a safety precaution. A defueling valve was fitted under the vehicle, next to a secondary pressure relief device set to crack at 80 PSI to prevent high pressures from damaging the injection system. A settling tank was installed inside of the passenger side frame rail to add volume to the delivery line to minimize pulsations caused by operation of the solenoid injector.

Hydrogen sensors were installed in four locations in the passenger compartment of the vehicle. The sensors were set to emit an audible alarm at a small fraction of the LFL of hydrogen in air to warn the operator of a leak or malfunctioning system. Redundant
sensors were installed in the vicinity of the tank, and in the headliner above the driver and passenger seats. This configuration was meant to detect leaks first in the hydrogen storage area, and secondly in the top of the passenger compartment where leaking hydrogen would likely collect.

The hydrogen storage tank’s pressure and temperature sensors were wired to a module that both powered the sensors, and also converted the sensors’ analog outputs to a CAN message available to the master vehicle controller (MVC). The MVC evaluated pressure and temperature of the tank before opening the solenoid valve, to prevent the tank from nearing atmospheric pressure, as equalization with atmospheric pressure could allow air into the tank, which is undesirable. An additional connector on the Equinox allowed interfacing between the tank temperature and pressure sensors and a hydrogen fueling station via data link. A resistor wired into the harness on the vehicle side was used to indicate the volume of the tank to the station, as well. Using the fixed tank volume, temperature and pressure were obtained by the communications link allowing the station to fill safely at a faster rate than is possible without tank feedback. The hydrogen tank was fueled without incident to 5000 PSI at the General Motors Milford Proving Grounds during testing at the Challenge X competition in 2007.
4.1.1.3 Hydrogen Injector System

The hydrogen injection system was made up of three main components including a solenoid-type gaseous fuel injector, an electronic driver circuit, and a controller. The fuel injector chosen was purchased from Quantum Technologies, the manufacturer of the hydrogen fuel tank. A solenoid type injector, shown in Figure 4-10, was chosen based on the simple linear relationship between input signal duty cycle and flow rate (Barkhimer and Wong 1995). Flow was controlled by commanded duty cycle. The Quantum injector was specifically chosen because of its design for gaseous fuels, large flow capacity to allow the use of only one injector, and its design for use with a peak-and-hold-type drive signal. Power savings and reduced heat dissipation are additional benefits of the peak-and-hold-type injector versus the saturated-drive-type injector, due to the large current necessary to quickly open the solenoid valve only being applied at the beginning of the injection (Barkhimer and Wong 1995). The injector current is then lowered to only the level needed to hold the solenoid open, as shown in Figure 4-12.
An injector drive circuit was constructed around a National Semiconductor LM-1949 injector drive controller chip to power the injector. The circuit constructed is shown in Figure 4-11. A square wave, pulse width modulated signal of the desired injector operating frequency and duty cycle controls is input to the chip. The circuit drives a power transistor, supplying current to injector.

Figure 4-10: Quantum gaseous fuel injector (Quantum 2003).
Figure 4-11: National Semiconductor LM-1949 injector drive circuit (National Semiconductor 1995).

Figure 4-12: Peak-and-hold current profile (National Semiconductor 1995).
Once the circuit and injector were verified to operate as expected, the injector was calibrated. An apparatus was constructed to supply the injector with hydrogen, regulated to 20 PSI through a flow meter and a settling tank, to prevent fluctuations in the flow meter. The outlet of the injector was piped through an electrically grounded steel tube to a safe height above the ground where the unburned hydrogen was exhausted during calibration. The control frequency of the injector was set at 50 Hz, and the duty cycle was swept throughout the operational range of the injector. The lowest duty cycle usable, governed by the reaction time of the solenoid, was about ten percent at 50 Hz, or two thousandths of a second. The low-duty-cycle limit was determined experimentally as the lowest duty cycle with consistent operation in the range of linear flow as correlated to duty cycle. The injector could be operated reliably up to 100% duty cycle for maximum flow; however, the calibration was only carried out to 45 percent duty cycle because of flow meter limitations. The results of the injector flow calibration testing are shown in Figure 4-13. The offset of the curve represents the nonlinearity between the flow rate and duty cycle that occurs below 10% for the frequency and injector used. This occurs below a certain response time for all injectors, but is dependent on the physical characteristics of the injector and the characteristics of the drive circuit (Barkhimer and Wong, 1995).
Once the flow calibration of the injector was complete, it was installed into the Equinox. A two-piece injector holder was constructed. The bottom part was machined from aluminum and welded to an aluminum air intake tube. The top piece was machined from steel, allowing it to be welded to the steel tubing used for hydrogen delivery. The two pieces were bolted together, holding the injector in place. O-rings on the injector sealed against the holes machined in the holder. The injector, installed in the holder and mounted in the vehicle, is shown in Figure 4-14. The injector was wired to a control box also located in the engine compartment.

Figure 4-13: Hydrogen injector calibration test data and linear fit.

\[
y = 1.1453x - 1.2143
\]
Control of the injector was accomplished by an embedded PIC microcontroller locally interfaced with the injector drive control circuit shown in Figure 4-11. This configuration allowed the Equinox’s MVC, which communicates to the vehicle’s distributed I/O modules via CAN, to control the injector. Injector controller on/off and two identical injector-duty-cycle CAN messages originating from the master vehicle controller were decoded by the PIC microcontroller. The PIC, upon receiving the signals, checked the two duty cycle commands for consistency. If the messages were identical, the injector drive circuit would be enabled, and a PWM signal corresponding to that commanded would operate the injector.

Figure 4-14: Hydrogen injector mounted in Equinox engine compartment.
The hydrogen injection system was activated only when the hydrogen storage tank temperature and pressure were sensed to be within their operational limits, and the engine was running. Hydrogen was not injected during engine cranking or startup. The amount of hydrogen to inject was based upon the instantaneous diesel fuel flow rate to the engine, with hydrogen injection beginning only when the diesel fuel flow rate was high enough to warrant a flow of hydrogen above the minimum limit of 10% injector duty cycle. Thus, low load and idle conditions typically did not activate the injection system because the system was configured to introduce only a small percentage of the fuel energy as hydrogen. The diesel fuel flow information was available to the MVC from the ECU. The programming of the MVC enabled the system to inject hydrogen on a percent total fuel energy basis, as was used for the steady-state dynamometer testing. A parameter in the control code allowed a user-specified percentage of hydrogen energy to be commanded. While there was no direct algorithm in the controller preventing the mixture of hydrogen in the intake air from entering the combustible regime, hydrogen energy levels up to 15 percent were verified to be below the lower limit of flammability in air throughout the operating range of the engine. Because ten percent was the maximum value used for testing, the mixture in the intake was maintained below the lower flammability limit with a factor of safety at all times. Comprehensive testing of the hydrogen system was carried out during the 2007 Challenge X competition. The engine was operated in dual-fuel mode during performance, drivability, and consumer acceptability testing. The onboard storage tank was nearly depleted and refueled to 5000 PSI, confirming the integrity of the refueling system. No system or drivability problems were encountered throughout several hundreds of miles of test driving.
4.1.2 Test Stand and Chassis Dynamometer Preparation

To conduct testing over a uniform drive cycle, a chassis dynamometer was used to load the vehicle drivetrain. The dynamometer was used to dissipate power as the vehicle was driven in place, transmitting power to dynamometer rollers through contact with the vehicle tires. The setup of the vehicle on the dynamometer is shown in Figure 4-15. The dynamometer allowed a simulated cycle to be driven after vehicle parameters were programmed into the system. A simplified vehicle model was used to program the dynamometer. For that model, coast-down data obtained for the Equinox was programmed into the dynamometer controller, allowing the dynamometer to calculate the correct load to correspond to vehicle acceleration and road-load power requirements. The 505 cycle was chosen for testing, as it represents the first 505 seconds of the UDDS cycle that was previously modeled, and discussed in Chapter 3. To log the vehicle power and speed during testing, a data logger built into the dynamometer control stand was used. Once the dynamometer was programmed with the necessary vehicle parameters and the desired speed trace for the 505 cycle, instrumentation for data collection was prepared.
Instrumentation was set up to record fuel mass, exhaust temperature, NO\textsubscript{X}, MAF, \%O\textsubscript{2}, and hydrogen injector duty cycle from the vehicle. The vehicle engine was fueled with B20 biodiesel blended fuel from an externally mounted tank and low-pressure pump during dynamometer testing. This method allowed the fuel mass to be measured by an electronic balance with an accuracy of +/- 1 gram continuously during the test cycles, eliminating the need to remove and weigh the vehicle tank before and after each test. Exhaust temperature was sensed by a K-type thermocouple installed in the exhaust, alongside the ECM NOXCAN probe. The NOXCAN sensor operated with an accuracy of +/- 30 PPM at NO\textsubscript{X} levels under 1000 PPM, and +/- 3\% above. MAF was measured
by the vehicle’s MAF sensor, and was interfaced to the data logger through a CAN-to-
RS-232 adapter. The MAF signal, along with each of the other sensors, was sent to a
National Instruments portable PXI chassis computer for data monitoring and logging.
The control station used for monitoring and data logging is shown in Figure 4-16. A
program was written in LabView to provide a GUI displaying all of the measured values
and to provide data logging.

Figure 4-16: Chassis dynamometer control stand.

Testing was performed with the vehicle on the dynamometer to ensure that the
dynamometer functioned correctly, and that the data logging software worked as
designed. Testing was delayed by a broken exhaust-temperature sensor on the vehicle.
The data from this sensor is used as an input into a soot-loading algorithm that is programmed into the vehicle ECU and is used to determine appropriate times to regenerate the diesel particulate filter (Zima 2007). The absence of the temperature signal triggered a fault mode in the vehicle ECU enabling a low-power, limp-home mode that greatly limited engine output until the problem was corrected. Once the sensor was replaced, the vehicle again functioned correctly with full engine power and was ready for testing.

4.2 Experimental Results

The vehicle was first run through the 505 test cycle with no hydrogen injected into the intake air as a baseline. The test was repeated with 5 and 10 percent hydrogen energy modes programmed, respectively.

4.2.1 Cycle Consistency

The desired 505 speed trace is shown, compared to each of the actual test speed traces in Figure 4-17. During each test, the vehicle speed deviated only slightly from the desired drive cycle speed.
Consistency between drive cycles not only depended on the vehicle speed trace being matched, but also consistency of the engine speed and load between trials. Figure 4-18 shows the engine speed throughout each of the trials.

Figure 4-17: Speed traces for the 505 test cycle.

Consistency between drive cycles not only depended on the vehicle speed trace being matched, but also consistency of the engine speed and load between trials. Figure 4-18 shows the engine speed throughout each of the trials.
Small differences in driver input affected when the transmission shifted during each run. The transmission behavior, along with small differences in wheel speed, account for some difference in engine speed from trial to trial. Figure 4-19 illustrates the cumulative error from the speed and load mismatches between trials, shown by a difference in integrated power absorbed by the dynamometer. The small divergence of the energy output throughout the cycles shows that only a small difference in energy was expended between tests.

Figure 4-18: Engine speed for each of the trials.
The details of the hydrogen injection are shown in Figure 4.20 and Figure 4.21 for 5 and 10% hydrogen, respectively, where the duty cycle is shown over the drive cycle time.

Equations 4.1 and 4.2 relate the hydrogen injector duty cycle to flow, and the duty cycle to corresponding diesel flow rate for the percent of hydrogen energy programmed.

$$H2 \text{ FLOW (SLPM)} = 1.11 \times (\text{Duty Cycle}) + 1.18$$

4.1

$$\text{DUTY CYCLE} = 2045.93 \times (\% \text{ H2 Energy}) \times (\text{Diesel Flow l/s}) - 1.06$$

4.2
Figure 4-20: Injector duty cycle for 5% hydrogen mode.

Figure 4-21: Injector duty cycle for 10% hydrogen mode.
Once it was confirmed that each of the cycles were run in a reasonably consistent manner and that the hydrogen injection system had functioned as planned, the temperature, NO$_X$ and efficiency data logged over each run were analyzed.

4.2.2 Exhaust Temperature

The exhaust temperatures over each of the cycles were logged. Figure 4-22 shows the traces of the data. The addition of hydrogen clearly increased the cycle temperature; however, one source of error during the measurements was a premature start of the 5% hydrogen mode cycle. The cycle was begun before exhaust temperatures had stabilized. The higher temperatures of the engine and exhaust may contribute to some error in the NO$_X$ measurements until the temperatures converge with the other traces around 100 seconds into the cycle.
Figure 4-22: Exhaust temperature for each mode tested over the cycle.

4.2.3 NO\textsubscript{X} Emissions

Figure 4-23 shows the NO\textsubscript{X} output in PPM throughout the drive cycle for each of the hydrogen injection modes tested. The average NO\textsubscript{X} output for each test cycle is shown in Table 4-1.
output in grams per mile is shown, as calculated from NO\textsubscript{X} concentration as in Chapter 3, in Figure 4-24. The large positive error bars are due to the uncertainty of how the NO\textsubscript{X} species are divided into NO and NO\textsubscript{2}. The NO\textsubscript{X} is initially assumed to be all

Figure 4-23: NO\textsubscript{X} emissions in PPM for baseline (0), 5, and 10 percent hydrogen.

Table 4-1: Average NO\textsubscript{X} output for each mode.

<table>
<thead>
<tr>
<th>NOx Output (PPM)</th>
<th>Baseline (0% H2)</th>
<th>5% H2</th>
<th>10% H2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Baseline (0% H2)</td>
<td>249.1 +/- 30</td>
<td></td>
<td></td>
</tr>
<tr>
<td>5% H2</td>
<td>252.3 +/- 30</td>
<td></td>
<td></td>
</tr>
<tr>
<td>10% H2</td>
<td>207.2 +/- 30</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>
NO, with the error bars accounting for the possibility that up to 100% of the NO\textsubscript{X} is NO\textsubscript{2}, however unlikely this may be. These calculations and error analyses are similar to those performed in Chapter 3, however the NO\textsubscript{X} is reported in grams per mile instead of grams per kilowatt-hour. The NO\textsubscript{X} data show that the hydrogen aspiration had little effect on the total NO\textsubscript{X} output however any conversion between the two species NO and NO\textsubscript{2} is unknown.

Figure 4-24: NO\textsubscript{X} reported in grams per mile.
4.2.4 Efficiency

Efficiency of each of the cycles was calculated by dividing the mechanical energy output by the vehicle over the cycle by the fuel energy supplied to the engine over the course of the cycle. Figure 4-25 shows the compared efficiency over the cycle for each mode. The efficiency was not affected greatly; however, the 10% hydrogen mode suffered a small penalty in efficiency. Figure 4-26 shows both fuel types used, and how much energy of each fuel type was used for each of the trials. While the 5% hydrogen mode displaced very little diesel fuel usage, the 10% mode allowed more hydrogen to be injected, and even though efficiency dropped somewhat, the amount of diesel fuel used is less than the baseline. The error bars were calculated in the same manner as those in Chapter 3.

![Graph showing efficiency comparison](image)

4-25: Overall efficiency of the cycle for each mode tested.
4.3 Chapter Summary

A complete onboard hydrogen-assisted combustion system was installed in modified 2005 Chevrolet Equinox equipped with a diesel engine. The vehicle was prepared with a compressed-hydrogen storage system, hydrogen fueling system, hydrogen delivery and injection system, and hydrogen detection system for the interior of the vehicle. Safety was of paramount concern in the design of the system, and all systems feature redundant fail safe modes designed around a failure mode analysis. The hydrogen injection system was designed around an electronic injector which was supplied with hydrogen from the on-board storage, and controlled by the MVC. The calibration done on the injector during the design of the system was used to program the MVC.

4-26: Fuel energy usage per type of fuel per trial.
Extensive qualitative testing of the vehicle operating in dual-fuel mode was carried out during the course of several hundred miles on test tracks and roads. To obtain quantitative data on emissions, efficiency, and temperature the vehicle was run through a test cycle on the dynamometer to compare baseline operation to the hydrogen dual fuel modes. Observations about changes over baseline testing obtained from the dynamometer testing are shown in Table 4-2.

Table 4-2: Changes from baseline test data.

<table>
<thead>
<tr>
<th></th>
<th>Average Exhaust Temperature</th>
<th>Average NO\textsubscript{X} Concentration</th>
<th>Overall Efficiency</th>
</tr>
</thead>
<tbody>
<tr>
<td>5% H\textsubscript{2} Mode</td>
<td>INCREASE</td>
<td>INCREASE</td>
<td>NO CHANGE</td>
</tr>
<tr>
<td>10% H\textsubscript{2} Mode</td>
<td>INCREASE</td>
<td>DECREASE</td>
<td>NO CHANGE</td>
</tr>
</tbody>
</table>
CHAPTER 5
CONCLUSIONS AND FUTURE WORK

5.1 Conclusions

Through the design and testing of a hydrogen-assisted diesel combustion system in a light-duty, production diesel engine, and integration into a vehicle the following conclusions have been drawn.

- Steady-state operation is not negatively impacted by the addition of small amounts of hydrogen, relative to the total fuel energy, into the engine. Emissions and efficiency were maintained, and a trend suggesting a decrease of total NOX emissions was observed with the addition of hydrogen.

- Addition of hydrogen to the intake air of the diesel engine raised the exhaust temperature during steady-state testing. These heightened temperatures could improve the light-off and efficiency of emissions control components.

- Hydrogen, in the amounts added in this experiment, did not have a noticeable qualitative effect on the reliability, noise level, or feel of the engine’s operation.

- The integration of a hydrogen system into a production vehicle can be practically and safely performed. The most substantial limitation is the size of hydrogen storage. The type of storage used in these experiments was suitable for only small amounts of hydrogen as tested, up to 10% of the total fuel energy. Using greater amounts of hydrogen would cause the hydrogen fuel supply to be exhausted.
before a typical diesel fuel tank would need to be refilled. The storage vessel also occupied a substantial volume in the passenger compartment, but the storage area was not rendered useless due to the large size of the vehicle interior.

- Operation of the diesel engine through transient cycles with hydrogen injection was maintained with little or no increase in NO\textsubscript{X} emissions and closely maintained efficiency. The hydrogen injection did cause higher average exhaust temperatures over the cycle tested, but these temperature increases were slight.

- The driving feel of the engine burning 5 and 10 percent hydrogen energy was identical to the baseline mode when operated on the dynamometer, road, and track. The hydrogen system operation caused no noticeable increase in noise.

### 5.2 Future Work

With the work described in this paper completed, the groundwork has been laid for further investigations utilizing an onboard hydrogen fuel system, along with an onboard NO\textsubscript{X} emissions measuring system. Having proved the idea is feasible and no ill effects have been encountered, further baseline testing could be completed on the engine dynamometer to map NO\textsubscript{X} reduction or augmentation through the engine operating range. This information could be used to map the most useful hydrogen injection strategy, and this information could be programmed into the MVC in an attempt to inject hydrogen in a concentration where it is beneficial or at least causes no increase in NO\textsubscript{X} and does not negatively affect engine efficiency.
Another item of interest would be the integration of the hydrogen fuel and NO\textsubscript{X} measurement systems with a selective catalytic reduction (SCR) system installed in the vehicle. If a viable automotive ammonia sensor could be obtained, as has been developed by Delphi, then closed-loop control of urea injection into the exhaust to react with the SCR catalyst to lower NO\textsubscript{X} could be attempted (Patrick 2008). A potentially more interesting idea would be to combine such a system with the use the hydrogen system to hasten the light off time of the catalyst, either by raising exhaust temperatures slightly as described in Chapters 3 and 4, or by oxidizing the hydrogen directly in the exhaust gas upstream of the SCR catalyst.

While the engine used in Chapter 4 has an algorithm to oxidize a diesel particulate filter on the vehicle using late injections to raise exhaust temperatures to oxide the built-up particulates, perhaps a system using hydrogen could be investigated to do the same.

More aggressive hydrogen fueling could be attempted, with fuel injected near the intake valve of the engine and with a greater percent of the fuel energy coming from hydrogen. This would require more modifications to the engine; however, with preliminary testing showing no detrimental effects from the use of small amounts of hydrogen in the diesel engine, adding more hydrogen may be useful.
BIBLIOGRAPHY


