Effects of Water Injection and Increased Compression Ratio in a Gasoline Spark Ignition Engine

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Abstract

A single cylinder, four stroke, gasoline, spark ignition engine was modified to test the effects of water injection in combination with an increased compression ratio in a engine. Three air/fuel ratios (13.7, 14.7 and 15.7), six water/fuel mass ratios (from 0 to .75) and two different compression ratios (6:1 and 7:1) were tested. It was found that water injection in combination with an increased compression ratio can increase torque output (up to 65%), reduce brake specific fuel consumption (up to 39%), lower exhaust temperature (up to 10%), lower BSNO emissions (by up to 78%) and lower BSCO emissions (by up to78%) but may increase BSHC emissions (up to 45%).

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Nomenclature

Table 1 Nomenclature

Symbol	Description
AFR	Air/Fuel Ratio
ATDC	After Top Dead Center
BHP	Brake Horsepower
BMEP	Brake Mean Effective Pressure
BSCO	Brake Specific Carbon Monoxide
BSFC	Brake Specific Fuel Consumption
BSHC	Brake Specific Hydrocarbon
BSNO	Brake Specific Nitric Oxide
BTDC	Before Top Dead Center
cc	Cubic Centimeters
CFR	Cooperative Fuels Research
CO	Carbon Monoxide
CSC	Clean Snowmobile Challenge
ECU	Electronic Control Unit
EFI	Electronic Fuel Injection
EGT	Exhaust Gas Temperature
H/C	Hydrogen/Carbon Ratio
HC	Hydrocarbon
HP	Horsepower
IMEP	Indicated Mean Effective Pressure
kW	kiloWatt
MAP	Manifold Absolute Pressure
MBT	Maximum Brake Torque
MEP	Mean Effective Pressure
NO	Nitric Oxide
NO_X	Mono-Nitrogen Oxides, NO and NO_2
PSIA	Pounds per Square Inch Absolute
RPM	Revolutions Per Minute
SFC	Specific Fuel Consumption
SI	Spark Ignition
TPS	Throttle Position Sensor
VR	Variable Reluctance
WFR	Water/Fuel Ratio
WOT	Wide Open Throttle
Φ	Equivalence Ratio

1.0 Introduction and Background

As the SI (spark ignition) engine has become widely used, inventors have sought ways to improve its performance. Introduction of water into gasoline SI engines has been researched for many years to improve the engine in various ways, including: to lower NO_X (oxides of nitrogen) emissions, to lower CO (carbon monoxide) emissions, to boost power output, to increase efficiency, to cool the engine and to reduce knock by increasing octane number. One detrimental effect of water addition is that it tends to increase the HC (hydrocarbon) emissions. There are various methods for adding water in SI engines such as: inlet manifold water injection, water mixed with fuel (emulsions) and direct injection of water into the combustion chamber. These methods do not yield identical results. The results of the same method may also vary from engine to engine. However, the results do generally exhibit the same trends, especially regarding the drop in NO_X emissions, increase in HC emissions and increase in fuel octane rating (i.e. the ability of the fuel to resist self-ignition). Higher octane rating means the fuel can be used in an engine with a higher compression ratio, without causing the engine to knock or the fuel to self-ignite. “Self ignition is when the pressure and temperature of the fuel/air mixture are such that the remaining unburned gas ignites spontaneously” pg 71 of (1). Also it is known that increasing the compression ratio in SI engines increases efficiency and power output. However, limited research has been found on water injection in combination with increased compression ratios. These experiments (which are elucidated in sections 2 and 3) were done using manifold water injection on a four stroke engine.

1.1 Water Addition Lowers NO_X Emissions

NO_X (NO and NO_2) formation increases with temperature, and “increase rapidly at temperatures above about 1800 K” pg 562 of (2). Reducing peak temperatures in SI engines can greatly lower NO_X emissions, because less energy is available to break up the triple N-N bond, which is the first step in NO formation via the Zeldovich or thermal mechanism. The Zeldovich NO mechanism is discussed in Turns (2) chapters 4, 5 and 15. Water addition is very effective at reducing combustion temperature because of its high latent heat of vaporization. Thus water is an effective in-cylinder control strategy for NO_X. Many studies confirm the effectiveness of water addition at lowering NO_X emissions. Different studies may report decreases in oxides of nitrogen in terms of NO, NO_2 or NO_X but the trends are similar.

Peters and Stebar (3) state that the peak NO_X level was lowered by nearly 40% with 40 weight (mass) percent water addition to gasoline. They state that both of their methods (direct manifold water addition and emulsified fuel-water mixture) were equally effective in diminishing NO_X emissions. See Figure 1. They further state that the most effective vaporization process was complete water vaporization at 170°BTDC (before top dead center).

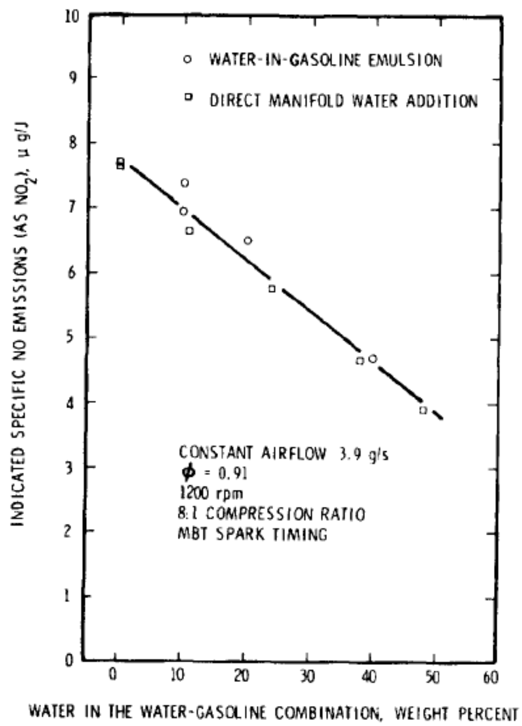

Figure 1 Effect of water on nitric oxide emissions (3)

Tsao and Wang (4) observed a 50% decrease in NO_X emissions when using 10% water by volume water/fuel mixture as compared to the base gasoline. But NO_X emissions were only lowered by about 40% when 15% water/fuel mixture was used. According to Nicholls, et al. (5) the injection of water on a water/fuel mass ratio of 1:1 is able to lower NO_X emissions by as much as 90%, depending on the equivalence ratio used. Lanzafame's (6) data showed a drop in NO_X production of over 50% with water/fuel mass ratio in the range of 1 to 1.25. Harrington (7) plotted NO_X across a wide range of equivalence ratios and showed that NO_X emissions can be lowered by up to 95%, depending on the amount of water added and equivalence ratio. Brusca and Lanzafame (8) showed a 50% drop in NO_X at a water/fuel mass ratio of 1. Wu, et al. (9) show a drop of about 50% in NO_X emissions with 15% water in the fuel by volume. Lestz, et al. (10) state that a 90% drop in NO_X is possible with a sacrifice in power, but 50% is possible without sacrificing power using maximum brake torque (MBT) spark timing. These studies all agree that water addition tends to lower NO_X emissions.

1.2 Water Addition Tends to Lower CO Emissions

Studies show water addition can also lower CO emissions. Tsao and Wang (4) concluded that CO emissions can be decreased by 40% with 10% water by volume. Wu, et al. (9) also found that water addition to fuel lowers CO emissions by approximately 50% for 15% water by volume. Peters and Stebar (3) found a small drop in CO emissions with water addition during their testing. This was only noticeable at rich conditions where CO levels were high with about a 10% drop for 40 weight percent water/fuel ratio. See Figure 2 Effect of water on carbon monoxide emissions.

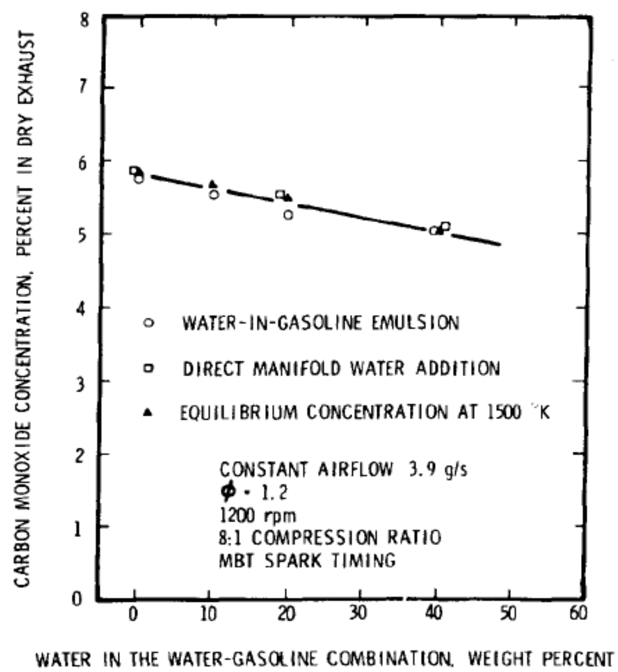

Figure 2 Effect of water on carbon monoxide emissions (3)

Lanzafame (6) pg 9, also found that decreases in CO production were “more noticeable with rich mixture running conditions where CO levels were relatively high” but did not state quantitatively how much. These studies indicate that water addition tends to lower CO emissions.

1.3 Water Addition May Increase HC Emissions

One main disadvantage of water addition is its tendency to increase hydrocarbon (HC) emissions. Multiple studies indicate that water addition tends to increase HC emissions (3, 4, 6, 9). Tsao and Wang (4) pg 86, report only "moderate increases" in HC emissions, i.e. a 30% increase in HC emissions for 15% by volume water/fuel ratio. Peters and Stebar (3) pg 1839, report that HC emissions "increased rapidly…a 40 weight percent water addition caused nearly a fourfold increase in hydrocarbon emissions". See Figure 3.

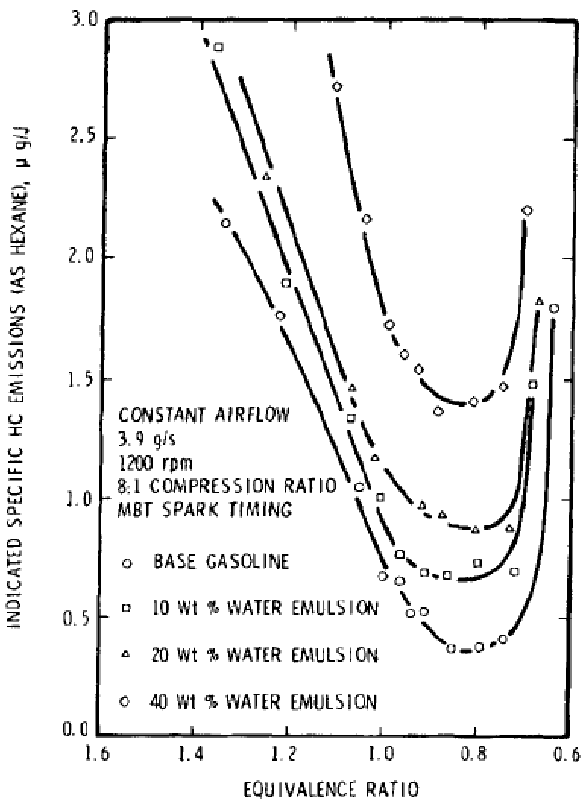

Figure 3 Hydrocarbon emissions with emulsified fuels (3)

Wu, et al. (9) indicate that on average, HC emissions increased 14.7% for 10% water/fuel volume ratio. Multiple sources (3, 4, 8, 9) agree that the higher HC emissions with water addition are likely caused by a change in the quenching layer of combustion and/or by a change in gas temperatures. Flame quenching is a process whereby a flame is extinguished a short distance from a cool surface (2). As the flame approaches the cooler combustion chamber walls it is extinguished, resulting in incomplete combustion. Because water cools the combustion chamber, it may increase the thickness of the quenching layer and thus also increase the amount of unburned hydrocarbons emitted during the exhaust stroke.

1.4 Water Addition Can Increase Power and Efficiency

Water injection may also improve power output and efficiency. Power is torque multiplied by RPM. A percent increase in torque results in the same percent increase in power for a given RPM. Torque is sometimes measured using mean effective pressure (MEP), with a higher MEP being more powerful. MEP is defined as:

$$MEP = \frac{work\ output\ per\ cylinder\ per\ mechanical\ cycle}{swept\ volume\ per\ cylinder}$$

Equation 1 Mean Effective Pressure

Improved efficiency is often reported as a reduction in specific fuel consumption (SFC), with lower SFC being more efficient.

$$SFC = \frac{mass\ fuel\ consumed\ per\ time}{power\ output}$$

Equation 2 Specific Fuel Consumption

Similar terms, brake mean effective pressure (BMEP) and brake specific fuel consumption (BSFC) are those calculated by using an engine dynamometer (brake) to measure work output. A useful equation relating work and BMEP is the following.

$$Brake\ Work\ Produced\ per\ Cycle = BMEP * swept\ volume$$

Equation 3 Brake Work Produced Per Cycle (1)

Efficiency can also be reported as follows, where $(-\mathrm{d}H)$ is the lower heating value of the fuel.

$$\eta = \frac{1}{SFC * (-\mathrm{d}H)}$$

Equation 4 Engine Efficiency

Multiple researchers state that maximum brake torque (MBT) spark timing may need to be employed with water addition in order to maximize power and minimize SFC. Moffitt and Lestz (11) found that after optimizing the spark timing, fuel economy and power were no better with water addition than without. Peters and Stebar (3) report that complete water vaporization at 170 degrees BTDC was most efficient (0.383%), but not significantly better than the base case with no water addition (0.371%). They also indicated that water addition did not improve indicated power output. However, multiple researchers have found that water addition can decrease SFC and increase power. Test results of Tsao and Wang (4) indicated that water addition (15% water in the fuel) increased engine output by 13% and reduced SFC by 9.5%.

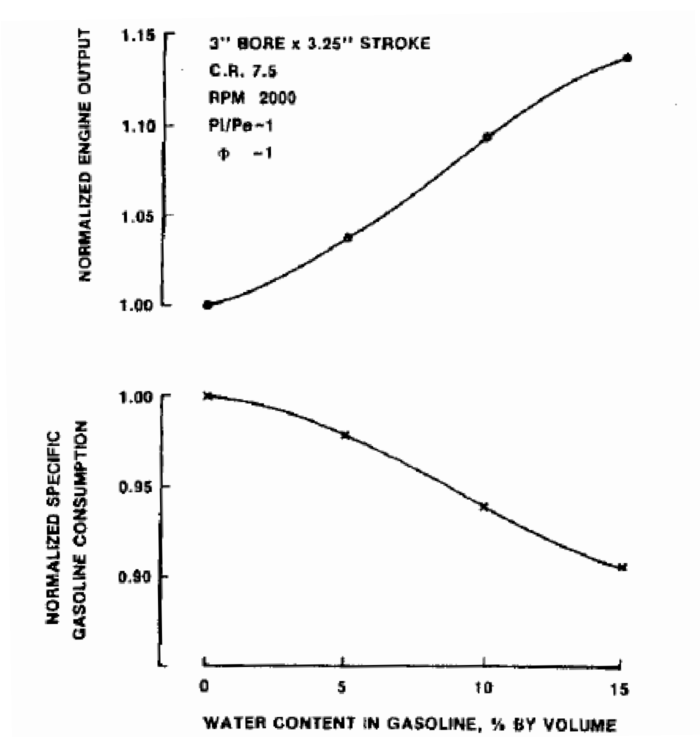

Figure 4 Normalized engine output and specific gasoline consumption (4)

Wu, et al. (9) found that 10% water by volume in the fuel increased engine torque by 4.1% and decreased brake specific fuel consumption (BSFC) by 3.4%; however engine performance deteriorated when water was increased to 15%. Nicholls, et al. (5) experimenting with a Cooperative Fuels Research (CFR) engine found that water/fuel mass ratios up to 0.75 yielded a slight increase (about 1-5%) in brake mean effective pressure (BMEP), whereas water/fuel mass ratios above 0.75 began to decrease BMEP until by a water/fuel ratio of about 1.25 BMEP diminished to the original value. Similarly, as water/fuel ratio increased to 0.75 then BSFC decreased (about 2-4%), further increase in water/fuel ratio increased BSFC.

Overall the research seems to indicate that adding water (up to a point) tends to improve power and improve (decrease) SFC. MBT spark timing may need to be employed to realize or maximize these benefits.

1.5 Possible Advantages of Direct Water Injection

Though the various methods of water addition yield generally similar results, direct water injection may be advantageous in some respects. Lestz and Meyer (10) concluded that direct cylinder injection results in a drop in NO_X emissions greater than 85% using about one-third the mass of water required by manifold injection for similar NO_X decreases. Juntarakod (12), in a theoretical study, concluded that direct cylinder water injection (up to 40% water/fuel molar ratio) after combustion could increase efficiency by 1-3% and increase indicated mean effective pressure (IMEP) about 2-7%.

1.6 Water Addition for Cooling

Water may also be used to cool the engine. Weatherford and Quillan (13) concluded that water addition by direct injection was adequate to cool the engine, i.e. water jackets and air fins could be eliminated. Modak and Caretto (14) confirmed that water addition could be used to cool internal combustion engines and used a computer program to conclude that the optimal time for direct water injection was early in the compression stroke. Nicholls, et al. (5) calculated theoretical flame temperatures across a span of water/fuel mass ratios, which showed that as water/fuel ratio increases temperature decrease. See Figure 5 Influence of water injection on theoretical equilibrium flame temperatures for isooctane

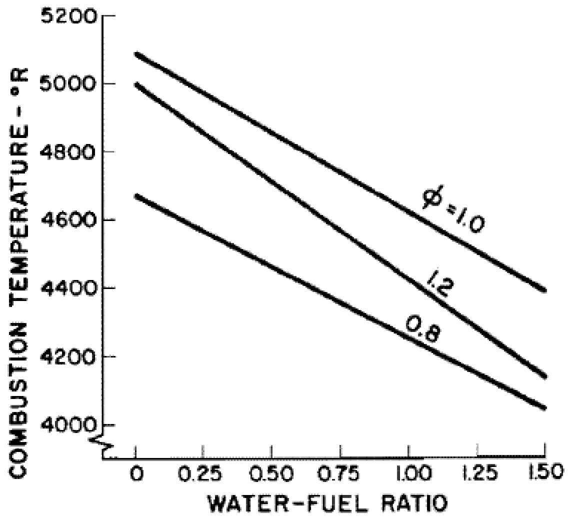

Figure 5 Influence of water injection on theoretical equilibrium flame temperatures for isooctane (5)

1.7 Water Addition Allows for Higher Compression Ratios

Another benefit of water addition is that it increases a fuel's octane number, which means that the fuel is more resistant to auto-ignition. Auto-ignition occurs at high pressures and temperatures, and can cause engine knock. Engines with a high compression ratio require fuels with a high octane rating, in order to avoid knock. Higher compression ratios result in increased power output and efficiency. Multiple studies agree that water has anti-knock characteristics (often expressed with higher octane numbers) (3, 4, 5, 6, 7, 8, 15, 16). Peters and Stebar (3) for example, graphed octane number across multiple water weight (mass) percents, see Figure 6 Effect of water content on the octane quality of water-gasoline emulsions. They showed octane number using the research method and the motor method.

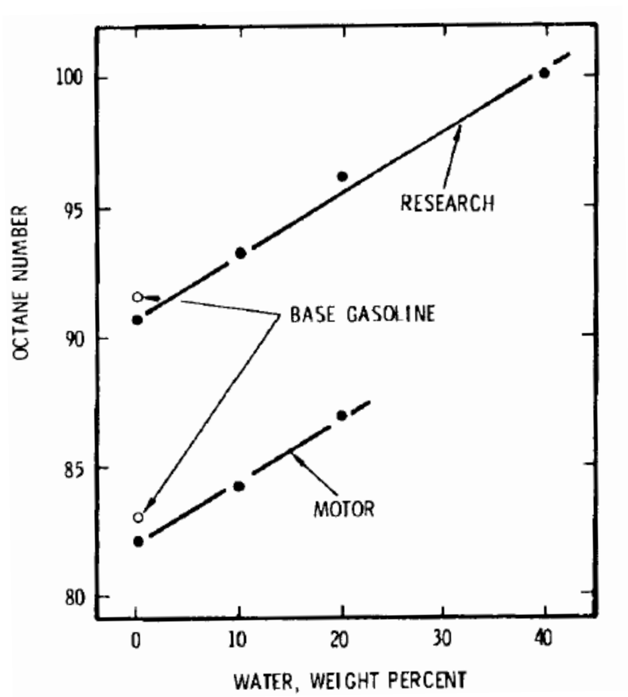

Figure 6 Effect of water content on the octane quality of water-gasoline emulsions (3)

Though many researchers state that water increases octane number, relatively few have implemented higher compression ratio engines to reap the benefits of the higher octane number. Described here are those found who have implemented higher compression ratios in combination with water addition.

Peters and Stebar (3) studied water addition (40% water by weight in the fuel) and increased compression ratio. The overall effect was that efficiency was slightly increased, HC emissions increased and NO_X emissions dropped. See Figure 7 below.

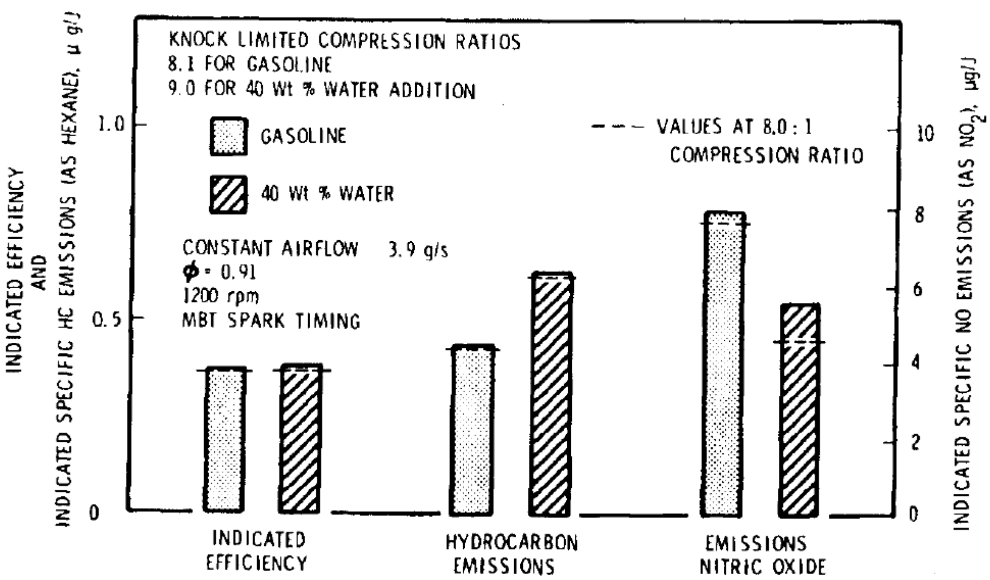

Figure 7 Effect of operating at the knock limited CR w/ and w/o 40 weight percent water (3)

Tsao and Wang (4) found significant improvements in specific gasoline consumption and power output when water addition was used in combination with an increased compression ratio. Ignition advance was increased as percent water increased. See Figure 8.

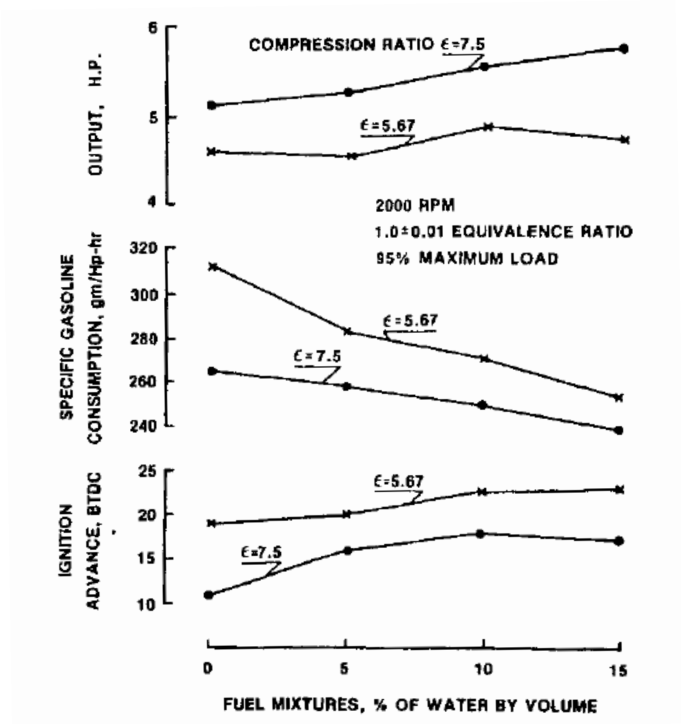

Figure 8 Effect of CR on engine performance of water gasoline fuels. (4)

Kettleborough and Milkins (16), using a variable compression ratio engine, found that adding water allowed them to raise the compression ratio of their engine from 6.2 to 7.55, which increased efficiency from 24.5% to 27.1%. Power output was also increased from 7.50 BHP to 9.02 BHP. They also recommended water injection to diminish carbon deposits in the engine.

In a theoretical part of their study, Nicholls, et al. (5) calculated the influence of compression ratio together with water injection on NO emissions, as shown in Figure 9. Water/fuel ratio here is on a mass bases. Theoretically, water addition greatly lowered NO emissions while increased compression ratios slightly increased them.

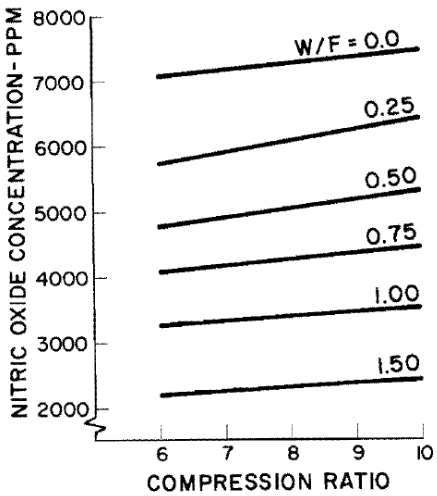

Figure 9 Influence of CR on theoretical effectiveness of water injection (5)

1.8 Summary of Background Research

To summarize, it has been shown the water addition to SI engines lowers NO_X emissions, tends to lower CO emissions, may increase HC emissions, may increase power and efficiency of the engine, cools the engine and increases octane rating which allows for higher compression ratios. Though many researchers agree that water addition allows for an increased compression ratio, relatively few studies have used higher compression ratios. Higher compression ratios also improve power output and efficiency.

1.9 Purpose of the Present Study

The present study shows the trends of multiple important effects on an SI engine due to water injection in combination an increased compression ratio.

Previous research has only considered three or fewer outcomes at a time, when considering water addition with an increase in compression ratio. Peters and Stebar (3) looked at efficiency, NO and HC emissions. Tsao and Wang (4) looked at ignition advance, specific gasoline consumption and power output. Others did less. No research was found that showed the effect on CO emissions caused by water addition with an increased compression ratio. The present study will show the effect of water injection and increased compression ratio on: torque, brake specific fuel consumption, temperature, NO emissions, CO emissions and HC emissions. Predictions were as follows:

- Torque output and efficiency were expected to be minimally affected by water added but increase with compression ratio.
- The engine exhaust temperatures were expected to be greatly cooled by water added but increase somewhat with compression ratio.
- NO_X levels were expected to diminish greatly with water added but increase slightly with increased compression ratio.
- CO emissions were expected to decrease with water added and be minimally affected by compression ratio.
- HC emissions were expected to increase with water added and with compression ratio.

It was desired to better understand and give greater insight on the effects of water injection with increased compression ratio.

To outline the remainder of this thesis, Section 2 describes experimental set up. This includes the engine used and modifications made to the engine, as well as equipment used to test the performance and emissions of the engine. Section 3 describes the experiment, the results and a discussion of possible reasons that water injection and increased compression ratio affect the engine in the way that they do. Section 4 summarizes the results of the experiment. Section 5 describes future work that would further expand what is known of the effects of water addition with increased compression ratios.

2.0 Experimental Setup

This section includes details of the engine used, modifications made to the engine, type of gasoline used and equipment used to measure outcomes.

A four stroke 362cc Briggs and Stratton engine (Model 222416, Type 0516-01) was used because it was relatively inexpensive, fairly robust, had a flat head (L-head) and could be relatively easily attached to the dynamometer (Borghi & Saveri) at the University of Idaho. The flat head (L-head) of this engine allowed for the compression ratio to be increased relatively easily by milling some of the material off the bottom of the head.

Table 2 Briggs and Stratton 222416 Baseline Specifications, (17, 18)

Engine Type	Single cylinder, L-head, air cooled engine
Model	222416
Type	0516-01
Bore	3-7/16 in. (87.31mm)
Stroke	2-3/8 in. (60.33 mm)
Displacement	22.04 cu. In. (361.2 cc)
Torque (Max.)	14.8 Ft.-Lbs (20.066 N-m) @ 3000 RPM
Compression Ratio	6:1

2.1 Engine Modifications

To provide the engine with an intake manifold fuel injection system and an intake manifold water injection system, products were purchased from MBE Motorsports (www.mbe-motorsports.com). This company makes kits to convert small carbureted engines to EFI (electronic fuel injection) engines. Some of the products in the kit were purchased twice (such as injectors and ECUs) to be able to control injection of both water and fuel. Most of the products from MBE Motorsports which enable water and fuel injection on the Briggs and Stratton engine can be seen in Figure 10 EFI Conversion Products.

Figure 10 EFI Conversion Products

Figure 11 is a schematic to assist the reader in understanding the engine setup. Various components shown here are explained in this section.

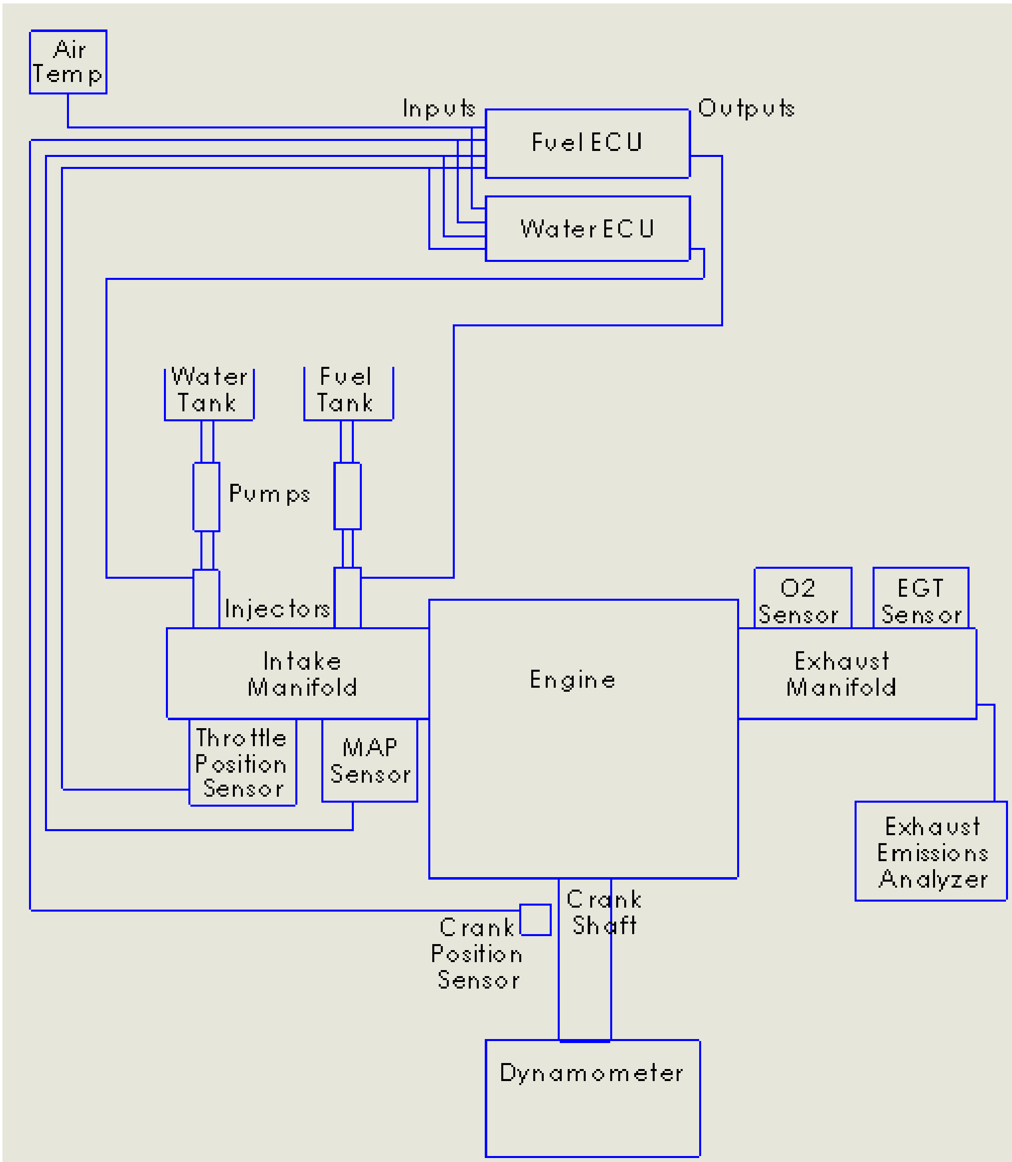

Figure 11 Engine Setup Schematic

An adapter was made to attach the intake manifold items from MBE Motorsports to the Briggs and Stratton engine. One end fit on the engine, the other end fit the intake manifold component that was closest to the engine. This adapter, shown below, was made with a 20 degree angle to keep the intake components a safer distance from the exhaust. A cork gasket was also made to seal the connection between this adapter and the engine, see Figures 12-15 below.

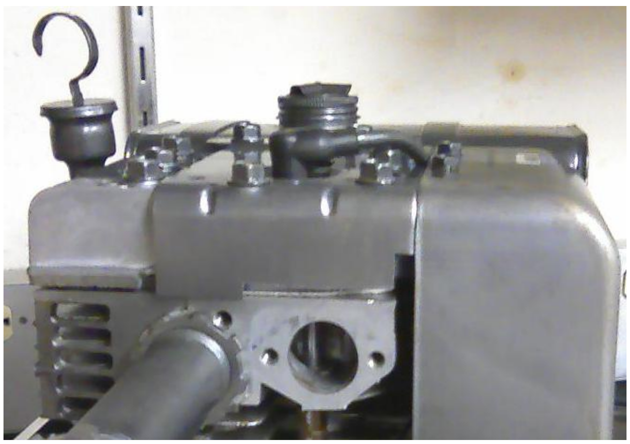

Figure 12 Engine, Intake Manifold Removed

Figure 13 Gasket and Adapter (rear)

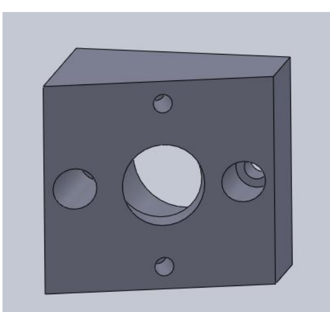

Figure 14 CAD Image of Adapter (front)

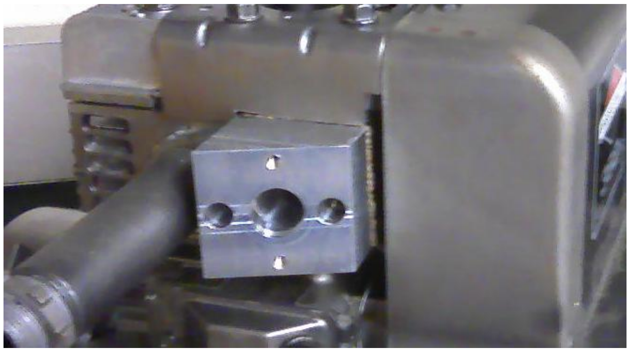

Figure 15 Engine with Adapter Attached

Next the first intake manifold part from MBE Motorsports was attached, which has one injector and seals using O-rings.

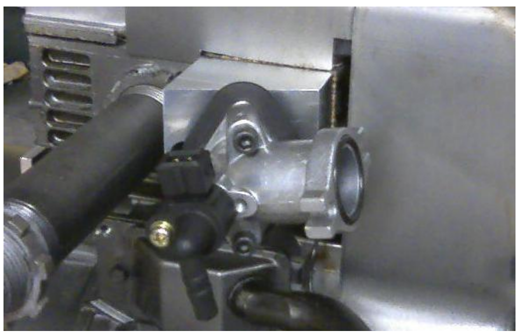

Figure 16 First Intake Manifold Part Attached

An adapter plate was machined to provide a concentric connection between the first and second intake manifold parts, and then the second manifold part was attached, which contained a throttle and throttle position sensor and another injector.

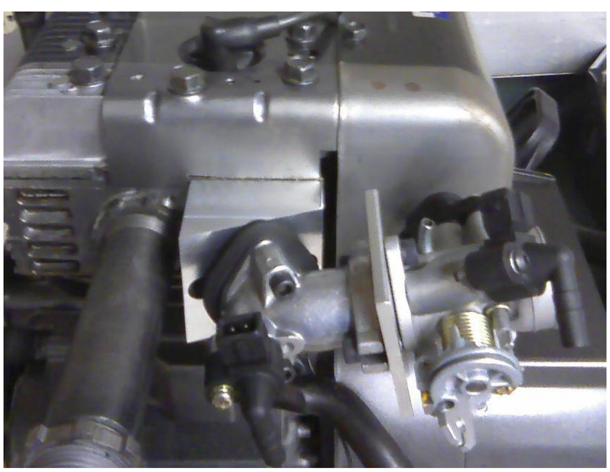

Figure 17 EFI Intake Manifold Assembly

A coupler was created to connect the engine to the dynamometer. This coupler had a one inch inside bore which slid over the crankshaft of the Briggs and Stratton engine. The bore had a broached keyway for a ¼ inch key. The coupler had a set screw that pressed down on the key to hold the coupler fixed on the crank shaft. It also had one end with four tapped holes which received screws to connect the dynamometer to the coupler. (See Figure 18 and Appendix F – Drawing Package of Modifications.)

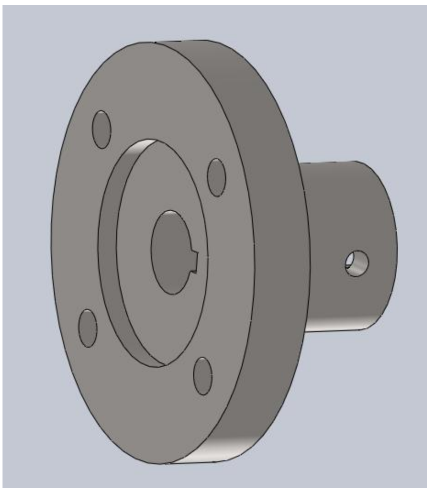

Figure 18 Engine to Dynamometer Coupler

An ECU is an electronic control unit. The two ECUs received signals from the sensors and then sent signals to control the engine. Sensors that sent a signal to the ECUs included: ambient air temperature sensor, crank position sensor, manifold absolute pressure (MAP) sensor, and throttle position sensor (TPS). See Figure 11 Engine Setup Schematic. The ECUs then sent signals to control the water injector and fuel injector. The ambient air temperature sensor, crank position sensor, manifold absolute pressure sensor, and the throttle position sensor were provided in the kit purchased from MBE Motorsports. The EFI Tune 2.25 software was provided by MBE Motorsports to control the ECU and tune the engine. This software is based on the MegaTune software.

To create a crank position signal a VR (variable reluctance) sensor was mounted on the engine a little above the coupler. A hole was tapped in the coupler for a ferrous screw. This screw provided the trigger tooth for the VR sensor. The VR sensor contained a magnet. When the screw head passed next to the magnet an electrical signal was created, which went to the ECU to provide crank position and RPM information. A mount was made to attach the VR sensor to the engine. See Figures 19 and 20 below.

Figure 19 VR Sensor

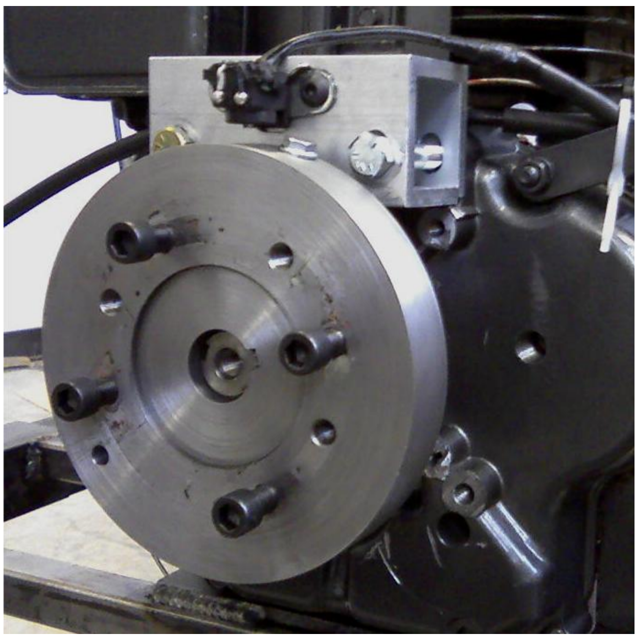

Figure 20 VR Sensor Mounted and Coupler with Trigger

The critical engine head geometry was modeled in a CAD system before the engine head was shaved. This CAD model was used to calculate the volume of the combustion chamber. Different amounts of material were shaved off the model to calculate the change in volume of the combustion chamber and the change to the compression ratio.

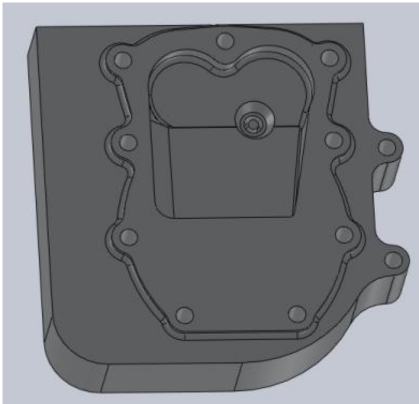

Figure 21 CAD Model of Engine Head

A thermocouple to measure exhaust gas temperature was placed in the exhaust pipe on the right in Figure 22. An O_2 sensor was also attached to the exhaust pipe to show the air/fuel ratio during initial tuning, on the left in Figure 22.

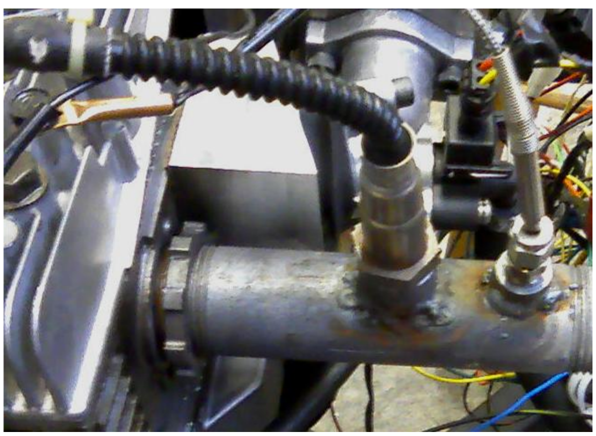

Figure 22 Thermocouple and O_2 Sensor on Exhaust Pipe

Figures 23-24 may help the reader visualize this particular four stroke engine setup.

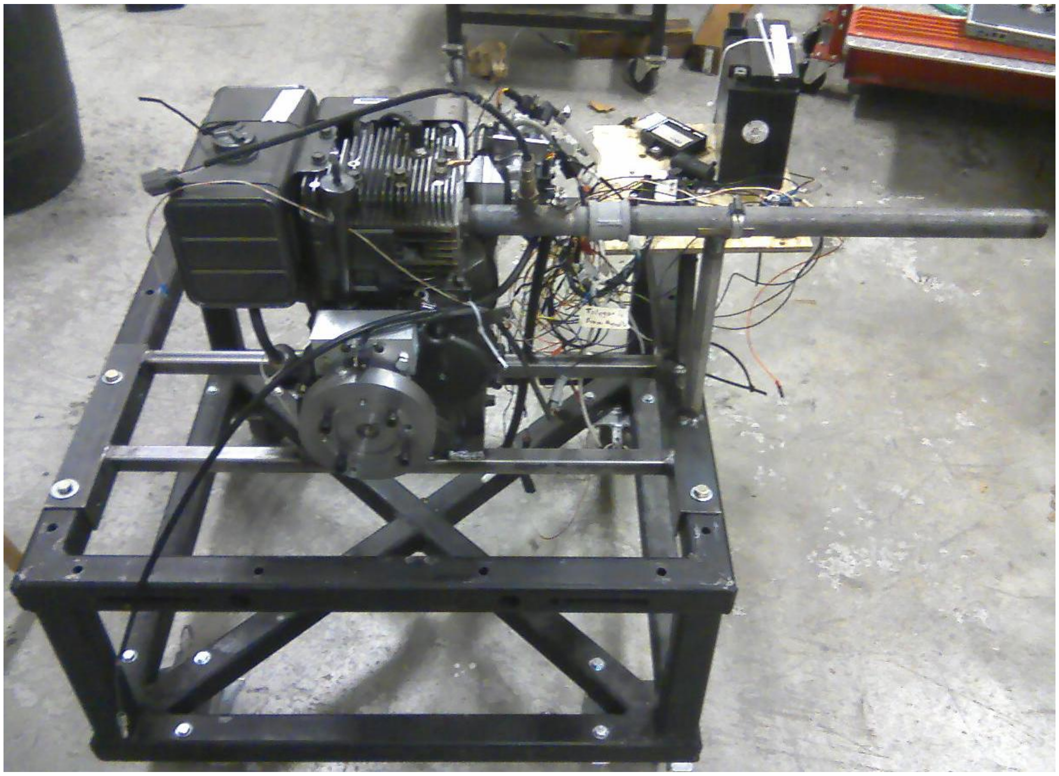

Figure 23 Engine on Test Stand

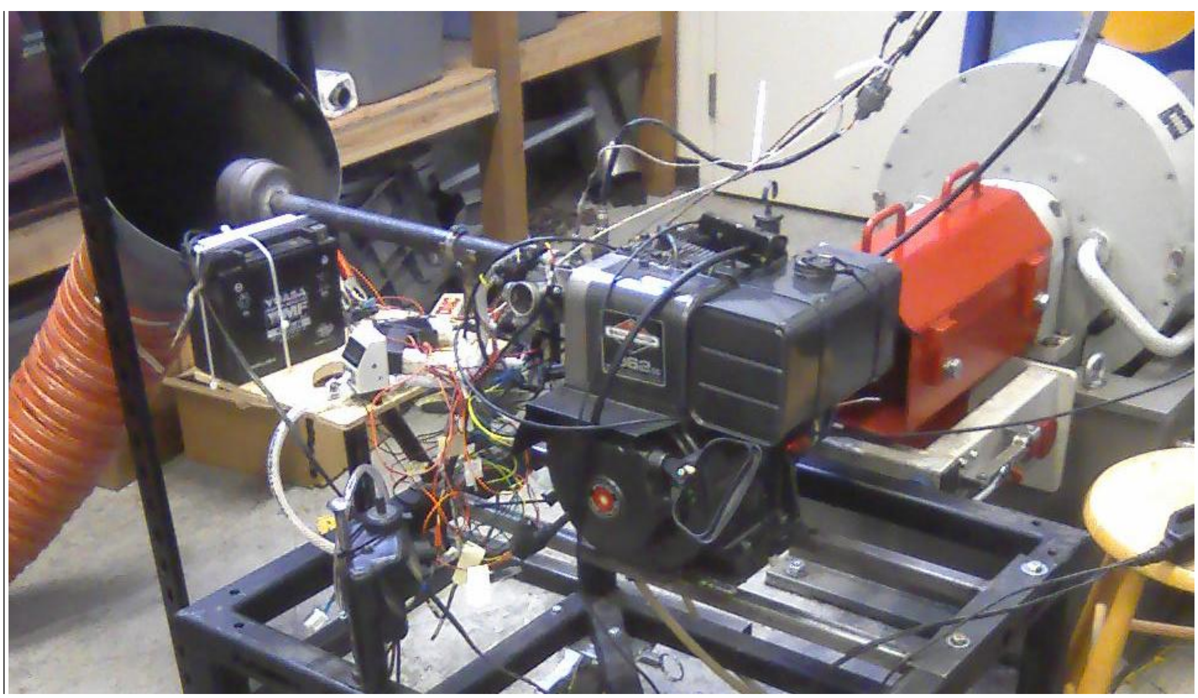

Figure 24 Engine and Dynamometer

The fuel was regular unleaded gasoline (no ethanol) with an octane rating of 87. This made the experiment more comparable to previous research, since this was what many researchers used.

2.2 Testing Equipment

A Borghi & Saveri eddy current dynamometer (Type FE 260 S) was used to test the engine with a SuperFlow XConsol SF-902 Data Acquisition system. This setup gave exhaust temperature to within one degree F, engine speed to one RPM, and torque (lb-ft) to one decimal place. Torque and power were reported as SAE corrected torque and power. English units were converted to the international system of units.

An Horiba automotive emission analyzer MEXA-584L was used to measure emissions. This emissions analyzer measured CO accurately within 0.03% volume or within 3% of the reading (whichever was larger). It measures HC (equivalent hexane) within 10 ppmvol or within 5% or reading (whichever was larger). It measured NO within 25 ppmvol or within 4% of reading (whichever was larger). No catalytic converter was used during these experiments.

To measure fuel consumption, a Max 710 Fuel Measurement System on which error was taken to be 1 gm. The fuel measurements were divided by time (measured with a hand held stop watch) to calculate fuel flow.

The experiments were done with distilled water to prevent the possibility of hard water build up. The water was measured using a graduated cylinder with marks to measure milliliters. It was assumed that one milliliter of water equaled one gram of water which at the ambient temperatures and pressures experienced during testing was an accurate assumption. These measurements were also divided by time (taken with a hand held stop watch) to calculate water flow. Human error using handheld stop watch was assumed to be ±0.2 seconds. Thus water/fuel ratios were on a mass bases.

Uncertainties are tabulate below.

Table 3 Uncertainties of Data

	Torque	Speed	Temp	NO	CO	HC	Water	Fuel	Time
Uncertainty	1 lb-ft	1 RPM	1° F	25 ppmvol or 4% of reading	0.03% vol or 3% of reading	10 ppmvol or 5% of reading	1 g	1 g	0.2 s

Included is a table of basic properties of water and gasoline. Density, dynamic and kinematic viscosity are from (19) and are reported at 15.6° C. Molecular formula, specific heat are from (20).

Table 4 Basic Properties of Water and Gasoline

	Density kg/m^3	Molecular Formula	Dynamic Viscosity N*s/m^2	Kinematic Viscosity m^2/s	Specific Heat kJ/kg*K	Heat of Vaporization kJ/kg
Water	999	H_2O	0.00112	1.12 e-6	4.18 @25°C	2257 @100°C
Gasoline	680	$C_nH_{1.87n}$	0.00031	4.6 e-7	2.4 @20°C	350 @25°C

3.0 Experiment, Results and Discussion

The engine was run at three different air/fuel ratios, with varied amounts of water injected and at two compression ratios. Torque, BSFC, exhaust temperature, as well as emission levels of NO, CO and HC were collected and reported. Emissions are reported on a brake specific basis. Graphs show averages of data taken over approximately one minute. Error bars represent the standard deviation of the data. The spreadsheet containing the data used for the various bar charts can be found in Appendix A. Equations used to calculate brake specific emissions are found in Appendixes C and D. For brake specific emissions, the hydrogen/carbon (H/C) ratio was assumed to be 1.87 .

A preliminary experiment was done on the engine with the fuel injection modifications, from which it was determined that the engine ran best (max torque) at approximately 2500 RPM and wide open throttle. Perhaps the temperature and pressure conditions in Moscow, Idaho are the reason max torque was not achieved at 3000 RPM as stated by the manufacturer. From this experiment it was determined to run the water injection tests at this RPM (2500) and throttle position (wide open). The average RPM varied up to 62 from the target RPM of 2500. The dynamometer used for testing was made for more powerful engines than the 10 HP Briggs and Stratton used. The dynamometer had difficulty holding the small engine at a constant speed. A small change in the load setting of the dynamometer (controlled by a PID system) caused a larger change in engine speed. In some cases this challenge resulted in data with standard deviations larger than would be preferred.

The stock engine spark timing was determined to be 26 degrees before top dead center. This same spark timing was used for all the experiments presented in this thesis.

Water addition can increase the O_2 levels in the exhaust (9). Because emissions analyzers usually calculate AFR based on O_2 levels, water addition can make AFR readings inaccurate. It was desired to run the tests with water and the tests without water at the same air/fuel ratios. To keep air/fuel ratios as close to target values as possible, three fuel maps were created (using the EFITune 2.25 software). These maps determined the amount of fuel provided to the engine per cycle, based on throttle position, RPM and air temperature. These maps were created with no water addition, and then were also used when water was added. It was assumed that the water injected did not significantly alter the air/fuel ratios. However, the air/fuel ratios are known with certainty only at 0 water/fuel ratio. Maps were made for three different target air/fuel ratios: 13.7 (rich), 14.7 (stoichiometric), and 15.7 (lean). Equivalence ratio the defined by equation 4, where (A/F) is air/fuel ratio.

$$\Phi = \frac{(\mathrm{A/F})\mathrm{stoichiometric}}{(\mathrm{A/F})}$$

Equation 5 Equivalence Ratio

Corresponding equivalence ratios (Φ) for the air/fuel ratios listed above are respectively: 1.07 (rich), 1.00 (stoichiometric) and 0.94 (lean). The actual air/fuel ratios varied up to 0.2 AFR (up to 0.015 Φ) from the target.

Target amounts of water were based on a water/fuel ratio (WFR) on a mass basis, e.g. 15g water per minute/ 100 g fuel per minute = 0.15 water/fuel ratio. Tests were run at each air/fuel ratio while incrementally increasing amounts of water (0, 0.15, 0.30, 0.45, 0.60, 0.75 water/fuel mass ratios). Actual water/fuel ratios varied up to 0.018 from the target. With the exception of one case (rich, compression ratio 6, target 0.75 WFR), where it was very difficult to tune the AFR, the error was 0.06 off of the target WFR. WFR ratio increases were stopped at 0.75 because it became very difficult to tune the water injection ECU for this amount and because engine performance had stopped making significant improvements with additional water.

Data at each combination of AFR and WFR were taken at the stock compression ratio of 6:1. Then 0.059 inches (1.5 mm) were shaved off of the bottom of the head, image of the engine head is shown Figure 21. Shaving the head decreased the volume of the combustion chamber to increase the compression ratio to 7:1. Tests at each combination of AFR and WFR were repeated.

Trends of torque, brake specific fuel consumption, temperature as well as emissions of NO, CO and HC are presented in the following subsections. The trends of power output are the same as the torque trends, because power is a function of torque and RPM, and all tests were run at approximately 2500 RPM. Efficiency trends are opposite the trends in BSFC. Brake specific fuel consumption is the amount of fuel consumed per unit of energy received from the engine (grams/kW-h). Hence, if less fuel is used per kW-h produced, then

BSFC has diminished and efficiency has increased (equation 3). Trends as AFR changes from lean to rich are known and can be found in *An Introduction to Combustion* by Turns (2). They will not be thoroughly discussed here.

3.1 Torque

At lean, stoichiometric and rich operating conditions, torque output tends to increase as water is increased. This improvement is most noticeable from 0 WFR to 0.30 WFR at a 6:1 compression ratio (CR). Water injection at a 7:1 compression ratio does seem to increase torque but the increase is much less significant. An increase in compression ratio increased the torque in almost all cases. The maximum improvement was seen at stoichiometric conditions; changing from 0 WFR and CR6 to 0.75 WFR and CR7 improved torque by 65%.

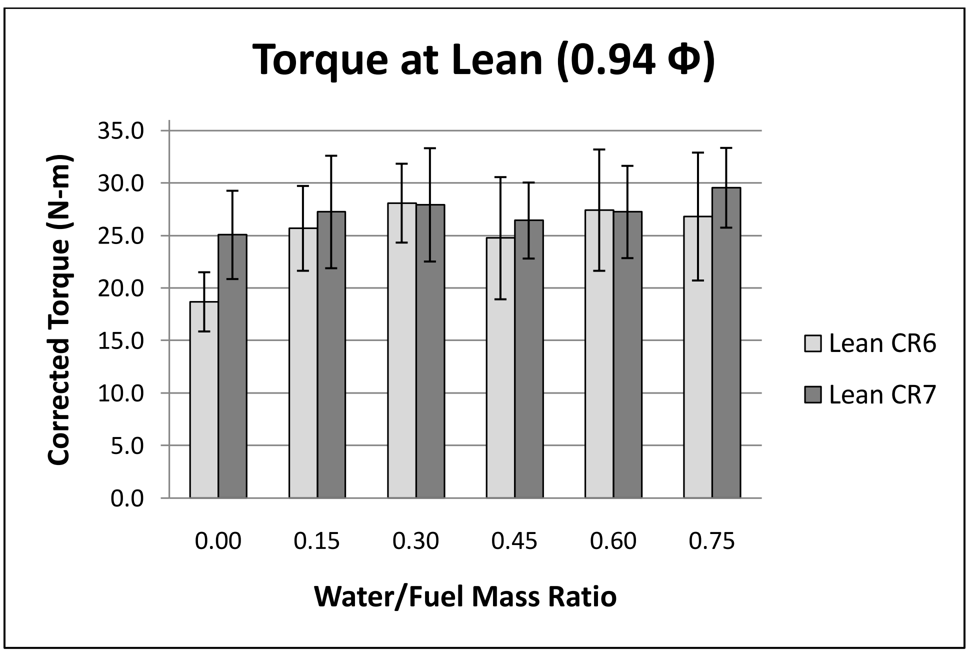

Figure 25 Torque at Lean Conditions

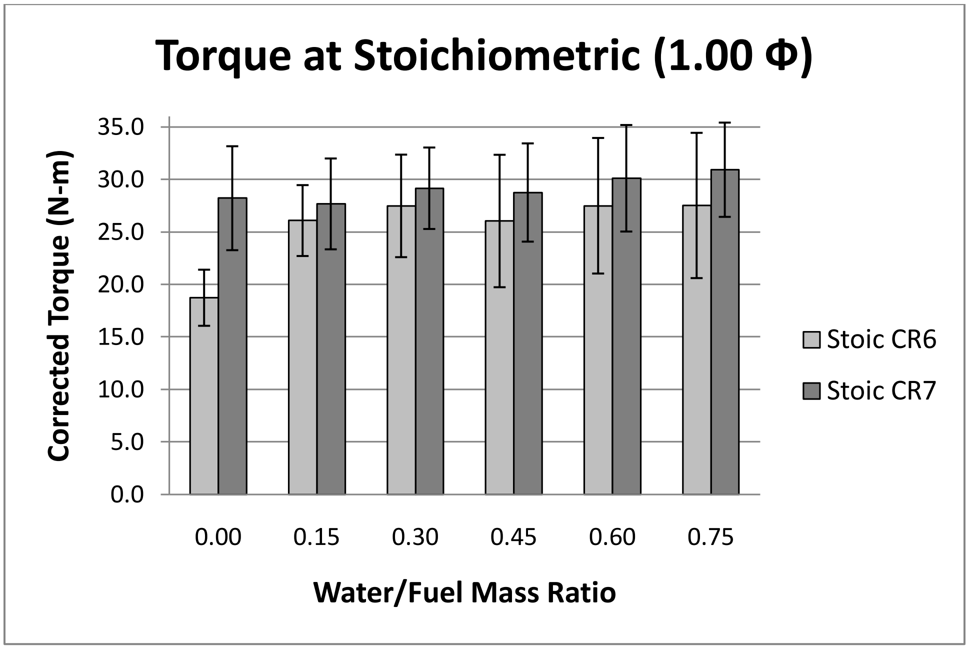

Figure 26 Torque at Stoichiometric Conditions

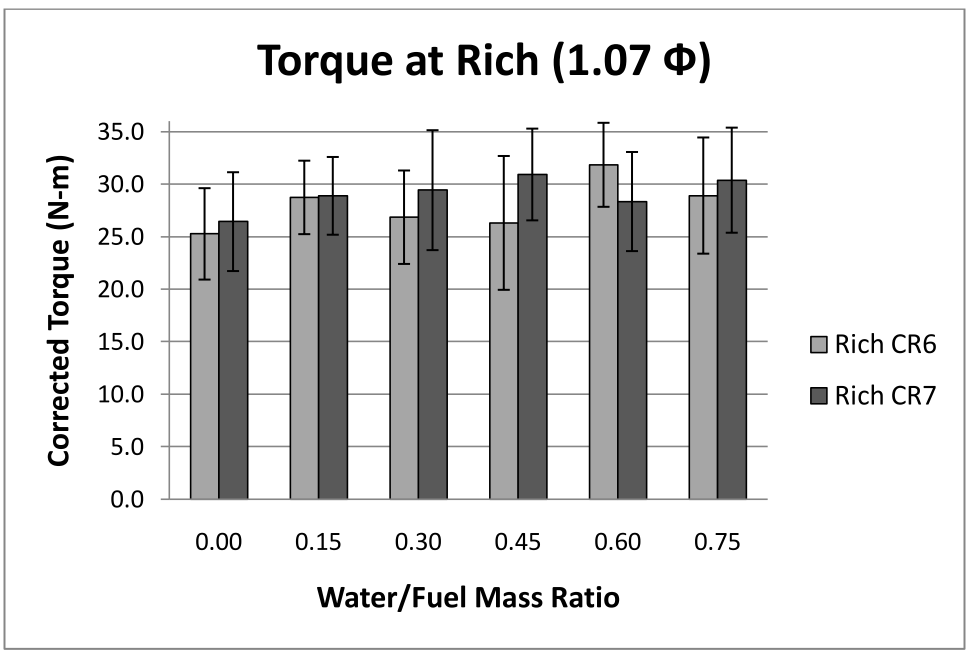

Figure 27 Torque at Rich Conditions

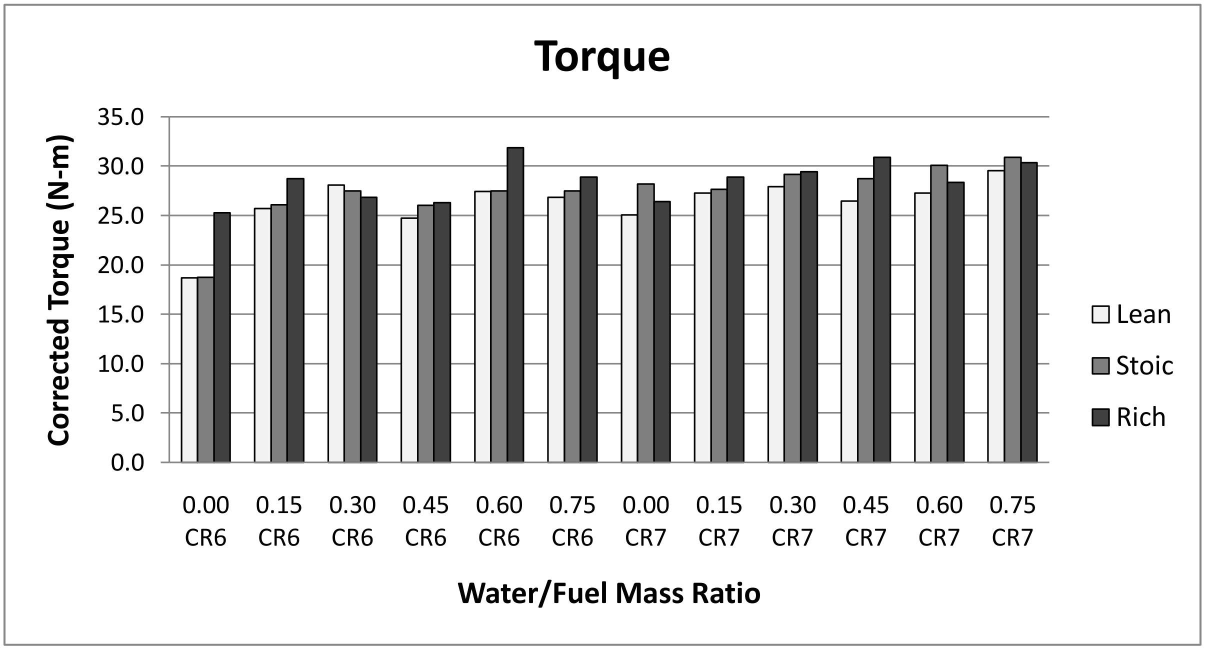

Figure 28 Torque at All Conditions

Water injection may increase the torque output of the engine in multiple ways: by reducing compression work, by increasing the work done during the power stroke, by reducing temperature which reduces heat loss, and by increasing the burn rate. Increased compression ratio also increases torque.

Water injection may reduce compression work and increase work out. When the water is injected into the intake manifold and drawn into the engine it is still a liquid, though it is in very small droplets. When the air-fuel-water mixture (the charge) is compressed, its temperature increases. However, the water droplets absorb some of this heat and are evaporated. Because some of the heat produced by compression goes to heating and evaporating the water, the charge is cooler and the pressure the charge reaches is reduced. This reduced pressure during the compression stroke reduces the work the engine must do to compress the charge. Steam expands more than air when heated. Thus, when heat is released in the charge during combustion the increase in pressure is greater due to the water vapor in the charge. The difference in pressure during the compression stroke and the power stroke is greater, which increases the torque and power output of the engine. This effect is most noticeable at lean and stoichiometric operating conditions. Improvements in torque are less at rich conditions, because extra fuel can have the same type of effect as the water injection in this case. At part load the water may not vaporize until combustion occurs.

The fact that water reduces the temperature can also increase torque. The flow of heat is driven by a temperature difference; the greater the temperature difference, the greater the flow of heat. Heat flow out of the engine reduces the temperature and consequently the pressure inside the engine. This reduces power output of the engine. Heat flows out of the engine due to the high temperatures inside the engine, which is greatest at the peak temperature. Because water reduces the peak temperature, heat lost during the cycle is reduced, resulting in higher pressure during the power stroke and more torque, see also Equation 3 Brake Work Produced Per Cycle .

According to Tsao and Wang (4), water may also increase the burn rate. This may also increase torque since a peak pressure could be reached sooner. The increased burn rate could be due to increased hydrogen and oxygen radicals in the charge, which come from increased amounts of H_2O. These radicals facilitate burning. An increased burn rate gives more crank angle degrees with a higher pressure, which increases the torque output. Spark timing may need to be changed to take full advantage of an increased burn rate (see recommendations for future work in Section 5).

Torque is also improved by an increase in compression ratio. With an increased compression ratio, the pressure after the compression stroke is higher. If the same amount of heat is added by combustion, then the pressure increases more than it would with a lower compression ratio. Higher pressure results in more torque. A visualization using the ideal

Otto cycle on a P-V diagram helps to explain this improvement. See Figure 29.

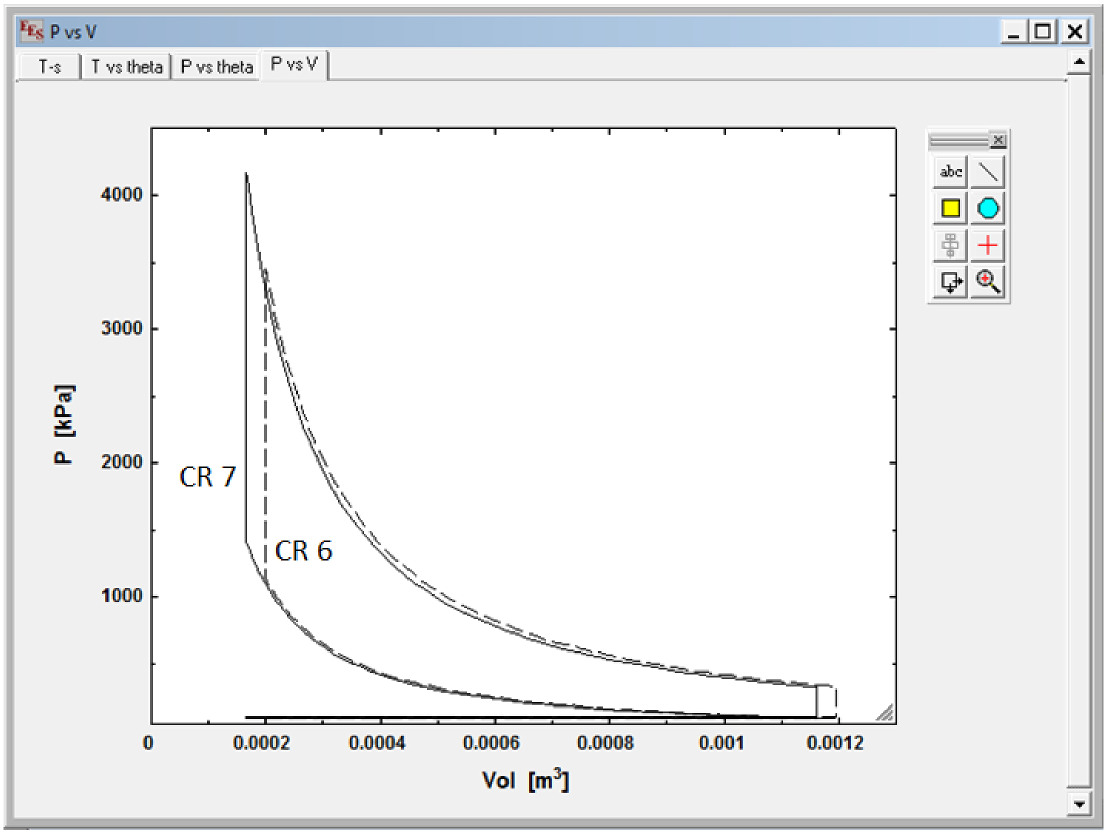

Figure 29 P-V Diagram at Two Compression Ratios

Note that this graph is intended to be helpful in visualizing changes in pressure and volume. It represents an ideal Otto cycle, not the actual cycle of the test engine

3.2 Brake Specific Fuel Consumption

For lean and stoichiometric operation, BSFC tends to be improved (lowered) by water injection. The amount of water seems to be less important than the fact that water is present once the water/fuel mass ratio is above 0.15. The presence of water seems to be less important at a compression ratio of 7:1 than it is at a compression ratio of 6:1. For lean and stoichiometric operation, an increase in compression ratio improved BSFC. Equations used to do error analysis on BSFC are shown in Appendix E. This drop in BSFC (increase in efficiency) tends to agree with the research of Tsao and Wang (4) but tends to disagree with Peters and Stebar (3). This could be because Tsao and Wang used a similar engine (an L head Wisconsin Model AENL) and similar compression ratios (5.67 and 7.5), while Peters and Stebar used a CFR engine and were operating at higher compression ratios (8.1 and 9.0). The CFR engines were made to test octane rating, but are not used in regular applications.

During rich operation, BSFC tends to be improved by water injection, but the improvement is much less significant than at lean and stoichiometric operation. Increased compression ratio also tends to improve BSFC at rich operation. The maximum improvement was seen at stoichiometric conditions; changing from 0 WFR and CR6 to 0.75 WFR and CR7 reduced BSFC by 39%.

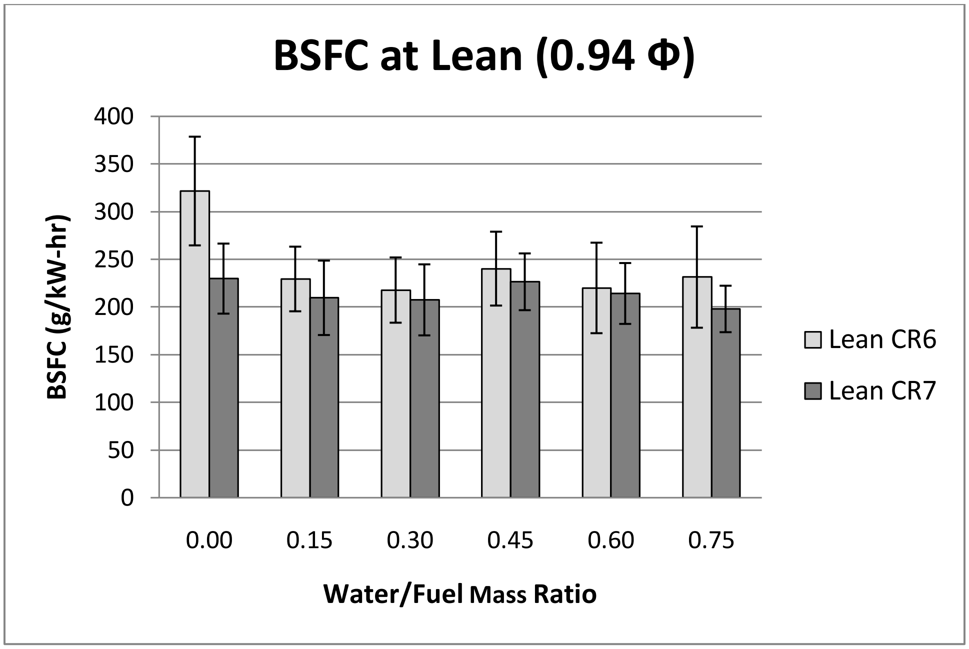

Figure 30 BSFC at Lean Conditions

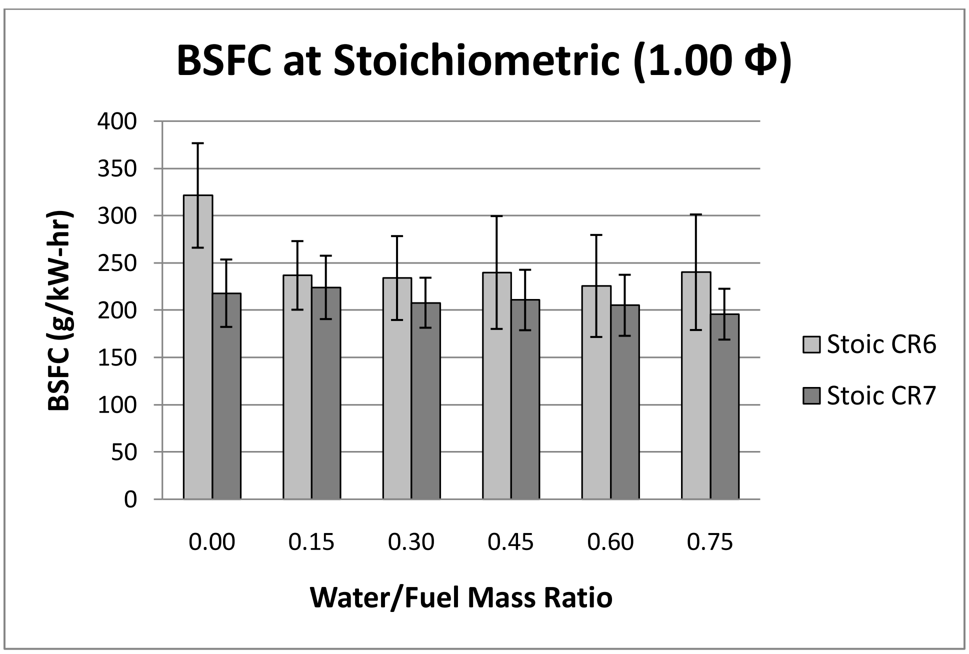

Figure 31 BSFC at Stoichiometric Conditions

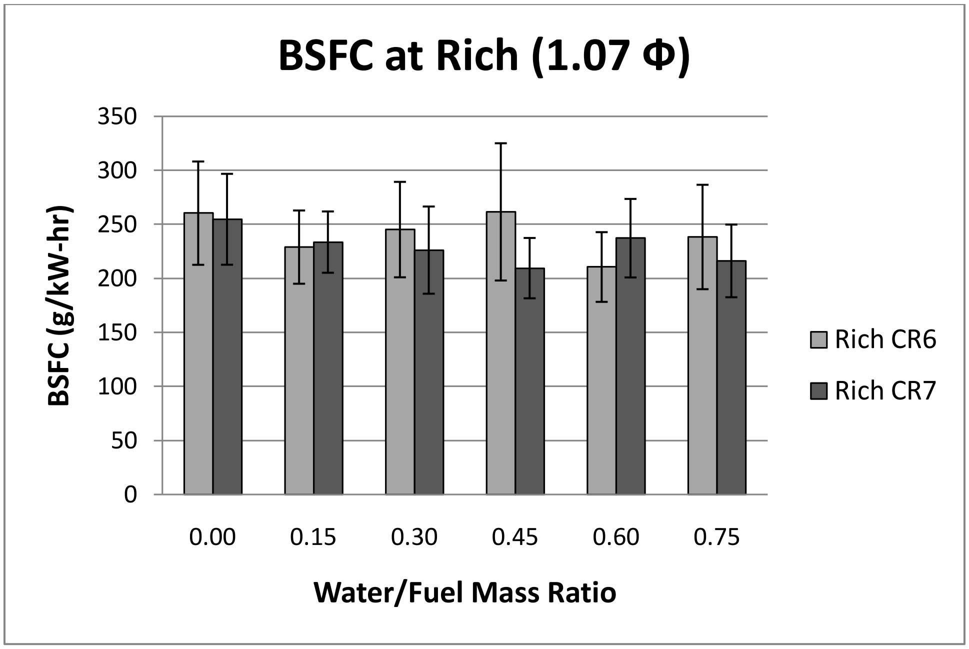

Figure 32 BSFC at Rich Conditions

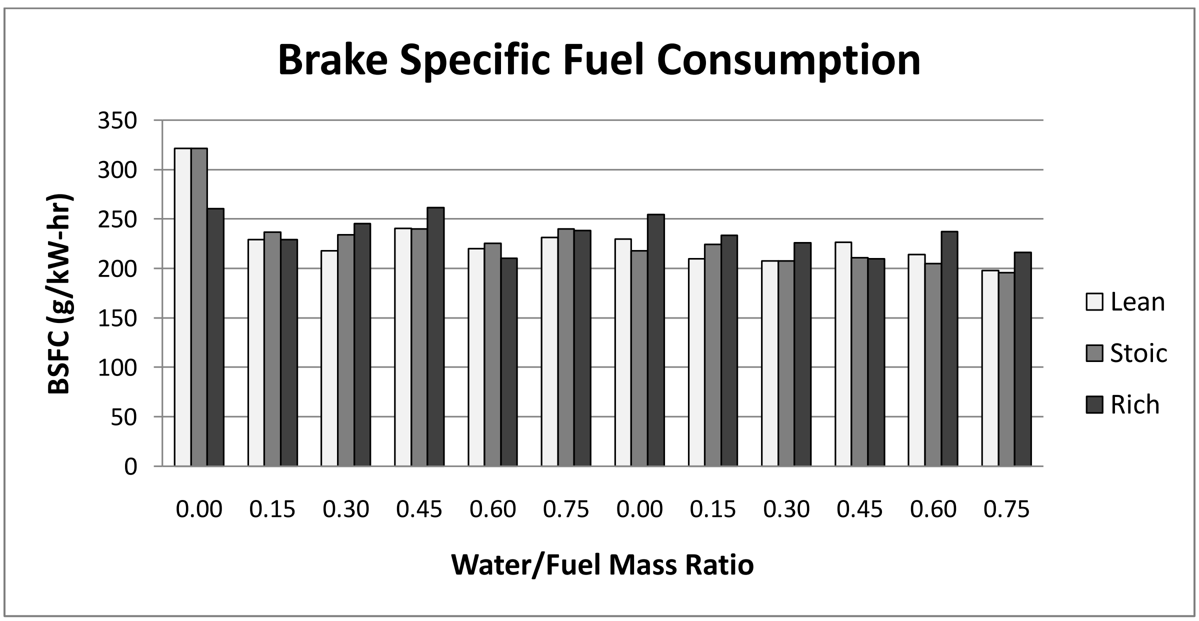

Figure 33 BSFC at All Conditions

Water can improve BSFC for the same reasons it may improve torque. If torque and power are increased and the same amount of fuel is used, then the BSFC was reduced, and efficiency was increased. Hence, water injection may decrease the BSFC of the engine in multiple ways: by reducing compression work, by increasing the work done during the power stroke, by reducing temperature to reduce heat losses and by increasing the burn rate. Increased compression ratio also increases torque and decreases BSFC. See Section 3.1 on torque for further explanation.

3.3 Exhaust Temperature

Water injection tends to reduce the exhaust temperature at lean, stoichiometric and rich conditions. The amount of water seems to be less important than the fact that water is present. Exhaust temperature is also reduced with an increase in compression ratio. The maximum temperature reduction is at stoichiometric, changing from 0 WFR and CR6 to 0.30 WFR and CR7 reduced the temperature by 10%.

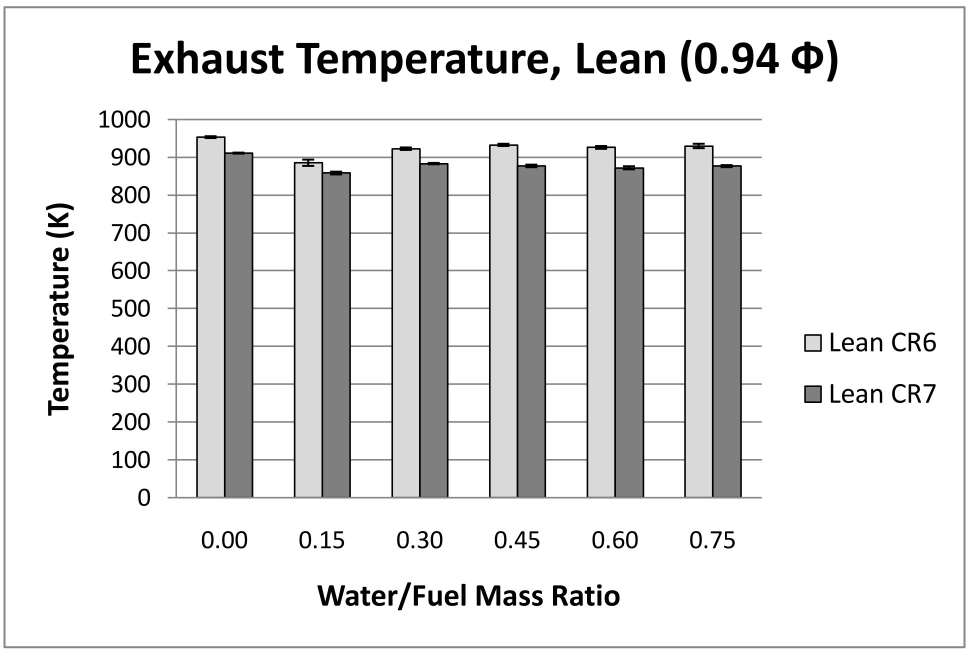

Figure 34 Exhaust Temperature at Lean Conditions

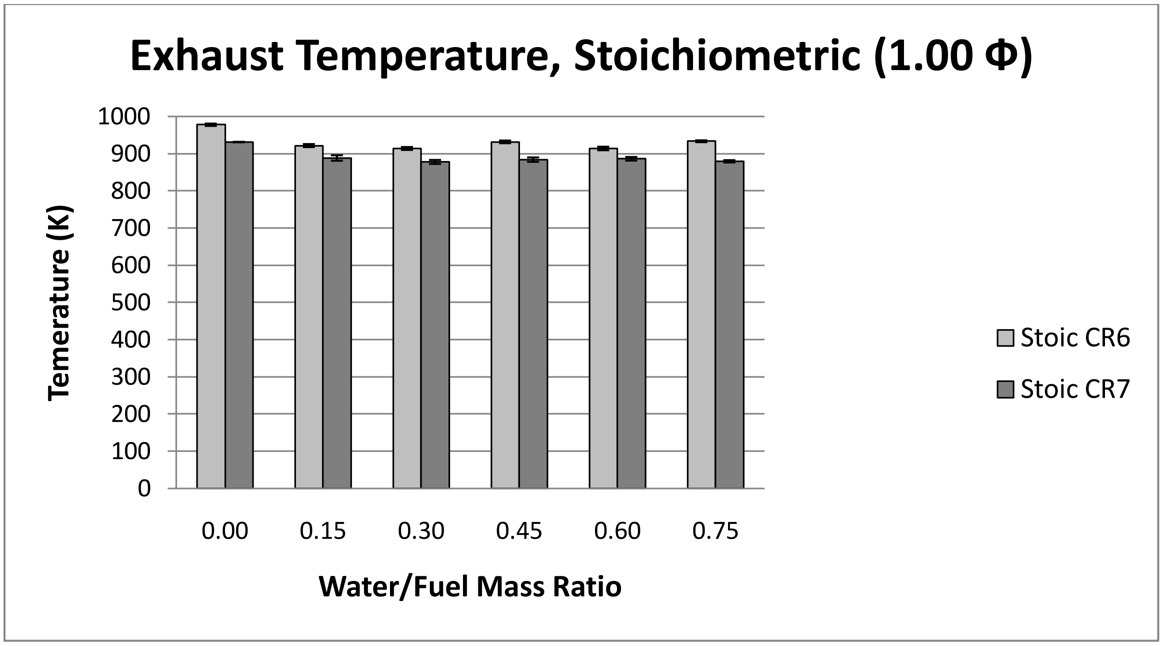

Figure 35 Exhaust Temperature at Stoichiometric Conditions

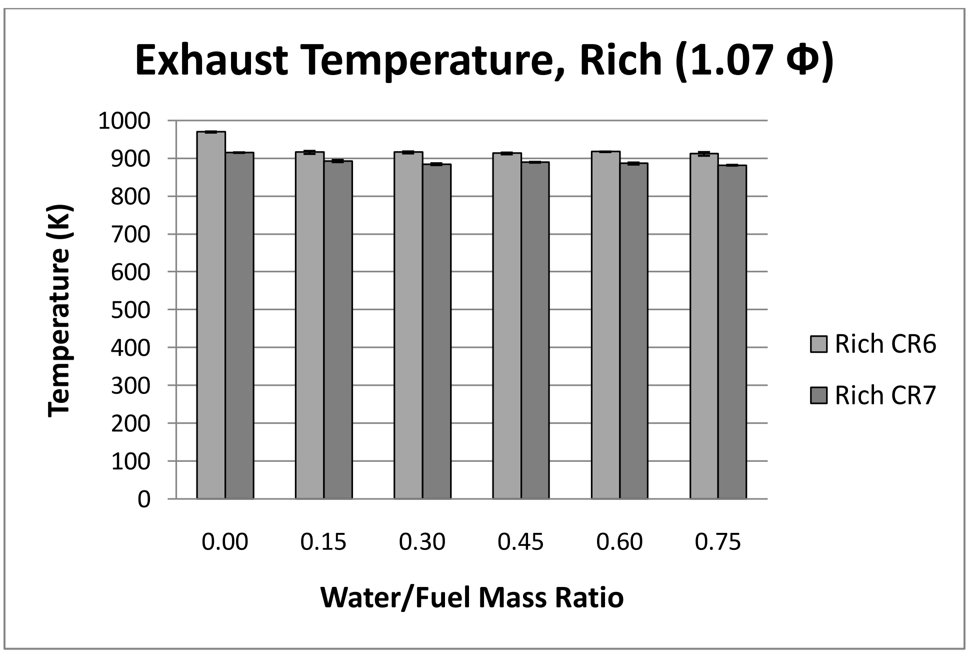

Figure 36 Exhaust Temperature at Rich Conditions

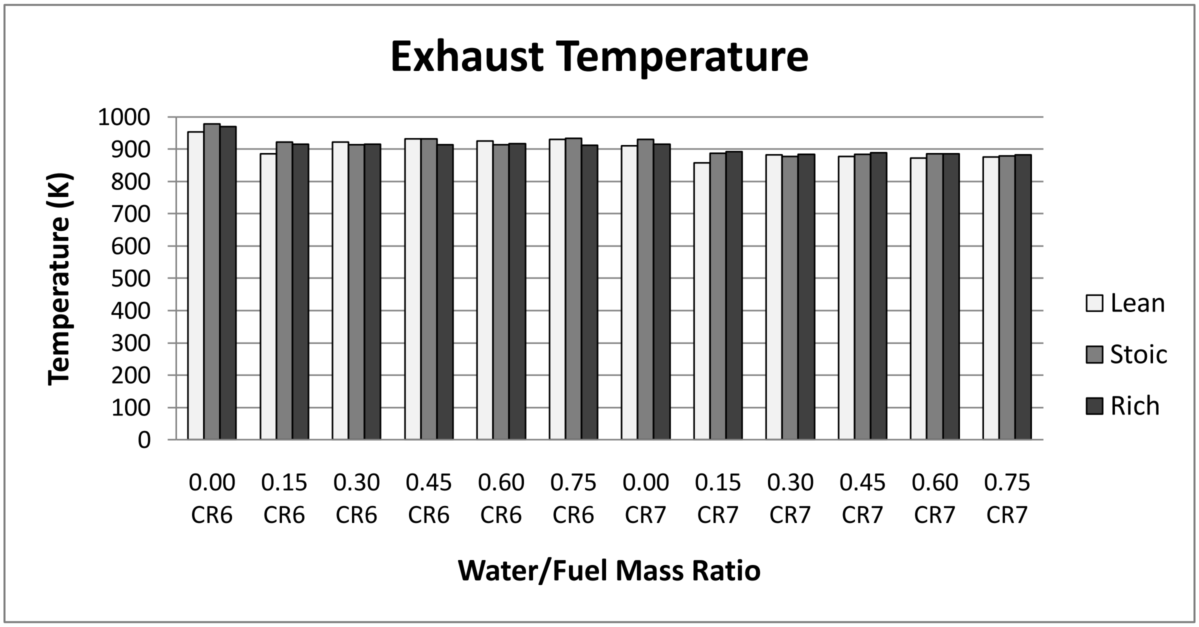

Figure 37 Exhaust Temperature All Conditions

We would naturally expect water injection to reduce exhaust temperature. Water has a very high latent heat of vaporization. Because the water evaporates during the cycle, the final temperature is reduced.

Increased compression ratio also decreases the exhaust temperature. The effect of an increase in compression ratio can be visualized on a T-s diagram. The peak temperature of the cycle increases with an increase in compression ratio, but at the next point, exhaust, the temperature decreases with an increase in compression ratio.

3.4 BSNO Emissions

Brake specific nitric oxide (BSNO) emissions at lean, stoichiometric and rich operating conditions tend to decrease as water is increased. This is to be expected since the temperature is reduced.

An increase in compression ratio without water injection tends to lower the level of BSNO. With water injection (WFR of 0.15 or above) an increase in compression ratio tends to lower the BSNO output. On the whole an increase in compression ratio in combination with water injection tends to lower BSNO. Standard deviations of the NO data were on the same order of magnitude as the error of the emission analyzer. Note that the reported emissions are without a catalytic converter. BSNO was calculated from ppmvol NO and power (see Appendixes A and D). Calculating the error for BSNO was not feasible. However, one may view the error of ppmvol NO (Appendix B) and the error of the torque (subsection 3.1).

The maximum improvement (when changing both WFR and CR) was seen at lean operation, changing from 0 WFR and CR6 to 0.75 WFR and CR7 lowered BSNO by 78%.

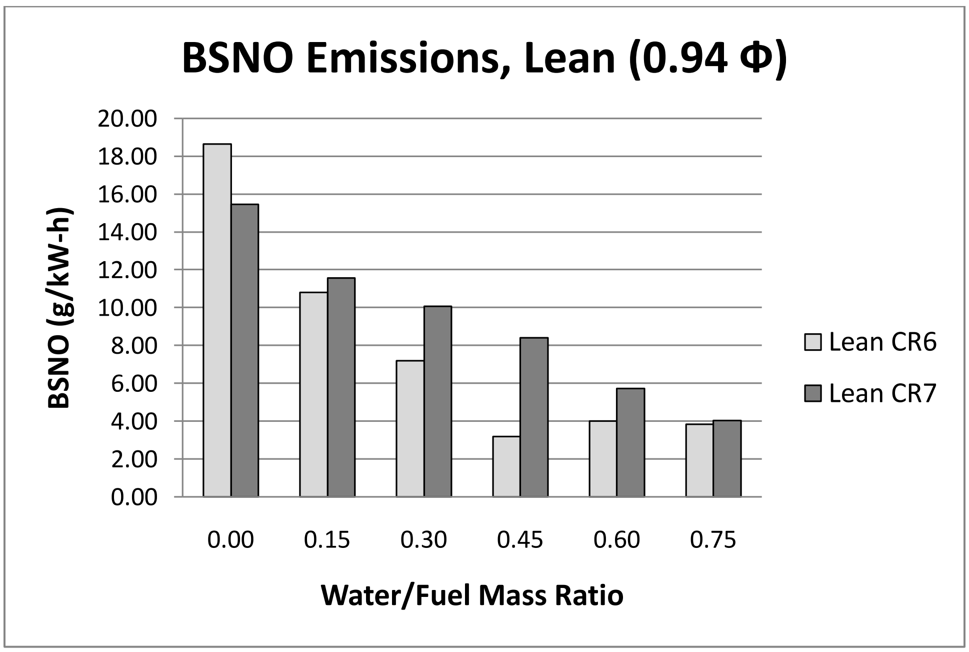

Figure 38 BSNO Emission at Lean Conditions

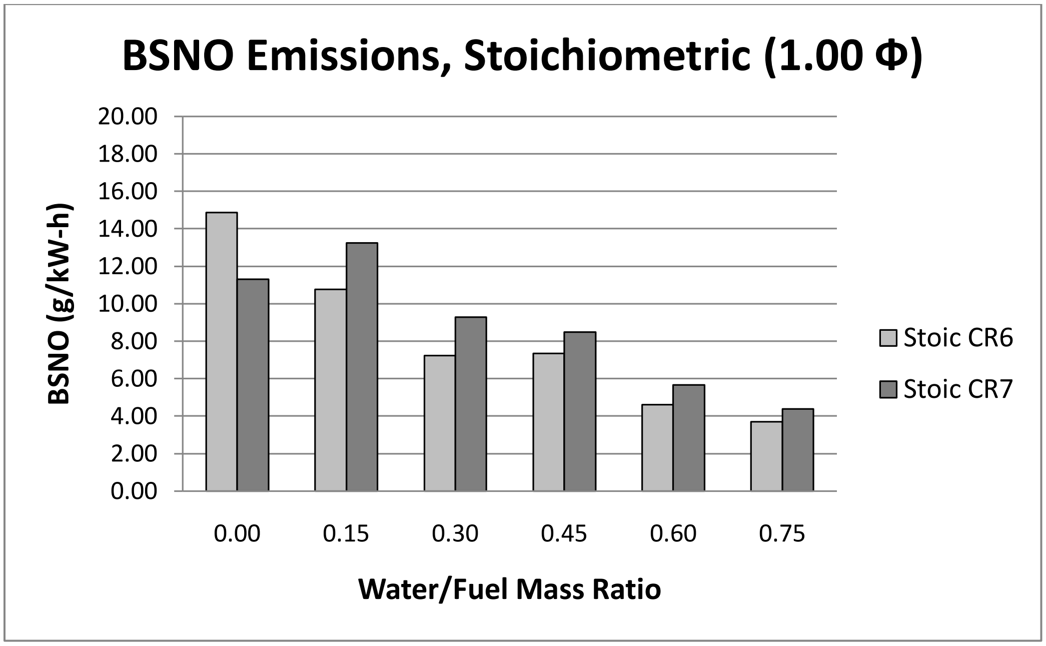

Figure 39 BSNO Emission at Stoichiometric Conditions

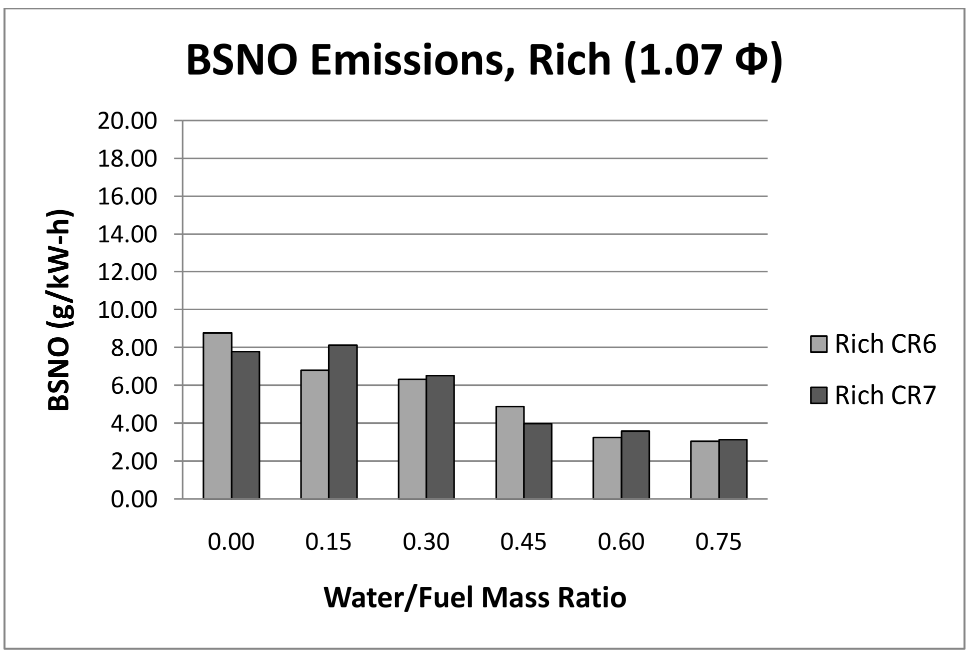

Figure 40 BSNO Emission at Rich Conditions

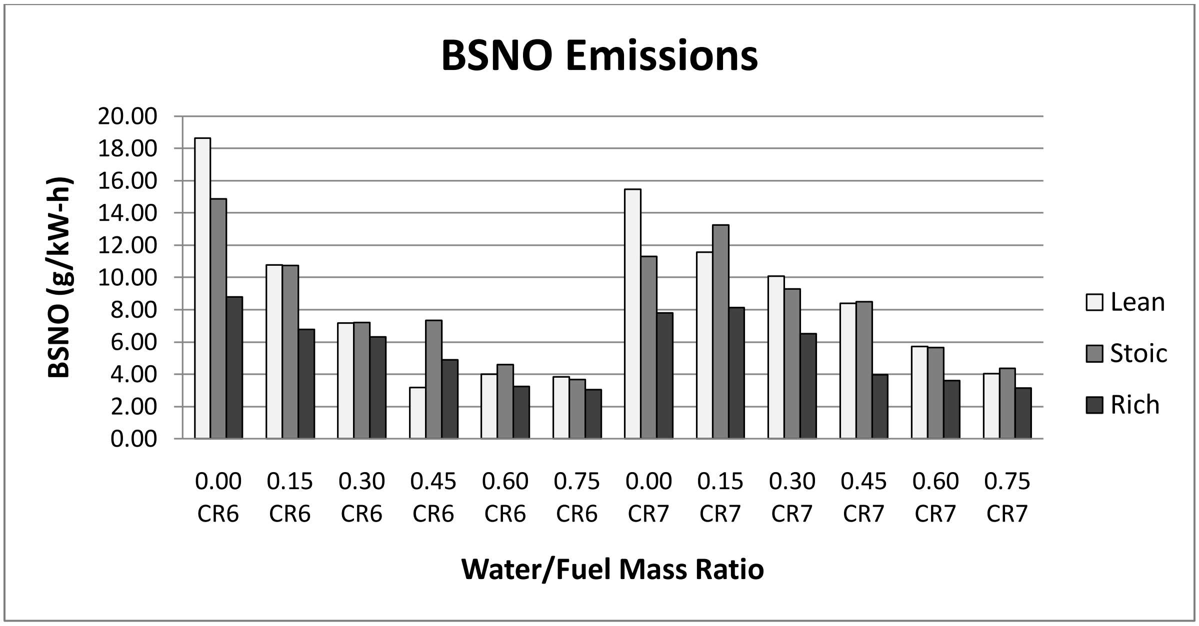

Figure 41 BSNO Emissions All Conditions

NO_X (NO and NO_2) formation increases with temperature and more rapidly at temperatures above 1,800 K (2780 F) (2). Due to its high latent heat of vaporization, water is very effective at reducing the temperatures in the engine, thus less energy is available to break up the triple N-N bond, which is the first step in NO formation via the Zeldovich or thermal mechanism. The Zeldovich mechanism is not the only mechanism for NO formation, but it is a major contributor and it is the mechanism most affected by temperature. Water has much less effect at rich conditions. With extra fuel present, oxygen is taken up by the carbon to form CO and CO_2 ; less is available for NO formation. Also, the extra fuel can have the same effects as water to cool the engine, thus lowering NO.

An increase in compression ratio increases the peak temperature that the cycle will reach. At this higher temperature, more NO forms. Thus, increasing compression ratio tends to increase the NO emissions. During lean and stoichiometric operations, at a 0 water/fuel ratio, the torque was so improved by an increase in compression ratio that although the ppm of NO was increased (see Appendix B), the BSNO was decreased.

3.5 BSCO Emissions

At rich operating conditions an increase in water from 0 WFR to 0.30 WFR can be effective at lowering brake specific carbon monoxide (BSCO) emissions. This agrees with previous research (3, 4, 6, 9). Further increases in WFR at rich operation seem not to improve BSCO. Higher compression ratios may tend to increase percent CO emissions, at least at rich conditions where CO is more prevalent (see Figure 56 %CO Emissions at Rich Conditions in Appendix B). However, the increased torque that comes with increased compression ratio often offsets the increase in percent CO for a smaller effect on BSCO. Water injection in combination with increased compression ratio can lower BSCO emissions. CO is much more prevalent at rich conditions because there is not enough oxygen for complete oxidation of the fuel to form CO_2.

At stoichiometric conditions increasing water from 0 WFR to 0.15 WFR can also lower BSCO emissions, but further increases tend not to improve BSCO emissions. Increased compression ratio may have only a small effect BSCO emissions at stoichiometric. However, this in not conclusive. To know for sure the effect of increased compression ratio on CO a more precise method of tuning AFR would be needed, since CO levels are very sensitive to AFR. As can be seen in Appendix A the AFR is not exact between compression ratios.

At lean operation the CO emissions are already very low (since there is plenty of oxygen for CO_2 formation). At lean conditions, the percent CO seems to be only slightly affected by water injection, but because the water increased the power, the BSCO may be lowered by water injection. From Figure 42 it appears that an increase in compression ratio may lower CO emissions during lean operation. This is not conclusive however, since the air/fuel ratios during the testing were not exactly on target. CO emissions are very sensitive to air/fuel ratio. It can be seen from the data in Appendix A that the AFR for lean CR7 was more lean (higher AFR) than the AFR for lean CR6. Thus it is inconclusive whether the change in CO emission for lean operation is due to increased CR or from the slight change in AFR. Further testing with more precise tuning would be required to definitively determine the effect of increase compression ratio at lean operation.

Except when the standard deviation of percent CO was very small (< 0.001) it was greater than the error of the emissions analyzer, see Appendix A. Note that the reported emissions are without a catalytic converter. BSCO was calculated from percent CO and power (see Appendixes A and D). Calculating the error for BSCO was not feasible. However, one may view the error of percent CO (Appendix B) and the error of the torque (subsection 3.1).

The maximum percent improvement was seen at stoichiometric conditions; changing from 0 WFR and CR6 to 0.15 WFR and CR7 lowered BSCO emission by 89%. The maximum number improvement was seen at rich conditions; changing from 0 WFR and CR6 to 0.15 WFR and CR7 changed BSCO from 67.2 to 44.3 g/(kW-h).

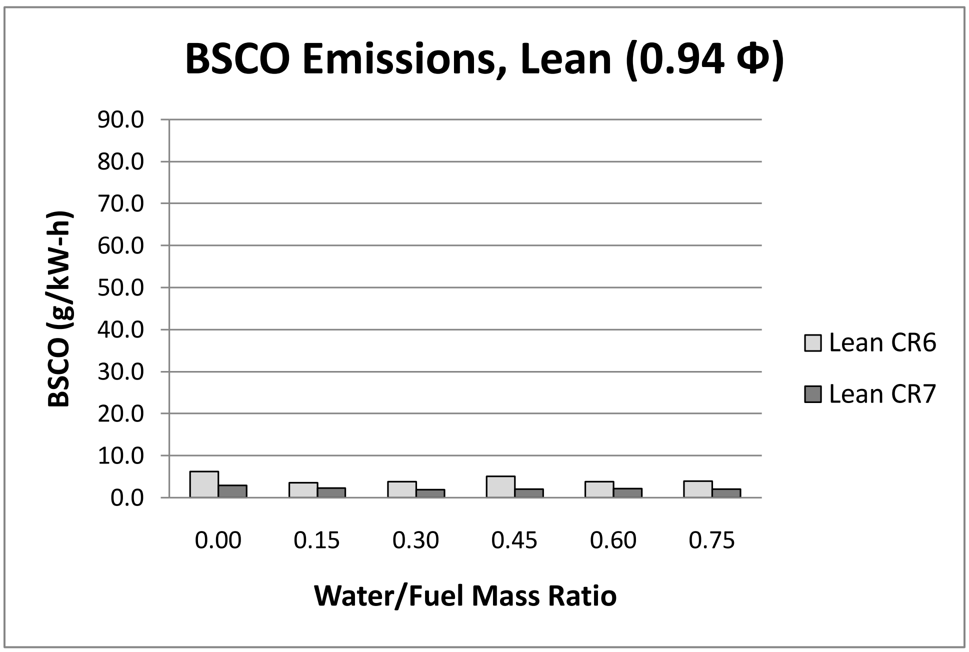

Figure 42 BSCO Emissions at Lean Conditions

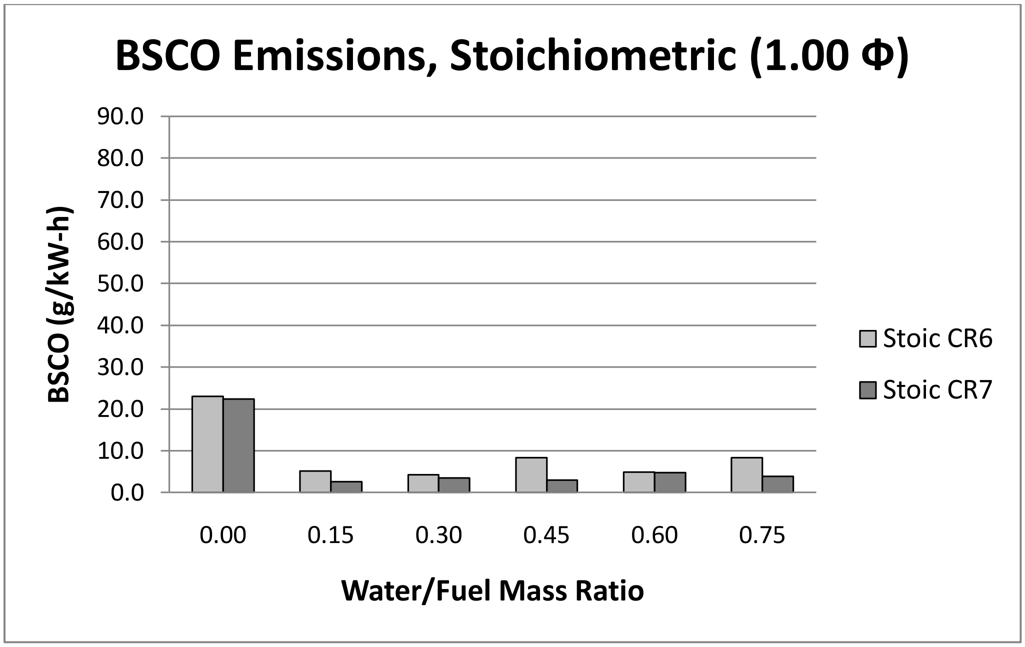

Figure 43 BSCO Emissions at Stoichiometric Conditions

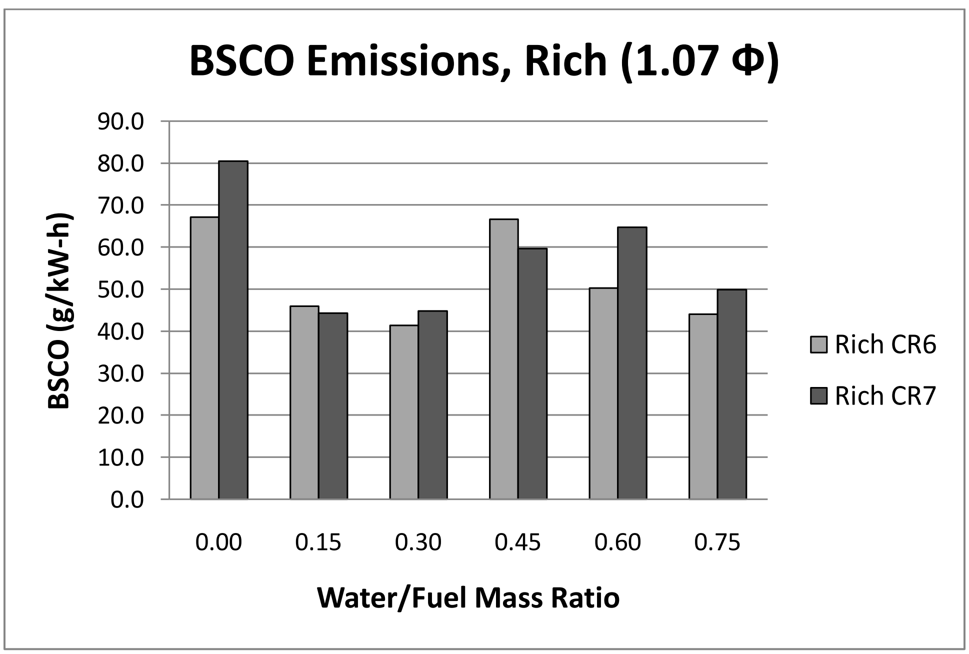

Figure 44 BSCO Emissions at Rich Conditions

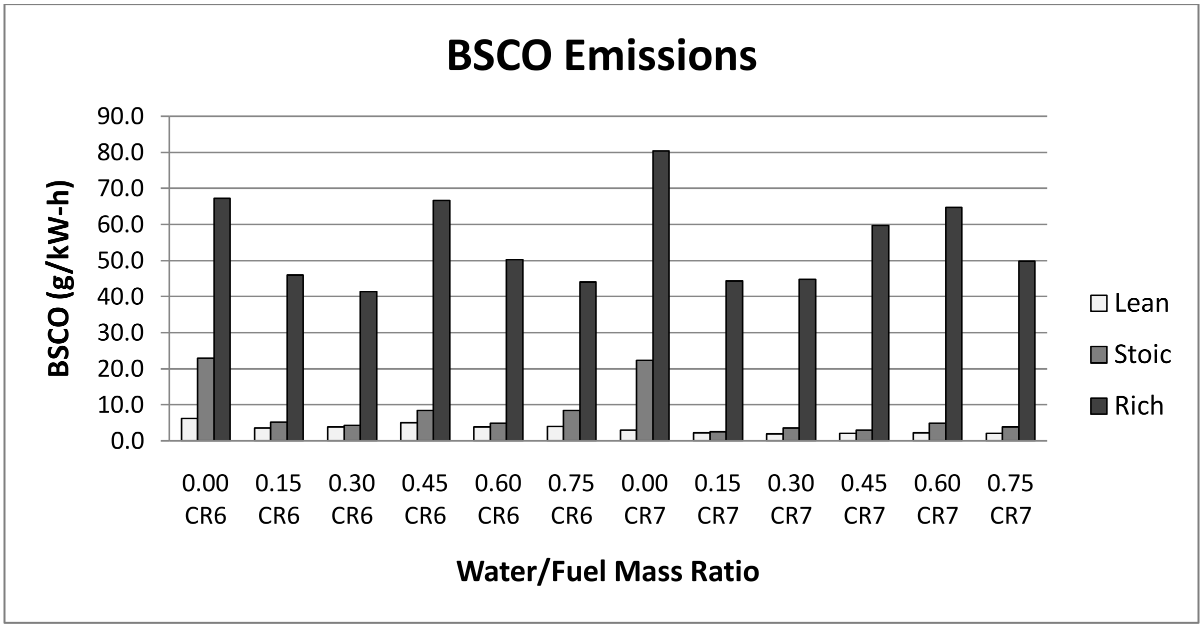

Figure 45 BSCO Emissions All Conditions

Additional water likely increases the number of hydrogen and oxygen radicals, which would promote a more complete combustion. This could be one reason that water may lower BSCO. For rich conditions the decrease in CO levels with water injection could also be attributed to the water-gas shift reaction, in which CO and H_2O shift to form CO_2 and H_2. The water-gas shift reaction is explained in detail by Turns (2) pgs 47-51.

Percent CO tends to increase with increased compression ratio at rich conditions. It is shown in chapter 2 of Turns (2) that CO_2 dissociation increases significantly with increases in temperature. Increased compression ratio increases the peak temperature of the cycle which may cause CO_2 to dissociate to CO and O_2. However, increase torque can offset the increase in CO, resulting in a lower change in BSCO. Water addition reduces temperature and would thus decreased dissociation. This could also be a reason that water injection tends to lower CO emissions.

It is inconclusive what effect an increased compression ratio might have on CO emissions when running at lean and stoichiometric conditions.

3.6 BSHC Emissions

Brake specific hydrocarbon emissions may tend to increase with increasing water. From the data collected it is difficult to determine if there is a BSHC trend with increase compression ratio. The standard deviations of the ppmvol HC data are on the same order of magnitude as the error of the emission analyzer. Also note that the reported emissions are without a catalytic converter; we could well assume that emissions would be much lower with the use of a catalytic converter. BSHC was calculated from ppmvol HC and power (see Appendixes A and D). Calculating the error for BSHC was not feasible. However, one may view the error of ppmvol HC (Appendix B) and the error of the torque (subsection 3.1).

The maximum detriment (when changing both WFR and CR) was seen at rich operation; changing from 0 WFR and CR6 to 0.30 WFR and CR7 increased BSHC by 45%.

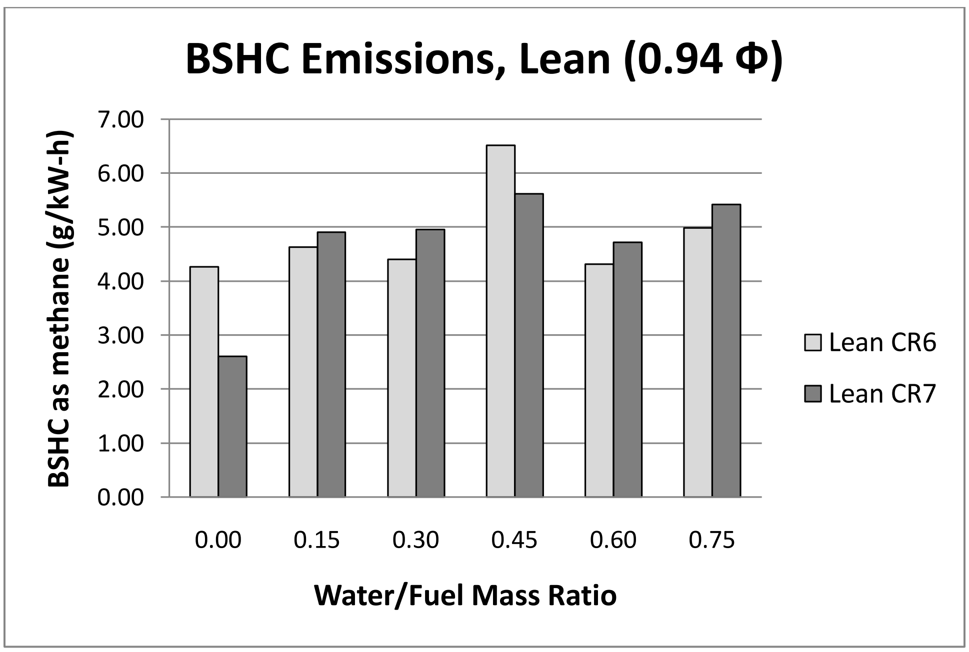

Figure 46 BSHC Emissions at Lean Conditions

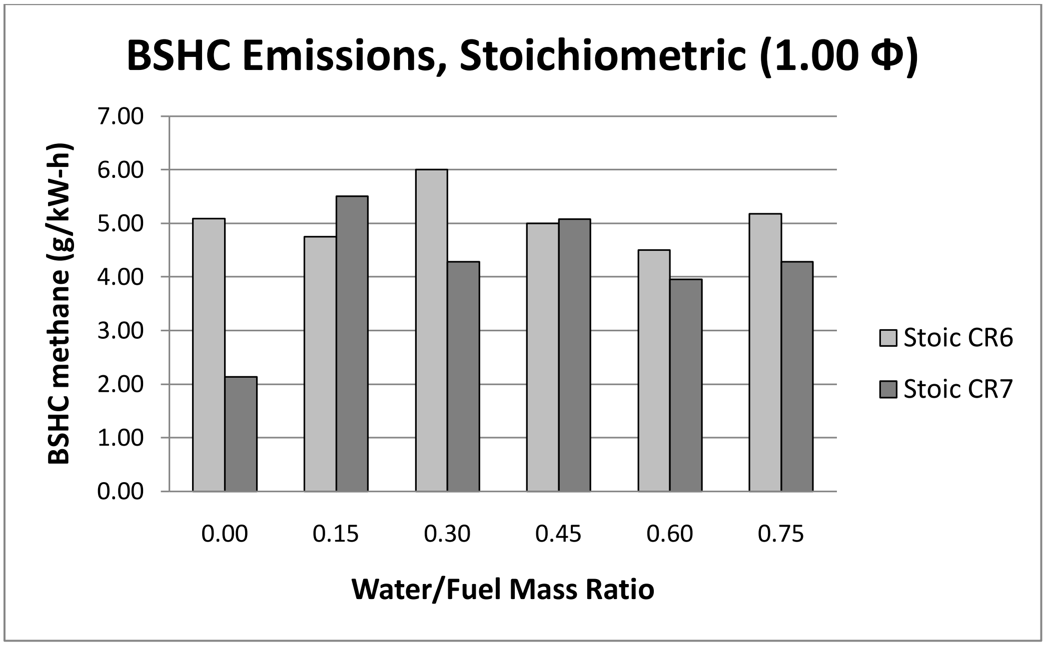

Figure 47 BSHC Emissions at Stoichiometric Conditions

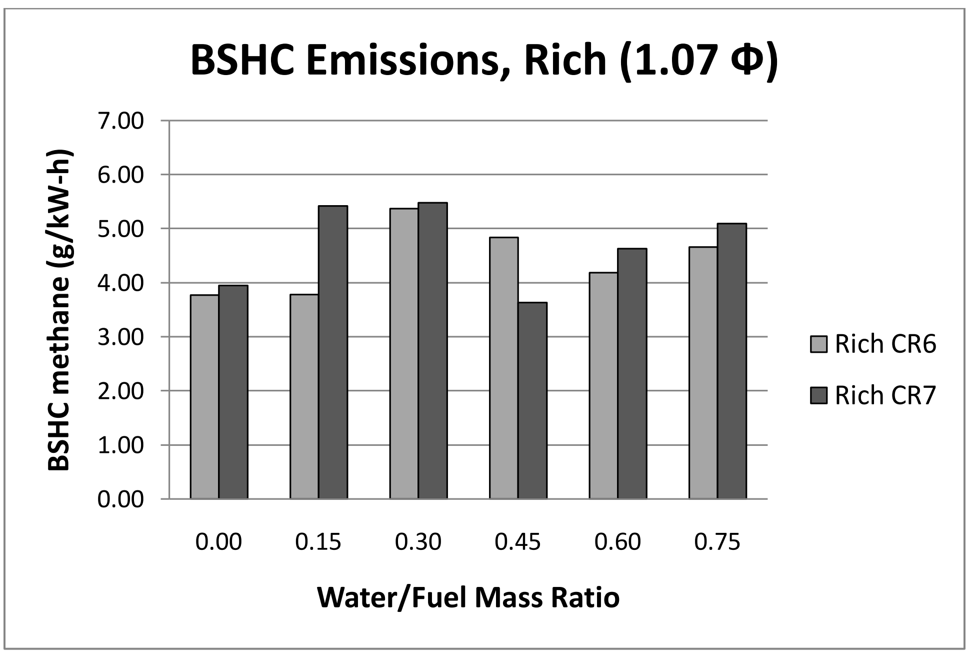

Figure 48 BSHC Emissions at Rich Conditions

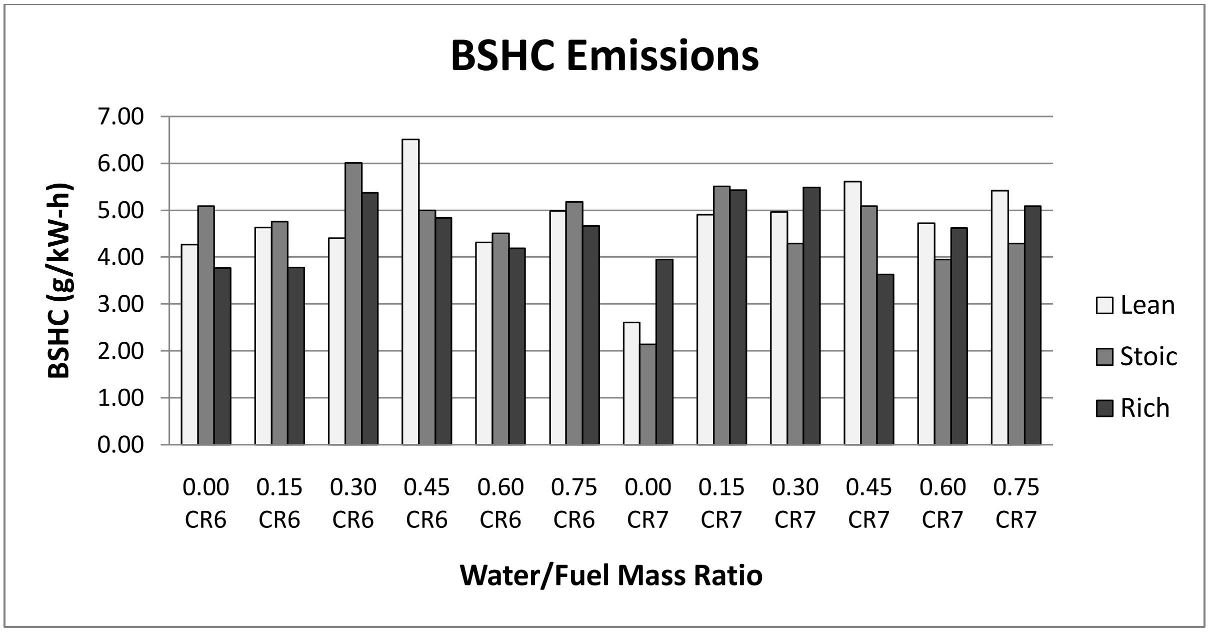

Figure 49 BSHC Emissions All Conditions

In agreement with other research (3, 9) the increased hydrocarbon levels may be due to increased quenching layer thickness. When the hot flame approaches the cooler cylinder wall, eventually the fuel (hydrocarbons) will lose too much heat to the wall and the flame will go out. Since the water cools the charge it may increase the thickness of the layer of air fuel mixture that is quenched and does not burn. Thus, there would be an increase in unburned hydrocarbon emissions.

From the data collected it is difficult to say if there is a BSHC trend with increased compression ratio.

4.0 Conclusions

The purpose of the present study was to show the trends of multiple important effects of water injection in combination with an increased compression ratio in a gasoline SI engine. The engine was modified to have manifold fuel injection and manifold water injection. Water/fuel mass ratio was varied between 0 and 0.75 . Compression ratio was varied from 6:1 to 7:1. Testing was done at three air/fuel ratios. Based on the experimental data it is concluded that water injection may:

- Increase the torque and power output of the engine
- Decrease brake specific fuel consumption and thus improve efficiency
- Reduce exhaust temperature of the engine
- Lower NO emissions
- Lower CO emissions, at rich and stoichiometric conditions
- Increase HC emissions

It is also concluded that increasing the compression ratio may:

- Increase the torque and power output of the engine
- Decrease brake specific fuel consumption and thus improve efficiency
- Reduce exhaust temperature of the engine
- Increase NO emissions
- Increase CO emissions, at rich conditions
- Have an inconclusive effect on HC emissions

Furthermore it is concluded that water injection in combination with an increase in compression ratio may result in the follow benefits:

- Increase in the torque (and power output) of the engine, (up to 65%)

- Decrease in brake specific fuel consumption (up to 39%) and thus improve efficiency
- Reduction of exhaust temperature of the engine (up to 10%)
- Lower NO emissions (up to 78%)
- Lower CO emissions, at rich and stoichiometric conditions (up to 89%)

However, there is also a detriment to water addition in combination with an increase in compression ratio:

- Increase in HC emissions (up to 45%)

It is concluded that water injection in combination with an increase in compression ratio may represents a means to significantly improve the torque, decrease BSFC, reduce temperature, lower NO and CO emissions, while increasing HC emissions. Note that emissions are prior to a catalytic converter. Naturally, in order to gain the greatest benefit of water injection with an increase in compression ratio, the engine would require some fine tuning. For a mobile use of this engine, it is recommended to use a 0.15 WFR and a 7:1 compression ratio and lean operating conditions. This amount of water would not significantly increase the mass of the vehicle, but provides significant improvement in torque and BSFC, less NO, already low CO and marginal increase in HC. Fine tuning would allow for the overall benefit of water injection with an increase in compression ratio to be maximized.

5.0 Future Work

Various experiments could be done to further our knowledge of water injection with increased compression ratios. Potential future work might include the following, prioritize with suggestions considered more important listed last.

1. If the same type of ECU is used in future research, note that cutting power or interrupting the ECU while maps or other settings are being burned or fetched to it may cause the ECU to be corrupted. Also, stop the engine when burning settings. If the ECU is receiving a trigger signal when it is turned on it may not work. Power should be turned off when removing ECU. If the spark plug is removed from the engine it must be grounded. All grounds should be connected to a common ground. Suggestions received from MBE Motorsports Inc.

2. Test water addition on a more stable setup. It should also be noted for future research- that a better combination of engine and dynamometer should be sought. In the present study the engine was somewhat small for the size of the dynamometer. The dynamometer had difficulty holding the small engine at a constant speed. A small change in the load setting of the dynamometer caused a larger change in engine speed. In this case a larger engine or a smaller dynamometer would have been preferred.

3. Measure in-cylinder pressure traces with water addition and multiple compression ratios. This would help verify the torque and efficiency benefits of water addition with varied compression ratios.

4. Calculate the in-cylinder temperature with water addition and an increase in compression ratio to see the size of the effect that reduced temperatures have on cycle efficiency.

5. Tune spark timing for maximum brake torque, along with water addition and increased compression ratio.

6. Vary methods of water injection: direct injection vs. manifold injection.

7. Research emissions with water injection and increased CR before and after a catalytic converter.

8. Perform similar research using a more modern four stroke engine.

9. Perform similar research for a two-stroke engine.

10. Researching the feasibility of condensing steam in the exhaust for reuse in water injection. Abundant amounts of water would be available since water is injected and a major product of combustion is H_2O. Recycling the water would be a great advantage for automotive applications since the consumer would not have to remember to fill up with water, the vehicle would not have to carry a water tank and what little water is carried could be expelled if the vehicle is stored at freezing temperatures. However, water condensed from exhaust may be acidic and precautions may need to be taken to prevent corrosion.

11. Increasing compression ratio further to find the knock limited compression ratio at various water/fuel ratios, then testing the effects at the knock limited compression ratios. Finding the knock limited compression ratios will allow the power and efficiency benefits of water addition to be maximized.

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Appendix A – Data

Description	Target Water/Fuel Mass Ratio	RPM	RPM StDev	Throttle Counts	SAE Torque (lb-ft)	Torque StDev (lb-ft)	SAE Torque (N-m)	Torque StDev (N-m)	Power			
									HP	HP StDev	kW	kW StDev
Compression Ratio 6:1												
Lean	0	2466	54.6	WOT	13.8	2.1	18.7	2.8	6.5	0.9	4.8	0.7
Stoichiometric	0	2478	52.8	WOT	13.8	2.0	18.7	2.7	6.5	0.9	4.9	0.7
Rich	0	2486	76.6	WOT	18.6	3.2	25.3	4.4	8.8	1.4	6.5	1.0
Lean	0.15	2489	41.6	WOT	19.0	3.0	25.7	4.0	9.0	1.3	6.7	1.0
Stoichiometric	0.15	2476	32.7	WOT	19.2	2.5	26.1	3.4	9.1	1.1	6.8	0.8
Rich	0.15	2464	38.6	WOT	21.2	2.6	28.7	3.5	9.9	1.1	7.4	0.8
Lean	0.30	2474	32.76	WOT	20.7	2.8	28.1	3.8	9.8	1.2	7.3	0.9
Stoichiometric	0.30	2475	51.1	WOT	20.3	3.6	27.5	4.9	9.5	1.5	7.1	1.1
Rich	0.30	2489	47.77	WOT	19.8	3.3	26.8	4.4	9.4	1.4	7.0	1.0
Lean	0.45	2494	89.01	WOT	18.3	4.3	24.8	5.8	8.6	1.1	6.4	0.8
Stoichiometric	0.45	2475	150.9	WOT	19.2	4.7	26.0	6.3	9.0	2.1	6.7	1.5
Rich	0.45	2457	154	WOT	19.4	4.7	26.3	6.4	9.0	2.0	6.7	1.5
Lean	0.60	2499	58	WOT	20.2	4.3	27.4	5.8	9.6	1.8	7.1	1.4
Stoichiometric	0.60	2477	84.2	WOT	20.3	4.8	27.5	6.5	9.5	2.1	7.1	1.5
Rich	0.60	2438	52.7	WOT	23.5	3.0	31.8	4.0	10.9	1.3	8.1	0.9
Lean	0.75	2459	96.3	WOT	19.8	4.5	26.8	6.1	9.2	1.9	6.9	1.4
Stoichiometric	0.75	2483	120.5	WOT	20.3	5.1	27.5	6.9	9.5	2.2	7.1	1.7
Rich	0.75	2484	71.7	WOT	21.3	4.1	28.9	5.5	10.1	1.8	7.5	1.3
Compression Ratio 7:1												
Lean	0	2493	33.3	WOT	18.5	3.1	25.1	4.2	8.8	1.4	6.5	1.0
Stoichiometric	0	2470	37.8	WOT	20.8	3.7	28.2	4.9	9.8	1.6	7.3	1.2
Rich	0	2491	48.3	WOT	19.5	3.5	26.4	4.7	9.2	1.5	6.9	1.1
Lean	0.15	2500	39.0	WOT	20.1	4.0	27.3	5.4	9.5	1.7	7.1	1.3
Stoichiometric	0.15	2492	33.2	WOT	20.4	3.2	27.7	4.3	9.6	1.4	7.2	1.0
Rich	0.15	2492	29.9	WOT	21.3	2.7	28.9	3.7	10.1	1.2	7.5	0.9
Lean	0.30	2492	43.7	WOT	20.6	4.0	27.9	5.4	9.8	1.7	7.3	1.3
Stoichiometric	0.30	2490	27.8	WOT	21.5	2.9	29.2	3.9	10.2	1.3	7.6	0.9
Rich	0.30	2486	50.3	WOT	21.7	4.2	29.4	5.7	10.2	1.8	7.6	1.3
Lean	0.45	2493	30.9	WOT	19.5	2.7	26.4	3.6	9.2	1.2	6.9	0.9
Stoichiometric	0.45	2495	51.3	WOT	21.2	3.5	28.7	4.7	10.0	1.5	7.5	1.1
Rich	0.45	2468	34.0	WOT	22.8	3.2	30.9	4.4	10.7	1.4	8.0	1.0
Lean	0.60	2487	43.6	WOT	20.1	3.2	27.3	4.4	9.5	1.4	7.1	1.0
Stoichiometric	0.60	2474	45.7	WOT	22.2	3.7	30.1	5.1	10.4	1.6	7.8	1.2
Rich	0.60	2487	45.1	WOT	20.9	3.5	28.3	4.7	9.9	1.5	7.4	1.1
Lean	0.75	2490	29.1	WOT	21.8	2.8	29.6	3.8	10.3	1.2	7.7	0.9
Stoichiometric	0.75	2483	45.6	WOT	22.8	3.3	30.9	4.5	10.8	1.4	8.1	1.1
Rich	0.75	2477	56.2	WOT	22.4	3.7	30.4	5.0	10.6	1.6	7.9	1.2

Description	Target Water/Fuel Mass Ratio	Fuel flow					Water Flow				BSFC	
		Time (s)	Mass (g)	Flow (g/s)			Time (s)	Mass (g)	Flow (g/s)	WFR	g/kw-hr	Error
Compression Ratio 6:1												
Lean	0	23.20	10	0.431				0	0	0.00	322	57
Stoichiometric	0	23.08	10	0.433				0	0	0.00	321	55
Rich	0	21.12	10	0.473				0	0	0.00	260	48
Lean	0.15	70.47	30	0.426			82.44	5	0.061	0.142	229	34
Stoichiometric	0.15	22.51	10	0.444			72.25	5	0.069	0.156	237	36
Rich	0.15	21.26	10	0.470			63.70	5	0.078	0.167	229	34
Lean	0.30	22.74	10	0.440			45.02	6	0.133	0.303	218	34
Stoichiometric	0.30	21.70	10	0.461			36.75	5	0.136	0.295	234	44
Rich	0.30	21.04	10	0.475			34.27	5	0.146	0.307	245	44
Lean	0.45	23.26	10	0.430			24.85	5	0.201	0.468	240	39
Stoichiometric	0.45	22.37	10	0.447			24.85	5	0.201	0.450	240	60
Rich	0.45	20.51	10	0.488			23.58	5	0.212	0.435	262	64
Lean	0.60	22.91	10	0.436			19.42	5	0.257	0.590	220	47
Stoichiometric	0.60	22.46	10	0.445			36.55	10	0.274	0.615	226	54
Rich	0.60	21.08	10	0.474			35.75	10	0.28	0.590	210	32
Lean	0.75	22.68	10	0.441			30.37	10	0.329	0.747	231	53
Stoichiometric	0.75	21.10	10	0.474			27.53	10	0.363	0.766	240	61
Rich	0.75	20.16	10	0.496			29.21	10	0.342	0.690	238	48
Compression Ratio 7:1												
Lean	0	71.96	30	0.417				0	0	0.000	230	37
Stoichiometric	0	67.83	30	0.442				0	0	0.000	218	36
Rich	0	61.75	30	0.486				0	0	0.000	255	42
Lean	0.15	72.73	30	0.412			80.80	5	0.062	0.150	210	39
Stoichiometric	0.15	67.34	30	0.446			73.75	5	0.068	0.152	224	34
Rich	0.15	61.40	30	0.489			67.59	5	0.074	0.151	234	28
Lean	0.30	71.60	30	0.419			78.40	10	0.128	0.304	207	37
Stoichiometric	0.30	68.53	30	0.438			75.55	10	0.132	0.302	208	26
Rich	0.30	62.73	30	0.478			68.15	10	0.147	0.307	226	40
Lean	0.45	69.53	30	0.431			50.22	10	0.199	0.462	226	30
Stoichiometric	0.45	68.74	30	0.436			49.66	10	0.201	0.461	211	32
Rich	0.45	64.62	30	0.464			47.45	10	0.211	0.454	209	28
Lean	0.60	71.18	30	0.421			58.63	15	0.256	0.607	214	32
Stoichiometric	0.60	67.90	30	0.442			55.49	15	0.27	0.612	205	32
Rich	0.60	61.68	30	0.486			50.75	15	0.296	0.608	237	36
Lean	0.75	71.04	30	0.422			48.15	15	0.312	0.738	198	24
Stoichiometric	0.75	68.53	30	0.438			59.76	20	0.335	0.765	196	27
Rich	0.75	63.22	30	0.475			56.25	20	0.356	0.749	216	34

Description	Target Water/Fuel Mass Ratio	%CO Avg	%CO StDev	%CO2 Avg	%CO2 StDev	HC ppm as hexane Avg	HC ppm as hexane StDev	NO ppm Avg	NO ppm StDev	AFR Avg	AFR StDev	EGT's Avg (F)	St Dev (F)	Avg (K)	St Dev (K)
Compression Ratio 6:1															
Lean	0	0.14	0.00	14.28	0.03	323	4	3206	19	15.5	0.05	1256	4	953	2
Stoichiometric	0	0.54	0.20	14.46	0.21	402	58	2667	201	14.9	0.15	1300	6	978	3
Rich	0	2.08	0.13	13.93	0.04	392	8	2073	55	13.9	0.07	1286	3	970	2
Lean	0.15	0.10	0.00	12.94	0.06	448	16	2368	88			1134	15	885	8
Stoichiometric	0.15	0.16	0.01	14.37	0.06	496	27	2547	50			1199	7	921	4
Rich	0.15	1.61	0.13	14.31	0.10	445	22	1816	37			1189	7	916	4
Lean	0.30	0.12	0.01	13.59	0.06	471	12	1744	73			1201	6	923	3
Stoichiometric	0.30	0.13	0.01	13.80	0.08	612	54	1670	24			1185	7	914	4
Rich	0.30	1.34	0.13	14.32	0.03	585	21	1562	28			1189	5	916	3
Lean	0.45	0.14	0.01	12.97	0.12	609	15	674	48			1219	5	933	3
Stoichiometric	0.45	0.26	0.07	14.47	0.16	523	29	1741	97			1217	7	931	4
Rich	0.45	2.06	0.29	13.95	0.15	503	34	1153	65			1184	5	913	3
Lean	0.60	0.12	0.00	13.64	0.07	459	14	964	57			1207	7	926	4
Stoichiometric	0.60	0.16	0.02	14.34	0.08	492	12	1141	31			1185	9	914	5
Rich	0.60	1.93	0.27	14.05	0.13	540	43	947	64			1192	2	918	1
Lean	0.75	0.12	0.01	13.77	0.13	510	10	888	156			1214	10	930	6
Stoichiometric	0.75	0.26	0.10	14.41	0.11	539	30	871	51			1220	5	933	3
Rich	0.75	1.47	0.21	14.26	0.10	523	12	776	30			1182	9	912	5
Compression Ratio 7:1															
Lean	0	0.09	0.01	14.12	0.06	272	6	3656	19	15.7	0.07	1180	2	911	1
Stoichiometric	0	0.78	0.09	14.39	0.02	250	10	3007	46	14.6	0.07	1216	1	931	1
Rich	0	2.55	0.15	13.48	0.08	421	12	1884	62	13.6	0.09	1188	2	915	1
Lean	0.15	0.07	0.00	13.01	0.05	522	18	2793	56			1085	7	858	4
Stoichiometric	0.15	0.08	0.01	13.77	0.17	582	28	3177	59			1139	13	888	7
Rich	0.15	1.49	0.20	13.98	0.09	613	26	2086	99			1148	6	893	3
Lean	0.30	0.06	0.00	13.20	0.08	541	20	2496	106			1130	3	883	2
Stoichiometric	0.30	0.12	0.02	13.99	0.11	495	29	2432	29			1120	10	878	5
Rich	0.30	1.56	0.19	13.95	0.09	642	17	1735	75			1133	5	885	3
Lean	0.45	0.06	0.01	13.27	0.09	565	20	1921	52			1119	6	877	3
Stoichiometric	0.45	0.10	0.02	14.02	0.09	582	30	2206	55			1131	11	884	6
Rich	0.45	2.27	0.21	13.52	0.11	464	13	1153	53			1142	2	890	1
Lean	0.60	0.07	0.01	13.39	0.10	505	32	1388	71			1110	7	872	4
Stoichiometric	0.60	0.17	0.02	14.12	0.05	468	24	1524	21			1135	9	886	5
Rich	0.60	2.17	0.14	13.55	0.09	521	10	918	34			1136	5	886	3
Lean	0.75	0.07	0.02	13.21	0.06	623	16	1055	39			1118	5	876	3
Stoichiometric	0.75	0.14	0.02	14.05	0.09	530	22	1227	21			1123	6	879	3
Rich	0.75	1.84	0.17	13.86	0.08	631	21	880	36			1128	2	882	1

For explanation as to why AFRs are not included for WFRs above 0 see section 3.0.

Description	Target Water/Fuel Mass Ratio	Brake Specific Setup	H2 dry	K factor	CO wet	CO2 wet	HC wet methane	NO wet	Total C	BSCO (g/kWh)	BSHC (g/kWh) as methane	BSNO (g/kWh)
Compression Ratio 6:1												
Lean	0	H/C	0.04	0.88	0.12	12.6	1708	2826	12.9	6.2	4.27	18.65
Stoichiometric	0	1.87	0.17	0.88	0.47	12.7	2119	2343	13.4	23.0	5.09	14.87
Rich	0		0.71	0.88	1.82	12.2	2058	1814	14.2	67.2	3.77	8.78
Lean	0.15	M_{CO}	0.03	0.89	0.09	11.5	2397	2111	11.9	3.5	4.63	10.79
Stoichiometric	0.15	28.01	0.05	0.88	0.14	12.7	2621	2243	13.1	5.2	4.75	10.75
Rich	0.15	g/mol	0.54	0.87	1.41	12.5	2335	1588	14.2	45.9	3.78	6.79
Lean	0.30	M_{NO2}	0.04	0.89	0.11	12	2506	1546	12.4	3.8	4.40	7.18
Stoichiometric	0.30	46.01	0.04	0.89	0.12	12.2	3250	1478	12.7	4.3	6.01	7.22
Rich	0.30	g/mol	0.44	0.88	1.17	12.5	3074	1368	14.0	41.4	5.37	6.32
Lean	0.45	M_{FUEL}	0.04	0.89	0.12	11.6	3256	601	12.0	5.0	6.51	3.18
Stoichiometric	0.45	13.895	0.08	0.88	0.23	12.7	2760	1531	13.2	8.4	5.00	7.33
Rich	0.45	g/mol	0.7	0.88	1.8	12.2	2641	1009	14.3	66.6	4.84	4.89
Lean	0.60	$H_{SPEC.}$	0.04	0.89	0.11	12.1	2441	854	12.4	3.8	4.32	3.99
Stoichiometric	0.60	3.0343	0.05	0.88	0.14	12.6	2601	1005	13.0	4.9	4.50	4.60
Rich	0.60	gm/kg	0.65	0.87	1.69	12.3	2835	829	14.3	50.2	4.18	3.23
Lean	0.75	K_H	0.04	0.89	0.11	12.2	2709	786	12.6	3.9	4.99	3.83
Stoichiometric	0.75	0.798	0.08	0.88	0.23	12.7	2846	766	13.2	8.4	5.18	3.69
Rich	0.75		0.49	0.88	1.29	12.5	2747	679	14.0	44.0	4.66	3.05
Compression Ratio 7:1												
Lean	0		0.03	0.88	0.08	12.5	1441	3228	12.7	2.9	2.61	15.45
Stoichiometric	0		0.25	0.88	0.68	12.6	1317	2639	13.4	22.4	2.13	11.31
Rich	0		0.89	0.88	2.23	11.8	2214	1651	14.3	80.4	3.95	7.79
Lean	0.15		0.02	0.89	0.06	11.6	2791	2489	11.9	2.2	4.90	11.55
Stoichiometric	0.15		0.03	0.89	0.07	12.2	3092	2813	12.6	2.5	5.51	13.25
Rich	0.15		0.5	0.88	1.31	12.3	3227	1830	13.9	44.3	5.42	8.13
Lean	0.30		0.02	0.89	0.05	11.7	2889	2221	12.1	1.8	4.96	10.08
Stoichiometric	0.30		0.04	0.88	0.11	12.4	2625	2149	12.7	3.5	4.28	9.27
Rich	0.30		0.52	0.88	1.37	12.2	3379	1522	13.9	44.7	5.48	6.52
Lean	0.45		0.02	0.89	0.05	11.8	3015	1708	12.2	2.0	5.61	8.41
Stoichiometric	0.45		0.03	0.88	0.09	12.4	3085	1949	12.8	2.9	5.08	8.49
Rich	0.45		0.78	0.88	1.99	11.9	2442	1012	14.1	59.6	3.63	3.97
Lean	0.60		0.02	0.89	0.06	11.9	2692	1233	12.2	2.2	4.72	5.7
Stoichiometric	0.60		0.05	0.88	0.15	12.5	2478	1345	12.9	4.8	3.95	5.7
Rich	0.60		0.74	0.88	1.9	11.9	2743	806	14.1	64.7	4.63	3.6
Lean	0.75		0.02	0.89	0.06	11.8	3326	939	12.2	2.0	5.42	4.0
Stoichiometric	0.75		0.04	0.88	0.12	12.4	2809	1084	12.8	3.8	4.29	4.4
Rich	0.75		0.62	0.88	1.61	12.2	3319	772	14.1	49.9	5.09	3.1

Appendix B – Emissions Charts on a Percent or ppmvol Basis

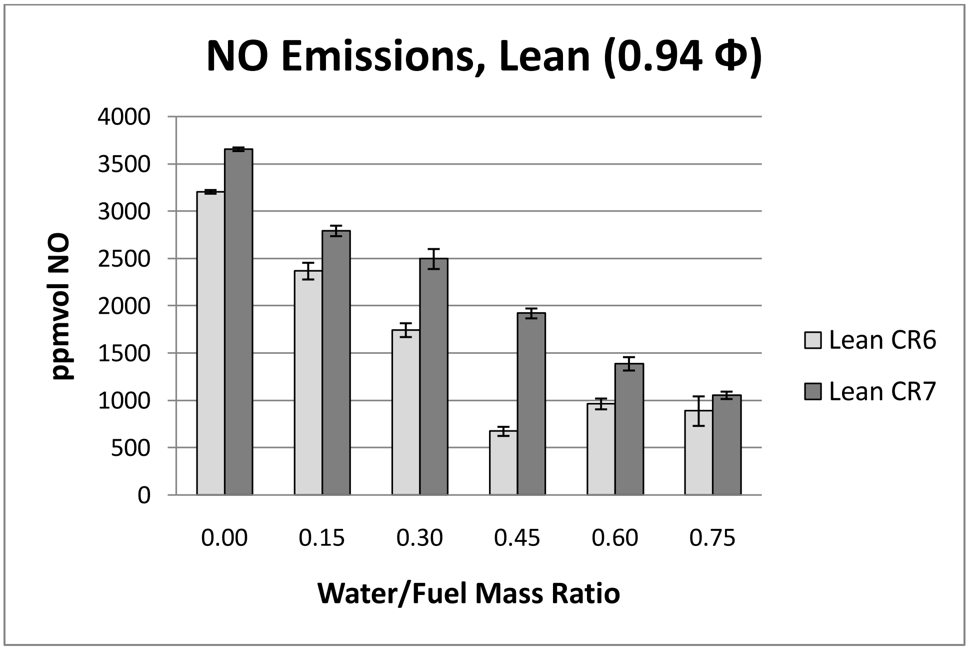


Figure 50 ppmvol NO Emissions at Lean Conditions

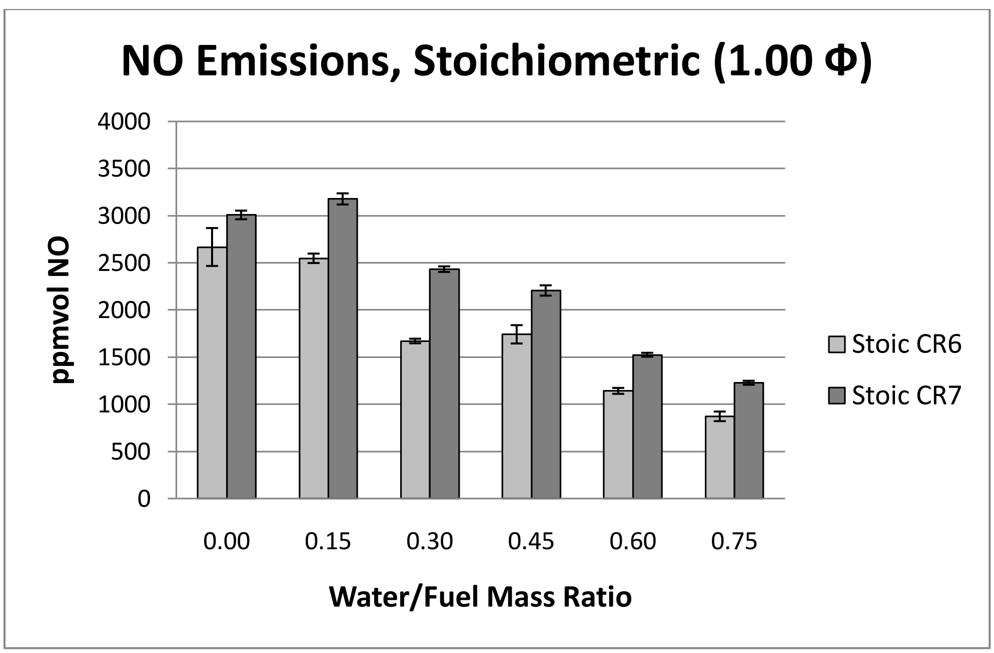


Figure 51 ppmvol NO Emissions at Stoichiometric Conditions

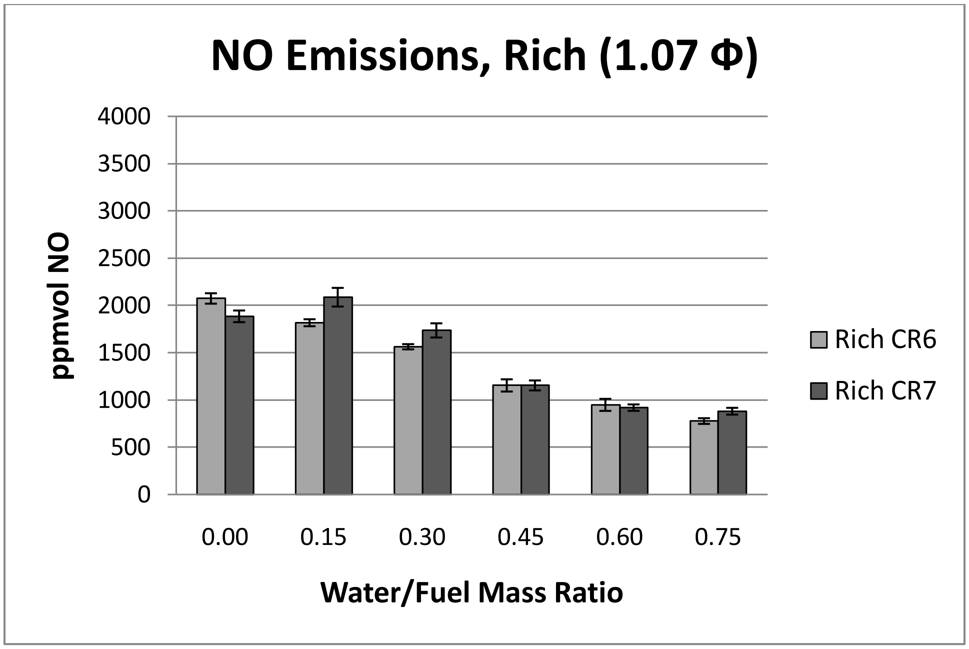


Figure 52 ppmvol NO Emissions at Rich Conditions

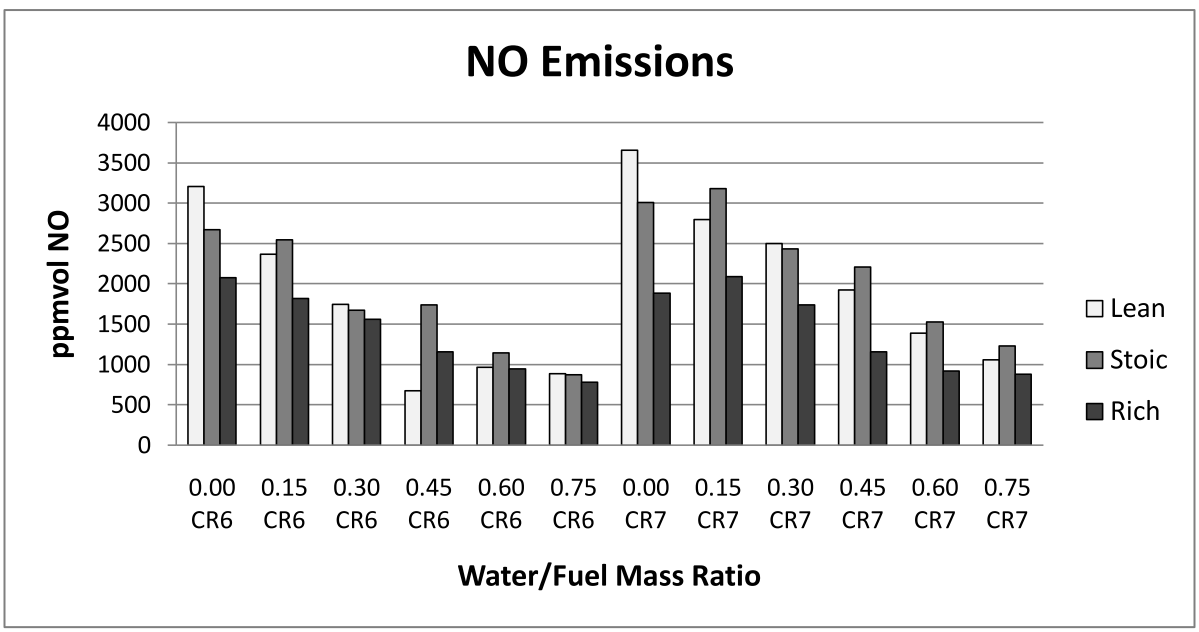


Figure 53 ppmvol NO Emissions All Conditions

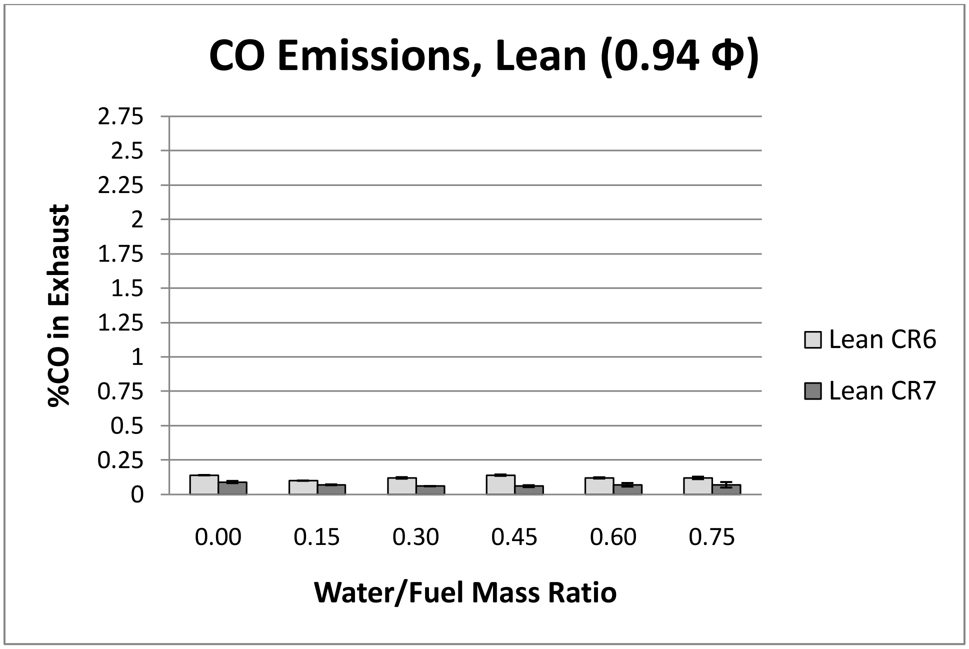


Figure 54 %CO Emissions at Lean Conditions

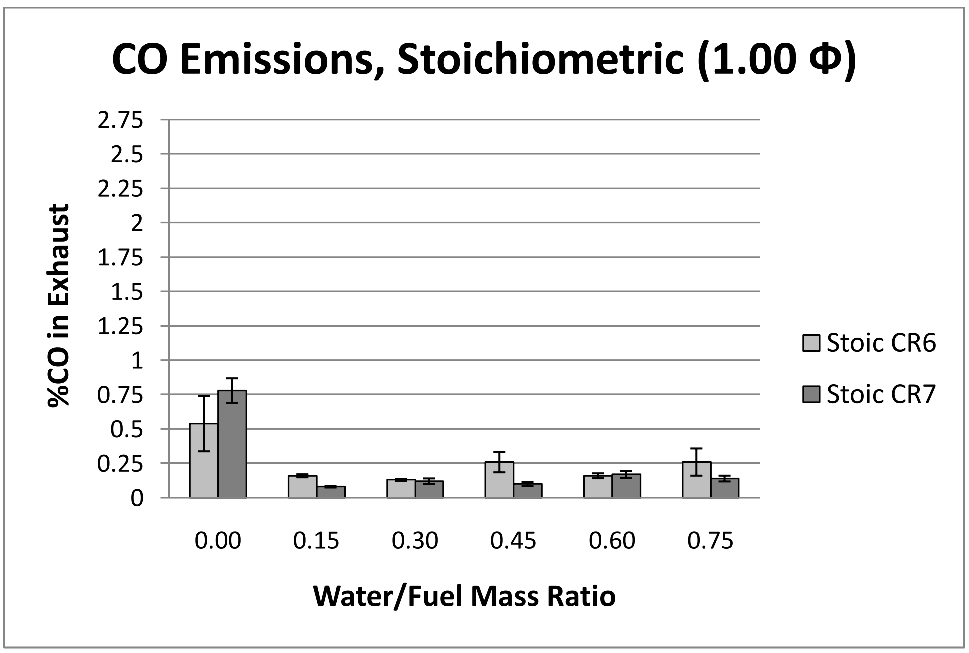


Figure 55 %CO Emissions at Stoichiometric Conditions

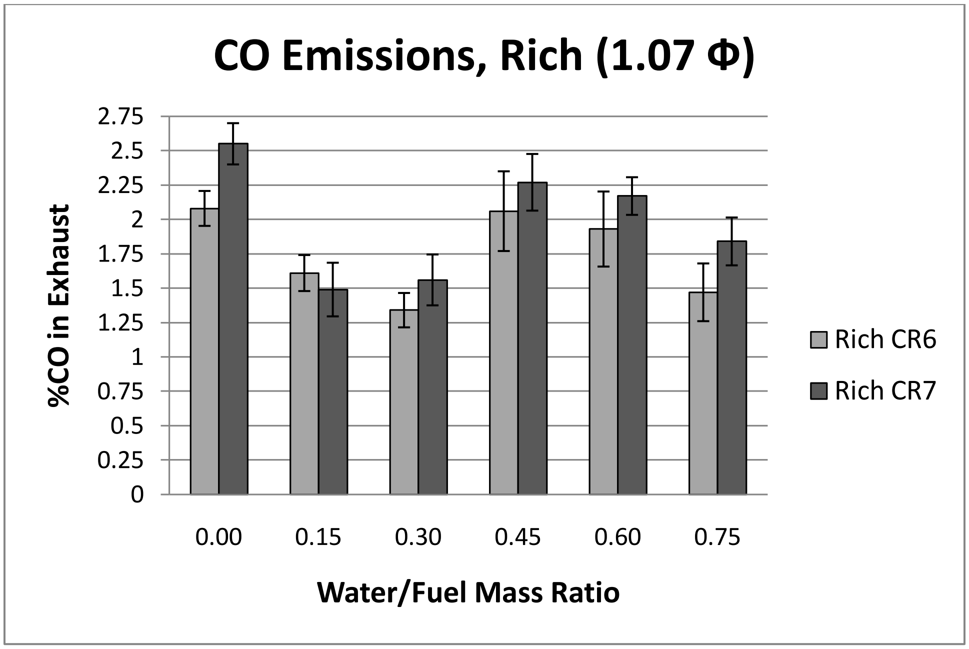


Figure 56 %CO Emissions at Rich Conditions

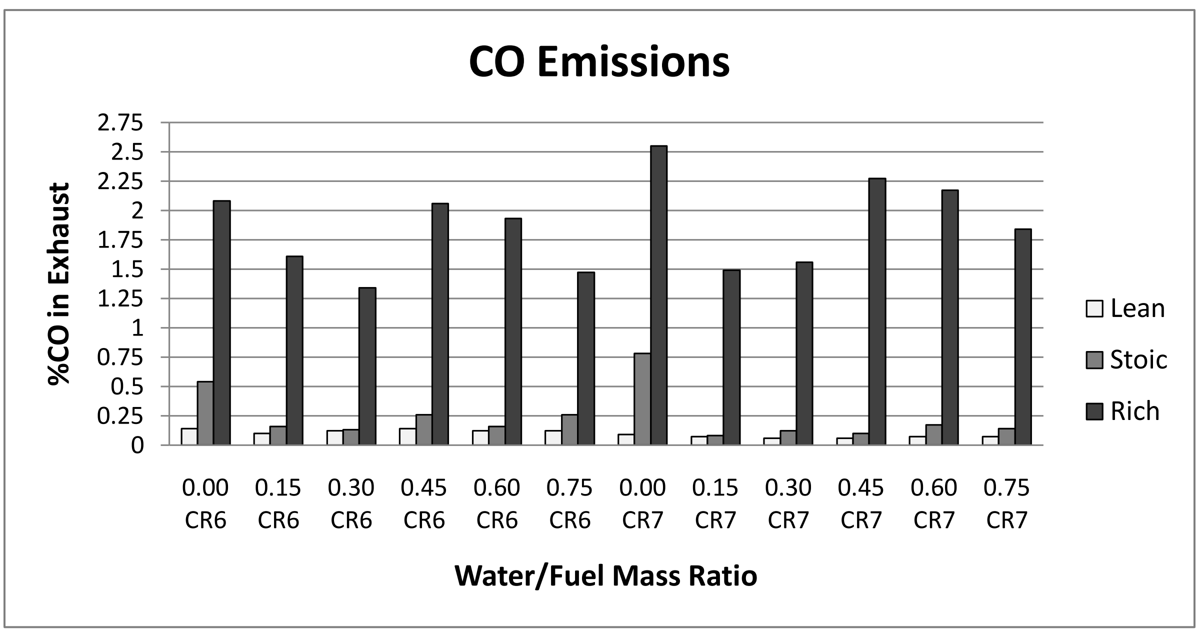


Figure 57 %CO Emissions All Conditions

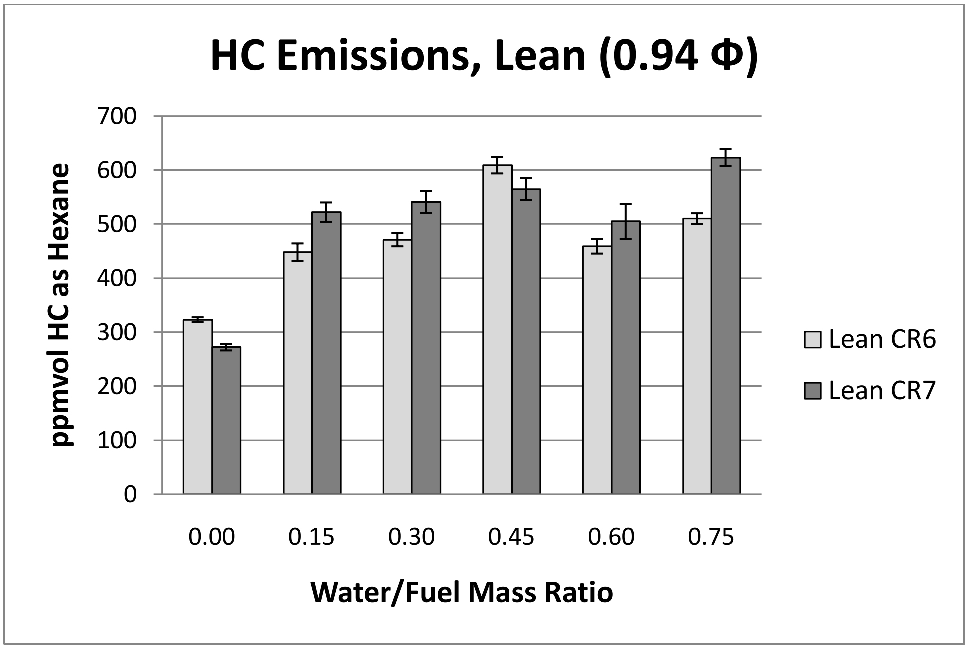


Figure 58 ppmvol HC Emissions at Lean Conditions

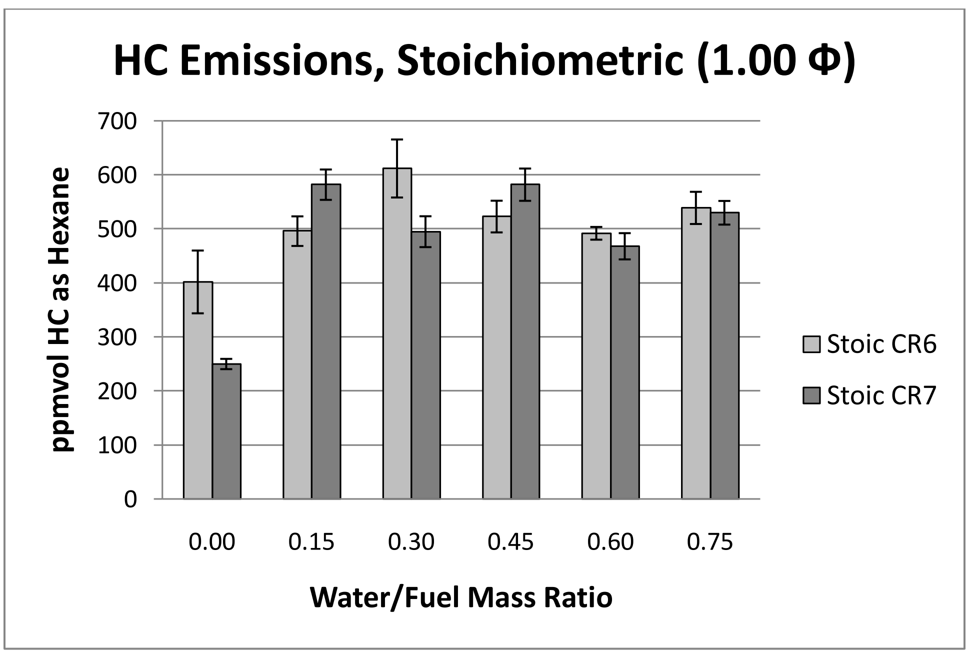


Figure 59 ppmvol HC Emissions at Stoichiometric Conditions

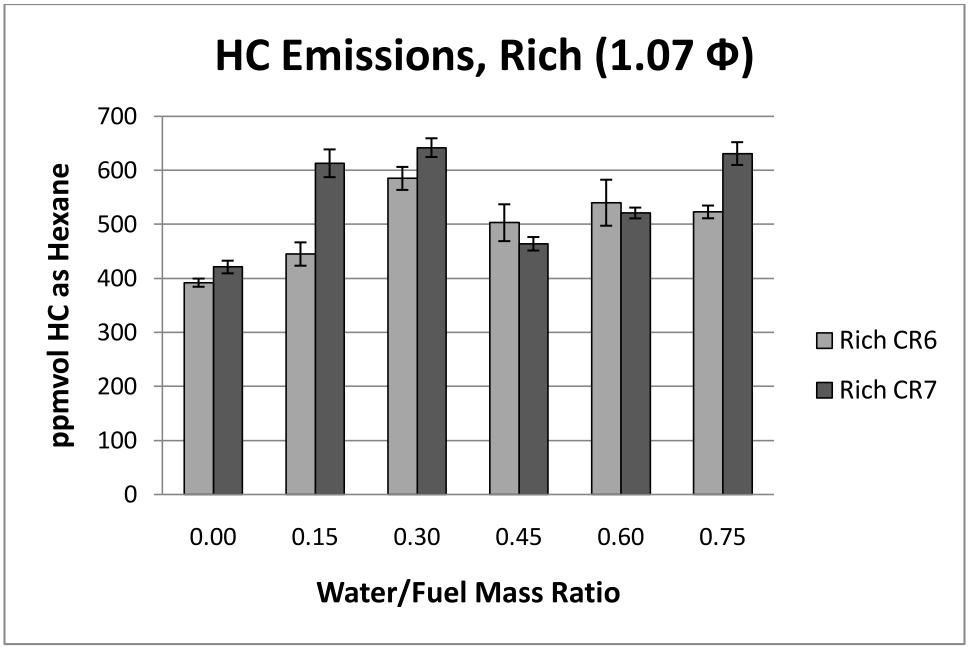


Figure 60 ppmvol HC Emissions at Rich Conditions

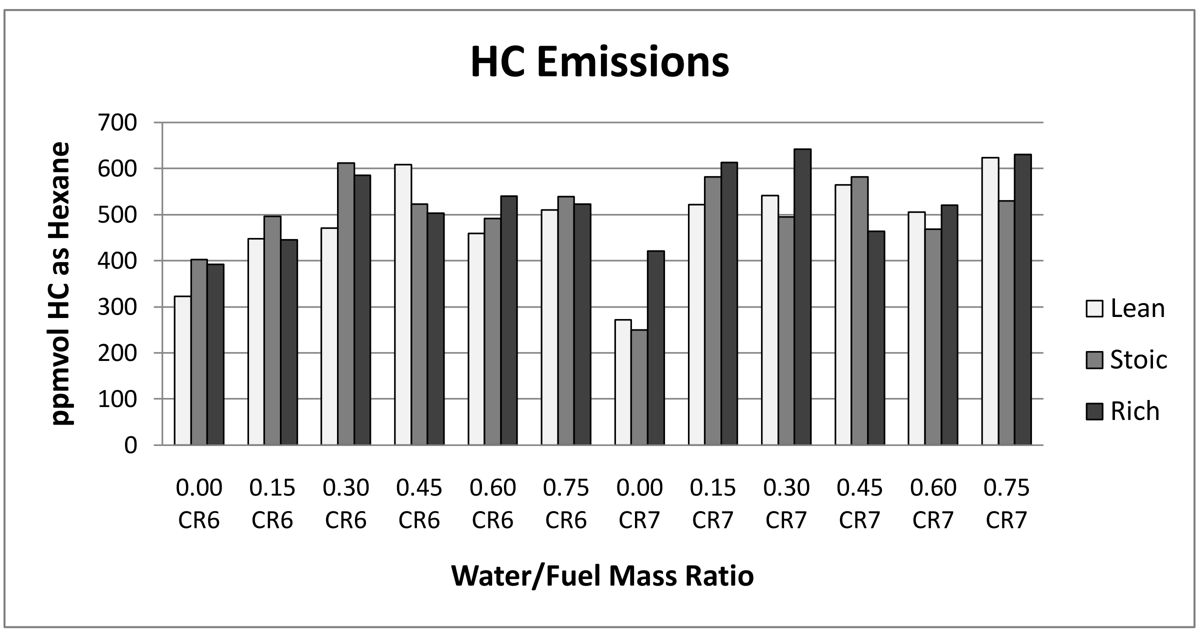


Figure 61 ppmvol HC Emissions All Conditions

Appendix C – Code of Federal Regulations (21)

Electronic Code of Federal Regulations
e-CFR TM

e-CFR Data are current as of May 29, 2008

Title 40: Protection of Environment

PART 91—CONTROL OF EMISSIONS FROM MARINE SPARK-IGNITION ENGINES

Subpart E—Gaseous Exhaust Test Procedures

Browse Previous | Browse Next

§ 91.419 Raw emission sampling calculations.

(a) Derive the final test results through the steps described in this section.

(b) Air and fuel flow method. If both air and fuel flow mass rates are measured, the following equations are used to determine the weighted emission values for the test engine:

(c) Fuel flow method. The following equations are to be used when fuel flow is selected as the basis for mass emission calculations using the raw gas method.

$$W_{HC} = \frac{G_{FUEL}}{TC} \times \frac{WHC}{10^4}$$

$$W_{CO} = \frac{M_{CO}}{M_F} \times \frac{G_{FUEL}}{TC} \times WCO$$

$$W_{NO_x} = \frac{M_{NOX}}{M_F} \times \frac{G_{FUEL}}{TC} \times \frac{WNO_X}{10^4} \times K_H$$

Where:

W_{HC}= Mass rate of HC in exhaust, [g/hr]

M_F= Molecular weight of test fuel; see following equation:

$$M_F = 12.01 + 1.008 \times \alpha$$

G_{FUEL}= Fuel mass flow rate, [g/hr]

TC = Total carbon; see following equation:

$$TC = WCO + WCO_2 + \frac{WHC}{10^4}$$

WHC = HC volume concentration in exhaust, ppmC wet

WCO = CO percent concentration in the exhaust, wet

DCO = CO percent concentration in the exhaust, dry

WCO_2= CO_2 percent concentration in the exhaust, wet

DCO_2= CO_2 percent concentration in the exhaust, dry

WNO_X= NO volume concentration in exhaust, ppm wet

WH2 = H_2 percent concentration in exhaust, wet

K = correction factor to be used when converting dry measurements to a wet basis. Therefore, wet concentration = dry concentration × K, where K is:

$$K = \frac{1}{1 + 0.005 \times \left(DCO + DCO_2\right) \times \alpha - 0.01 \times DH_2}$$

DH_2= H_2 percent concentration in exhaust, dry, calculated from the following equation:

$$DH_2 = \frac{0.5\times\alpha\times DCO\times(DCO+DCO_2)}{DCO+(3\times DCO_2)}$$

W_{CO}= Mass rate of CO in exhaust, [g/hr]

M_{CO}= Molecular weight of CO = 28.01

W_{NOx}= Mass rate of NO_X in exhaust, [g/hr]

M_{NO2}= Molecular weight of NO_2 = 46.01

K_H= Factor for correcting the effects of humidity on NO_2formation for four-stroke gasoline engines; see the equation below:

$$K_H = \frac{1}{1-0.0329\times(H-10.71)}$$

Where:

H = specific humidity of the intake air in grams of moisture per kilogram of dry air.

For two-stroke gasoline engines, KH should be set to 1.

Appendix D – Method of Calculating Brake Specific Emissions

Method for calculating Mass Flow Rate of Emissions using Volumetric Emissions and Mass Fuel Flow Rate Data from the Small Engine Research Facility
EPA CFR Chapter 40 Part 91.419

Unit Definitions

$rev := 2\cdot\pi\cdot rad$ $\quad ppm_{Methane} := 1$ $\quad ppm_{NOx} := 1$ $\quad ppm_{C1} := 1$

Molecular Weights of Gases

$M_{CO} := 28.01\cdot\frac{gm}{mol}$ $\quad M_{CO2} := 44.01\cdot\frac{gm}{mol}$ $\quad M_{NO2} := 46.01\cdot\frac{gm}{mol}$

Calculating Correction Factor

$H_{specific} := 3.0343 \quad \frac{gm}{kg}$ — Specific humidity of intake air in grams of moisture per kilogram of dry air

$K_H := \frac{1}{1 - 0.0329\cdot(H_{specific} - 10.71)}$ — Correction factor for effects of humidity on NO_2 formation $\quad K_H = 0.798$

Fuel Properties (must be changed for each fuel)

$HC_{ratio} := 1.87$ — Ratio of hydrogen atoms to carbon atoms

$MW_{fuel} := (12.01 + 1.008\cdot HC_{ratio})\cdot\frac{gm}{mol}$ $\quad MW_{fuel} = 13.895\cdot\frac{gm}{mol}$

Measured Engine Data (must be changed for each data point)

$RPM_{measured} := 2466\cdot\frac{rev}{min}$ — Engine RPM at point

$Torque_{measured} := 13.8\cdot ft\cdot lbf$ — Engine torque measured on dynamometer

$G_{fuel} := 1.5517\cdot\frac{kg}{hr}$ — Fuel mass flow rate at point

Measured Emissions Data - Dry Measurement (must be changed for each data point)

$CO2_{dry} := 14.28\%$ — CO_2 reading on analyzer

$CO_{dry} := .14\%$ — CO reading on analyzer

$HC_{dry} := 323\cdot 6\cdot ppm_{Methane}$ — HC reading on analyzer

$NOx_{dry} := 3206\cdot ppm_{NOx}$ — NO_x reading on analyzer

Calculated Data

$$Power_{measured} := Torque_{measured} \cdot RPM_{measured} \qquad Power_{measured} = 4.832 \cdot kW$$

Must calculate the percent dry H_2 in the exhaust in order to find the correction factor K
K is used to convert between dry and wet measurements

$$H2_{dry} := \frac{0.5 \cdot HC_{ratio} \cdot CO_{dry} \cdot (CO_{dry} + CO2_{dry})}{CO_{dry} + 3 \cdot CO2_{dry}} \qquad H2_{dry} = 0.044 \cdot \%$$

$$K_{factor} := \frac{1}{1 + \left[.005 \cdot (CO_{dry} + CO2_{dry}) \cdot HC_{ratio} - 0.01 \cdot H2_{dry}\right] \cdot 100} \qquad K_{factor} = 0.882$$

$$HC_{wet} := HC_{dry} \cdot K_{factor} \qquad HC_{wet} = 1.708 \times 10^{3} \cdot ppm_{C1}$$

$$CO_{wet} := CO_{dry} \cdot K_{factor} \qquad CO_{wet} = 0.123 \cdot \%$$

$$CO2_{wet} := CO2_{dry} \cdot K_{factor} \qquad CO2_{wet} = 12.588 \cdot \%$$

$$NOx_{wet} := NOx_{dry} \cdot K_{factor} \qquad NOx_{wet} = 2.826 \times 10^{3} \cdot ppm_{NOx}$$

Calculated Mass Emissions

Must calculate the total carbon percent (TC). This ratio helps calculate the mass flow of emissions based on the mass flow of the fuel.

$$TC := \left(CO_{wet} + CO2_{wet} + \frac{HC_{wet}}{10^{6}}\right) \cdot 100 \qquad TC = 12.883 \text{ \% Carbon}$$

$$HC := \frac{G_{fuel}}{\frac{TC}{100}} \cdot \frac{HC_{wet}}{10^{6}} \qquad \boxed{HC = 20.578 \cdot \frac{gm}{hr}}$$

$$CO := \frac{M_{CO}}{MW_{fuel}} \cdot \frac{G_{fuel}}{\frac{TC}{100}} \cdot CO_{wet} \qquad \boxed{CO = 29.966 \cdot \frac{gm}{hr}}$$

$$CO2 := \frac{M_{CO2}}{MW_{fuel}} \cdot \frac{G_{fuel}}{\frac{TC}{100}} \cdot CO2_{wet} \qquad \boxed{CO2 = 4.802 \times 10^{3} \cdot \frac{gm}{hr}}$$

$$NOx := \frac{M_{NO2}}{MW_{fuel}} \cdot \frac{G_{fuel}}{\frac{TC}{100}} \cdot \frac{NOx_{wet}}{10^{6}} \cdot K_H \qquad \boxed{NOx = 89.994 \cdot \frac{gm}{hr}}$$

Brake Specific Emissions

$$BSHC := \frac{HC}{Power_{measured}}$$

$$\boxed{BSHC = 4.259 \cdot \frac{gm}{kW \cdot hr}}$$

$$BSCO := \frac{CO}{Power_{measured}}$$

$$\boxed{BSCO = 6.202 \cdot \frac{gm}{kW \cdot hr}}$$

$$BSCO2 := \frac{CO2}{Power_{measured}}$$

$$\boxed{BSCO2 = 993.95 \cdot \frac{gm}{kW \cdot hr}}$$

$$BSNOx := \frac{NOx}{Power_{measured}}$$

$$\boxed{BSNOx = 18.626 \cdot \frac{gm}{kW \cdot hr}}$$

Comparing carbon flow in and carbon flow out

This section is a comparison of the mass flow of carbon out the exhaust compared to the fuel used. The two final values should match one another. If they do not, this indicates a source of error.

Mass flow of carbon from each emission source

$$m_{dot\_carbon\_HC} := \frac{HC}{(12.01 + 1.008)\cdot\frac{gm}{mol}}\cdot 12.01\cdot\frac{gm}{mol} \qquad m_{dot\_carbon\_HC} = 18.984\cdot\frac{gm}{hr}$$

$$m_{dot\_carbon\_CO} := \frac{CO}{M_{CO}}\cdot 12.01\cdot\frac{gm}{mol} \qquad m_{dot\_carbon\_CO} = 12.849\cdot\frac{gm}{hr}$$

$$m_{dot\_carbon\_CO2} := \frac{CO2}{M_{CO2}}\cdot 12.01\cdot\frac{gm}{mol} \qquad m_{dot\_carbon\_CO2} = 1.311\times 10^{3}\cdot\frac{gm}{hr}$$

$$m_{dot\_carbon\_exh} := m_{dot\_carbon\_HC} + m_{dot\_carbon\_CO} + m_{dot\_carbon\_CO2}$$

$$\boxed{m_{dot\_carbon\_exh} = 1.342\times 10^{3}\cdot\frac{gm}{hr}}$$ Mass flow out exhaust

$$m_{dot\_carbon\_fuel} := \frac{G_{fuel}}{MW_{fuel}}\cdot 12.01\cdot\frac{gm}{mol}$$

$$\boxed{m_{dot\_carbon\_fuel} = 1.341\times 10^{3}\cdot\frac{gm}{hr}}$$ Mass flow in to engine

Appendix E –Method of Error Analysis of Brake Specific Fuel Consumption

Brake Specific Fuel Consumption
Error Analysis Calculations
J. Parley Wilson

$$BSFC := \frac{gmsFuel \cdot 3600}{time \cdot Pwr}$$

$$\frac{d}{dgmsFuel} BSFC \rightarrow \frac{3600}{Pwr \cdot time}$$

$$\frac{d}{dtime} BSFC \rightarrow -\frac{3600 \cdot gmsFuel}{Pwr \cdot time^2}$$

$$\frac{d}{dPwr} BSFC \rightarrow -\frac{3600 \cdot gmsFuel}{Pwr^2 \cdot time}$$

$gmsFuel := 10$ $\quad$ $gmsFuelError := 1$

$time := 23.2$ $\quad$ $timeError := .2 = 0.2$

$Pwr := 4.824679$ $\quad$ $PwrError := .7046865$

$$BSFC := \frac{gmsFuel \cdot 3600}{time \cdot Pwr} = 321.622$$

$$\boxed{BSFCerror := \sqrt{\left(\frac{3600}{Pwr \cdot time} \cdot gmsFuelError\right)^2 + \left(-\frac{3600 \cdot gmsFuel}{Pwr \cdot time^2} \cdot timeError\right)^2 + \left(-\frac{3600 \cdot gmsFuel}{Pwr^2 \cdot time} \cdot PwrError\right)^2} = 56.998}$$

Verification using another method

$$\boxed{BSFCerror := BSFC \cdot \sqrt{\left(\frac{gmsFuelError}{gmsFuel}\right)^2 + \left(\frac{timeError}{time}\right)^2 + \left(\frac{PwrError}{Pwr}\right)^2} = 56.998}$$

Appendix F – Drawing Package of Modifications

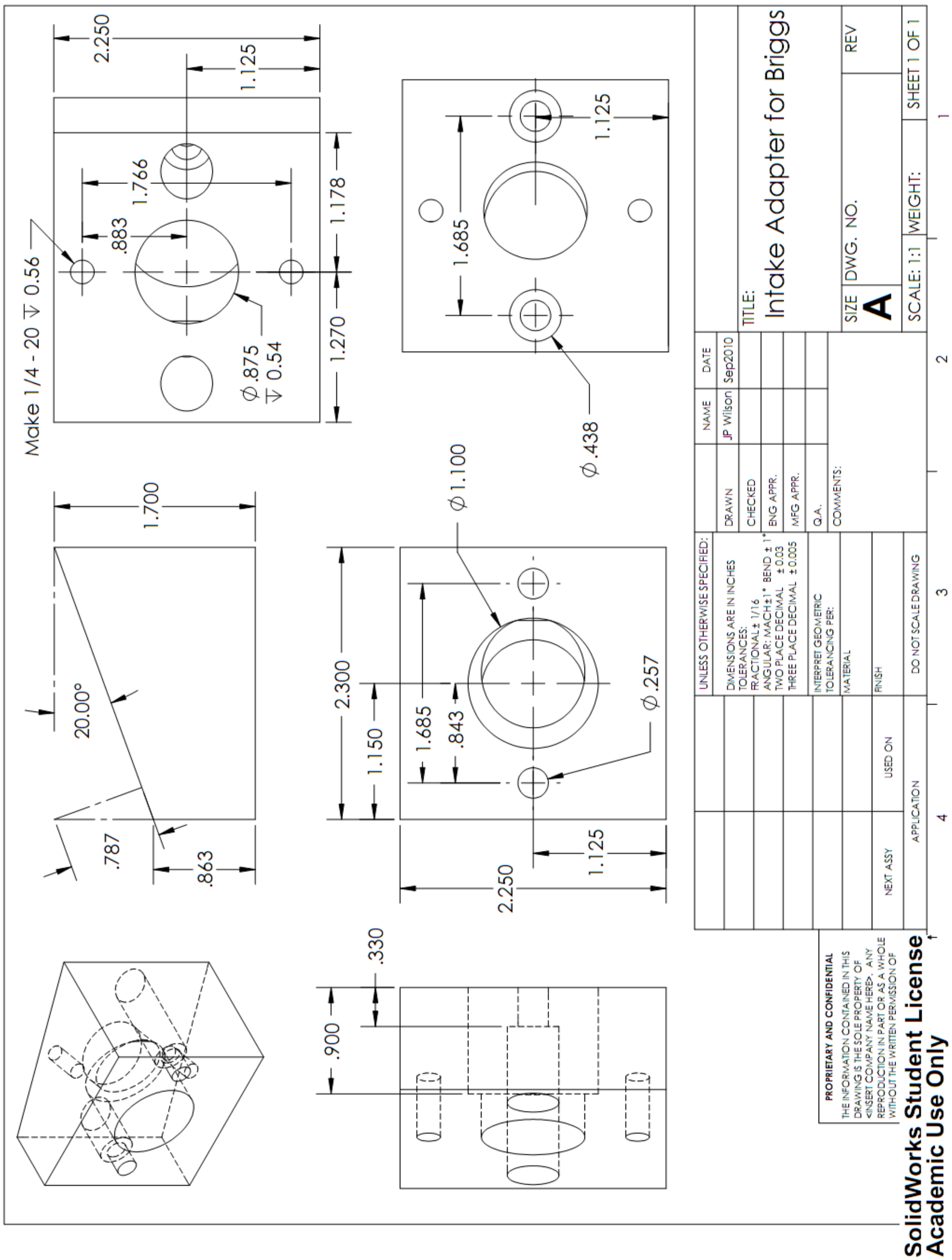
In order:

Intake Adapter

Throttle body Adapter

Engine Coupler

Crank Pickup Mount


Make 1/4 - 20 ↧ 0.56
2.250
1.125
1.766
.883
1.178
1.270
Ø.875
↧ 0.54
1.685
1.125
Ø.438
1.700
20.00°
.787
.863
2.300
1.150
1.685
.843
Ø1.100
Ø.257
2.250
1.125
.900
.330
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/16
ANGULAR: MACH ± 1° BEND ± 1°
TWO PLACE DECIMAL ± 0.03
THREE PLACE DECIMAL ± 0.005
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
JP Wilson
Sep2010
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
Intake Adapter for Briggs
SIZE
A
DWG. NO.
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
PROPRIETARY AND CONFIDENTIAL
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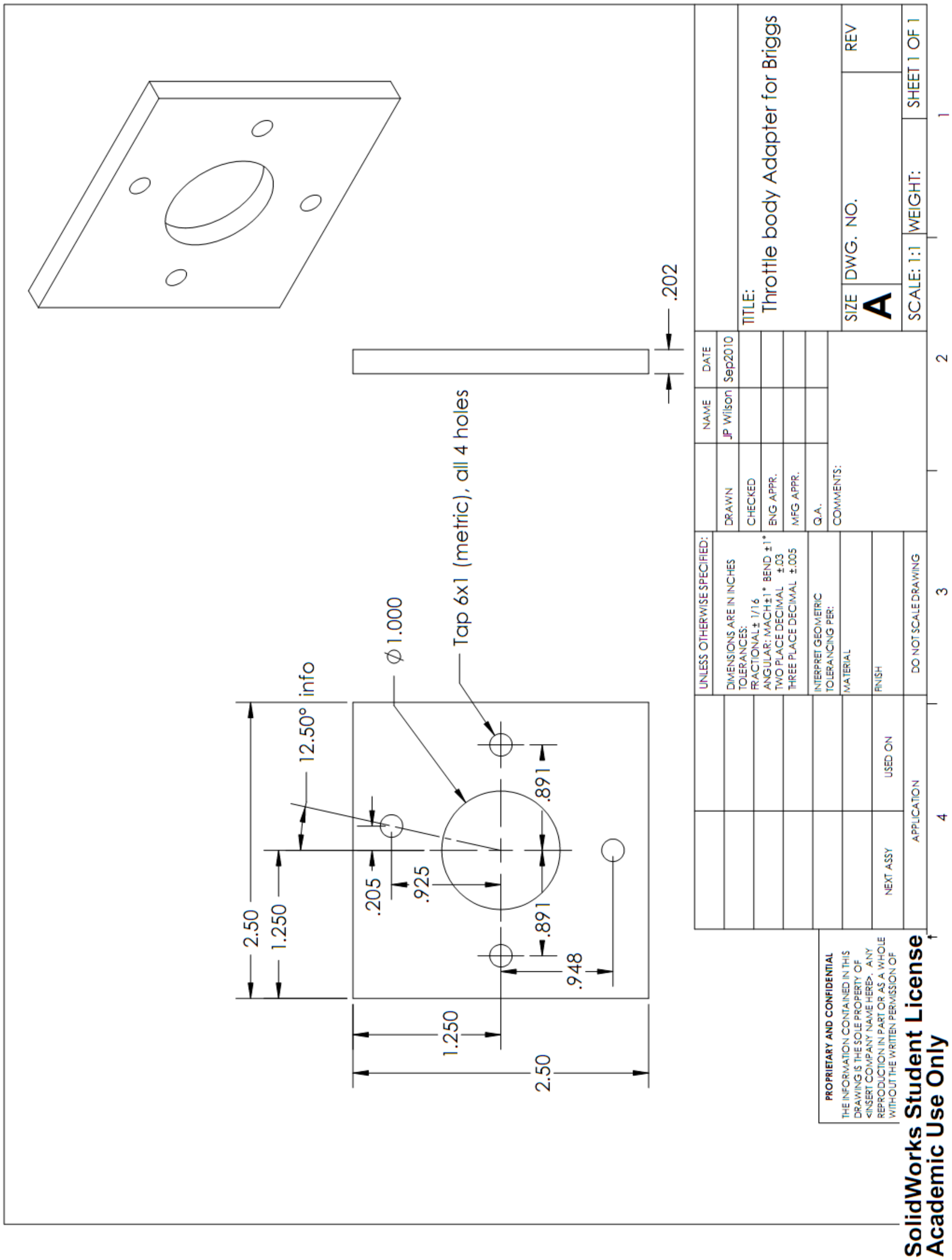

Throttle body Adapter for Briggs
TITLE:
Tap 6x1 (metric), all 4 holes
⌀1.000
12.50° info
2.50
1.250
.205
.925
.891
.891
.948
1.250
2.50
.202
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL ± 1/16
ANGULAR: MACH ± 1° BEND ± 1°
TWO PLACE DECIMAL ±.03
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
JP Wilson
Sep2010
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
SIZE
A
DWG. NO.
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
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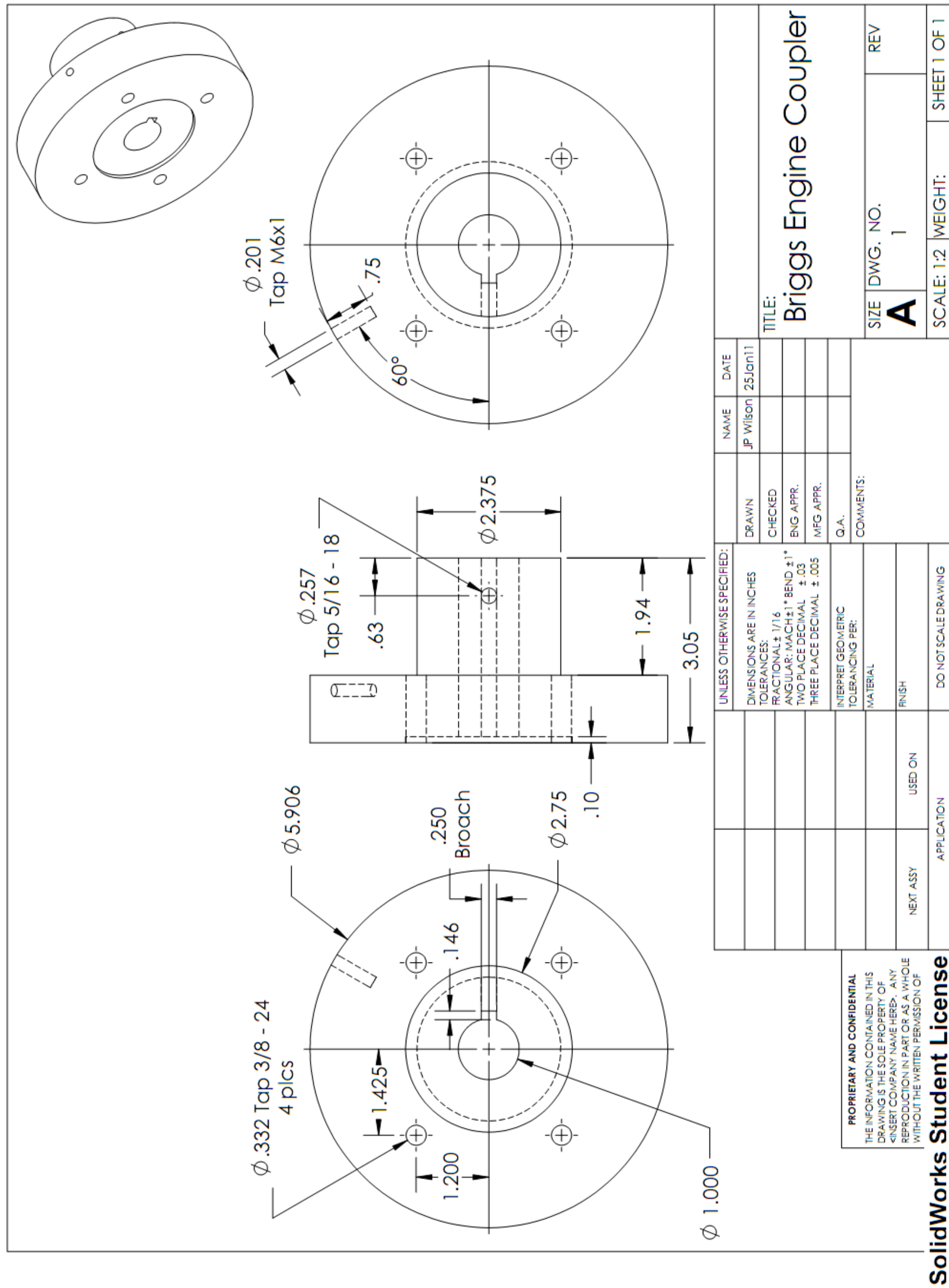
⌀.332 Tap 3/8 - 24
4 plcs
1.425
1.200
⌀5.906
.146
.250
Broach
⌀2.75
.10
⌀1.000
⌀.257
Tap 5/16 - 18
.63
1.94
3.05
⌀2.375
⌀.201
Tap M6x1
.75
60°
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL± 1/16
ANGULAR: MACH±1° BEND ±1°
TWO PLACE DECIMAL ±.03
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC
TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
JP Wilson
25Jan11
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
Briggs Engine Coupler
SIZE
A
DWG. NO.
1
REV
SCALE: 1:2
WEIGHT:
SHEET 1 OF 1
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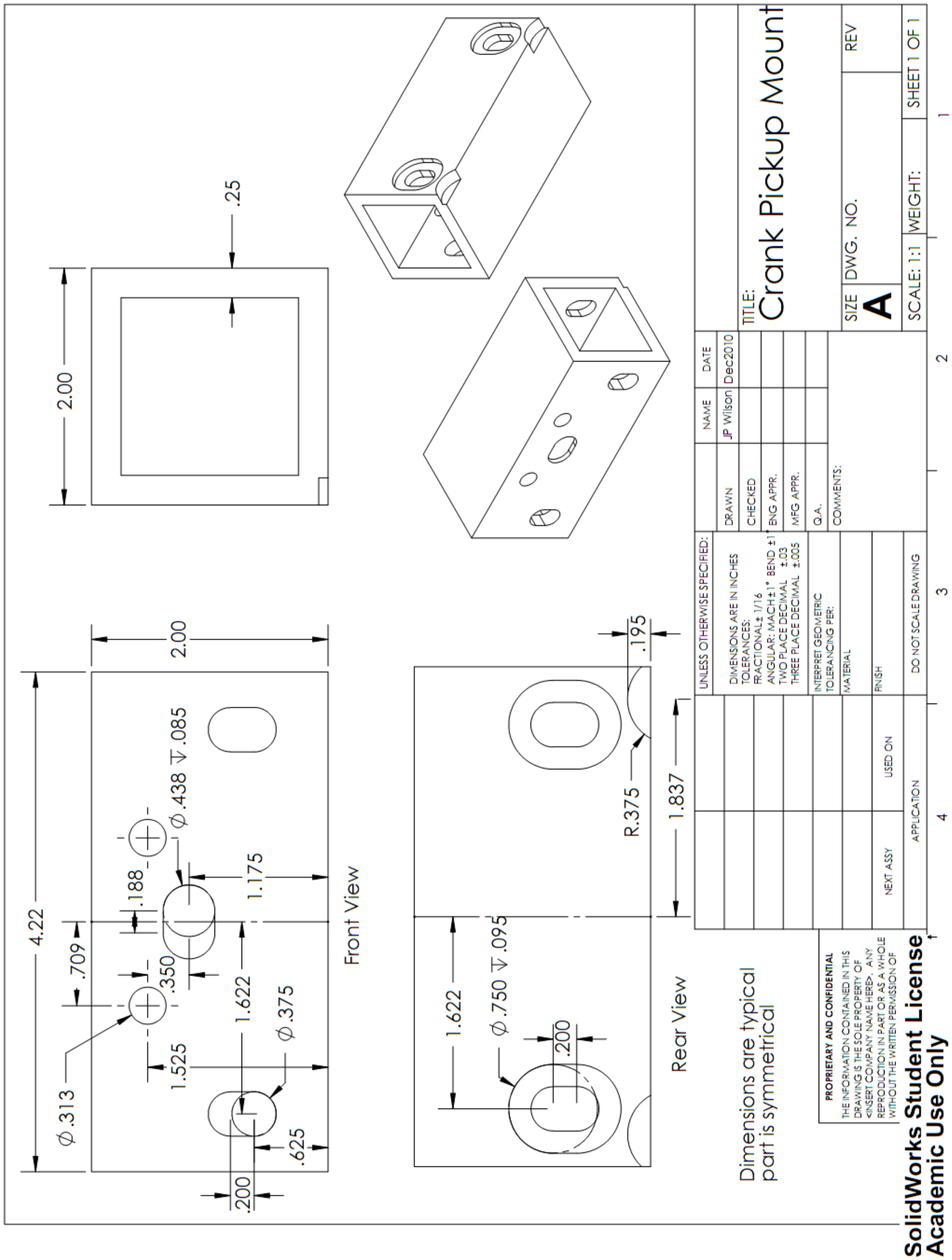

Front View
Rear View
Dimensions are typical
part is symmetrical
4.22
2.00
.709
.350
.188
1.175
1.525
1.622
.625
.200
ϕ.313
ϕ.375
ϕ.438 ↧ .085
ϕ.750 ↧ .095
R.375
1.837
.195
.25
UNLESS OTHERWISE SPECIFIED:
DIMENSIONS ARE IN INCHES
TOLERANCES:
FRACTIONAL± 1/16
ANGULAR: MACH ±1° BEND ±1°
TWO PLACE DECIMAL ±.03
THREE PLACE DECIMAL ±.005
INTERPRET GEOMETRIC
TOLERANCING PER:
MATERIAL
FINISH
DO NOT SCALE DRAWING
NEXT ASSY
USED ON
APPLICATION
NAME
DATE
DRAWN
JP Wilson
Dec2010
CHECKED
ENG APPR.
MFG APPR.
Q.A.
COMMENTS:
TITLE:
Crank Pickup Mount
SIZE
A
DWG. NO.
REV
SCALE: 1:1
WEIGHT:
SHEET 1 OF 1
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