CHARACTERIZATION OF
CATALYTIC IGNITER PERFORMANCE
AND EMISSIONS

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University of Idaho

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Steven Beyerlein, Ph.D. and Judith Steciak, Ph.D.
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15. Abstract  
Work on this project was focused in three main areas. Continuation of emissions and performance characterization of the ethanol/water fueled transit van was done by improving the cold-start ability, changing to a more powerful Engine Control Unit (ECU), and five-gas emissions were collected using a modal test cycle. Further data reduction was performed on the Yanmar diesel conversion that yields more accurate results of brake-specific fuel consumption and emissions. Lastly, in preparation for creating the new laboratory space in the UI Boise Center a state-of-the-art literature review was conducted on ethanol/water fuel and catalytic ignition. Initial setup of the laboratory space has already been undertaken.  

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EXECUTIVE SUMMARY
Work on this project was focused in three main areas. Continuation of emissions and
performance characterization of the ethanol/water fueled transit van was done by improving
the cold-start ability, changing to a more powerful Engine Control Unit (ECU), and five-gas
emissions were collected using a modal test cycle. Further data reduction was performed on
the Yanmar diesel conversion that yields more accurate results of brake-specific fuel
consumption and emissions. Lastly, in preparation for creating the new laboratory space in
the UI Boise Center a state-of-the-art literature review was conducted on ethanol/water fuel
and catalytic ignition. Initial setup of the laboratory space has already been undertaken.

Ethanol/Water Transit Van
The platform explored in this research is a transit van converted to operate on either gasoline,
using spark ignition, or ethanol-water fuel mixtures, using catalytic ignition. Catalytic
ignition systems allow combustion initiation over a much wider range of stoichiometry and
water content than traditional spark ignition systems. Chassis dynamometer tests comparing
vehicle efficiency as well as brake specific emissions of carbon monoxide, carbon dioxide,
nitrogen oxides, and hydrocarbons have been obtained for 70 percent ethanol - 30 percent
water fuel mixtures operating at air to fuel ratios of $\lambda = 1$ and $\lambda = 1.15$. Substantial reduction
in nitrogen oxides and carbon monoxide has been observed for water fuels compared to
gasoline. Analyzing differences in emissions and power between fuels with dissimilar
compositions requires an emissions model that takes into account fuel chemistry and engine
power output. A new emissions model was developed in this work and validated against
previous engine test data involving diesel and ethanol-water fuels.

Yanmar Diesel Conversion
Data was previously collected in the Small Engine Laboratory (SEL) on a 3-cylinder Yanmar
diesel engine converted to catalytic ignition and ethanol/water fuel. While the raw data was
taken correctly, some errors in the analysis of the data was found. Work was done to create a
new model for analyzing and comparing emissions produced from diesel to ethanol-water
fuels. A summary of the engine modifications and testing methods, experiment results and
conclusions will be submitted to Diesel.net. This section will contain explanation of the engine modifications required for operation on ethanol-water fuel, description of the methods and equipment used for data collection, and outline the new analysis model and results for brake specific emissions and engine performance.

State-of-the-Art Literature Review
A literature search was performed to find research results relating to the catalytic ignition of water-hydrocarbon-air mixtures. We found that, outside of our group, very little work is currently being pursued on the catalytic ignition of aqueous fuels at the high temperatures needed for combustion. Research was found on the effect of adding steam or water to gas-phase hydrocarbon combustion, and on the decomposition of hydrocarbons on catalytic surfaces. As a basis for comparison, this search included research relating to the determination of combustion catalysts, the ignition of hydrocarbon fuels in air over platinum, the combustion of aqueous fuels, the catalytic combustion of aqueous fuels, and steam injection during combustion. This information was gathered to support our research and development of internal combustion transportation engines fueled with aqueous ethanol under homogeneous charge, catalytically assisted compression ignition (HCCCI).
PART I – ETHANOL/WATER TRANSIT VAN

INTRODUCTION

Previous research on catalytic igniters and ethanol-water fueled engines has shown potential for lowering emissions and increasing engine efficiency over conventional engine configurations. Catalytic ignition systems allow combustion of lean mixtures also having a lower volatility. Lean burning in piston engines affords a means of achieving important environmental and fuel economy objectives. The primary drawbacks of traditional lean burn engines are de-rated power output per unit displacement and incompatibility with oxidation/reduction catalysts used in conventional exhaust clean-up systems. The need to overcome the difficulties related to lean burning originally spurred the development of the catalytic igniter [1]. Over the last eight years, the University of Idaho along with Automotive Resources, Inc. has combined catalytic igniter design with ethanol-water fuel technology to capture many of the benefits of lean burning without sacrificing power output, or increasing harmful emissions. The fuels examined in this research include a mixture of 70 percent ethanol and 30 percent water by volume, diesel, and gasoline.

DESCRIPTION OF PROBLEM

Catalytic Ignition

The catalytic igniter is a self-contained ignition system that may be retrofitted to existing spark-ignition (SI) and compression-ignition (CI) engines. An exploded view of the system and its parts is illustrated in Figure 1. The catalytic igniter consists of a ceramic rod with an embedded heating element and a coating of platinum catalyst. The catalytic core is enclosed in a custom-machined brass shell that forms a pre-chamber adjacent to the main combustion chamber. The shell fits into existing spark plug holes on SI engines, or direct fuel injection ports on CI engines, requiring the fuel to be injected upstream resulting in premixed fuel air mixture.
Ignition begins as fresh mixture contacts the catalyst during the compression stroke. Because of the reduced activation energy associated with heterogeneous catalysis, this reaction occurs at temperatures far below the normal gas-phase ignition temperature [3]. Combustion products and intermediate species then accumulate in the pre-chamber surrounding the catalytic core. After sufficient temperature is achieved due to compression, multi-point homogeneous ignition results [3, 4]. The burning mixture is then rapidly expelled through the nozzles at the bottom of the igniter. The nozzles direct the flame to ignite the entire combustion chamber. Upon reaching operating temperature, the ignition process becomes self-sustaining and no longer requires power from an external source [5]. Although the reaction becomes self-sustaining, changing the power supplied to the catalytic igniter can act to advance or retard timing by increasing or decreasing the surface temperature of the catalyst. With any homogenous charge compression ignition engine, controlling ignition timing is a critical problem.

Figure 1 Components of Catalytic Igniter [2]

Early experimental work explored a mechanical means of controlling catalytic ignition. Adjusting the position of the catalyst in the pre-chamber had a large effect on ignition timing. Changing igniter core length changes the crank angle where the fresh mixture first contacts the catalytic surface. Currently, this is the main means of setting the initial ignition timing in the converted engines. However, this does not allow for ignition advance at increasing engine speeds. Typically ignition timing should advance with engine speed, and retard with load so that peak pressure is reached at or shortly after TDC [6]. Since timing cannot be adjusted,
higher volatile fuels cause pre-ignition especially at higher loads and speeds. Altering the catalyst surface temperature controls the rate of reaction on the catalyst. The higher the catalyst temperature, the earlier ignition will occur. Currently surface temperature is only changed for cold starting. At high speeds and loads, the catalytic surface retains enough heat to be self-sustaining without electric heating. Starting requires 25 watts per igniter from an external power source of 12 volts. In recent engine conversions an increased supply voltage up to a maximum of 18 volts, providing 38 watts per igniter has shown an improved cold starting ability. Engine operation using alcohol fuels require special measures particularly at low temperatures to deal with the higher boiling point in comparison to the initial boiling point, and the lower vapor pressure in conjunction with the stronger cooling from the high evaporation heat [7].

**Ethanol-Water Fuel**

Igniting aqueous fuels requires a more energetic ignition source than gasoline or diesel fuels. A high-energy spark can initiate combustion, but the increased water vapor content in the fuel air mixture prevents complete combustion. Standard compression ignition of aqueous fuels has been unsuccessful due to problems controlling ignition timing. The catalytic igniter provides a reliable and tunable ignition source that promotes complete combustion of the mixture in the main chamber. The catalytic igniters have successfully demonstrated cold start operation and supported combustion of ethanol-water mixtures containing up to 50 percent water.

For testing purposes, the exact water content of each tank is determined and adjusted to contain 70 percent ethanol and 30 percent water. Commercially available ethanol is typically denatured with 0.0006 percent Denatonium Benzoate by weight. The ethanol used in this research was supplied by AAPER Alcohol and Chemical Company. It is 190 proof ethanol that contains 7.58 percent water by weight. The current fuel system does not monitor the water content or adjust the percentage of each fuel component in any way. Control of water content would be possible if the ethanol and water were handled in separate systems. Separate fuel storage, fuel handling systems and fuel sensors that would monitor the water...
content and make flow rate adjustments of each stream, would add excessive cost and complexity to the system. From a refueling perspective, the ethanol fuel would need to contain less water at refueling, adding additional cost to ethanol production. While there may be other benefits in the areas of emissions or combustion efficiency or engine starting, changing water concentration appears to have an insignificant effect on ignition timing. However, water content will have an effect on the rate of combustion after the onset of ignition.

**APPROACH**

The platform explored in this research is a transit van converted to operate on either gasoline or ethanol water fuel mixtures. Special attention was devoted to improve cold starting and installing additional on board sensors and equipment for future testing. System features include integration of a wide band oxygen sensor, state-of-the-art engine management system, exhaust gas temperature sampling using platinum thin film resistive temperature devices, and variable voltage control of catalytic igniters using DC-DC boost converters. Extensive engine performance and emissions testing for 70 percent ethanol 30 percent water fuel mixtures operating at air to fuel ratios (AFR) of $\lambda = 1$ and $\lambda = 1.15$ have shown a substantial reduction in NOx and CO emissions without the use of exhaust after treatment compared to gasoline emissions. Lean mixtures also show reduced emissions and increased thermal efficiency compared to stoichiometric conditions. Chassis dynamometer tests comparing thermal efficiency, and brake specific emissions of NOx, CO2, CO, and hydrocarbons for the ethanol-water fuel mixtures over a wide range of operating conditions are shown.

**Transit Van Conversion**

The vehicle test platform shown in Figure 2 was designed to store gasoline and ethanol-water fuel mixtures with the ability to operate on either fuel by way of a simple conversion process [8]. Performing tests using blends of ethanol containing as much as 30 percent water required significant engine improvements, including engine management, electrical, exhaust, and catalytic ignition systems. A diagram of each systems location is shown in Figure 3 following an in depth description of each subsystem. Substantial improvements were made to
develop an engine capable of sustained operation on ethanol-water fuel mixtures and allow data collection. The modifications to these systems allowed the test vehicle to operate under a wider range of conditions and requirements, specifically at lower temperatures and without the use of a pilot fuel for starting.

Figure 2 Multi-Fuel Vehicle Test Platorm

Figure 3 Location of Van Conversion Components
Previous vehicle testing resulted in base line comparisons of engine performance and emissions between ethanol-water fuel mixtures and gasoline [9]. However, to more thoroughly evaluate engine performance and emissions with catalytic ignition of ethanol-water fuel an emissions model was developed to calculate brake specific fuel consumption bsfc, vehicle efficiency $\eta$ and brake specific emissions of carbon monoxide (CO), carbon dioxide (CO2), hydrocarbons (HC), and nitrogen oxides (NOx) at all engine speeds (rpm) and brake mean effective pressures (bmeP).

**Engine Management System**

A user defined engine management system (EMS) allows control of engine components based on the fuel mixture of either ethanol-water or gasoline. This provides easy changing between fuels without the need for replacing fuel hardware as would be necessary with carburetion. The primary function of the EMS is to meter the fuel flow maintaining a desired air/fuel ratio (AFR) even though the total amount of air/fuel mixture varies based on engine load, speed, and other parameters. Determining the amount of fuel required is accomplished by establishing a base fuel map shown which specifies the amount of fuel to be injected into each cylinder by setting a pulse width value. The base fuel map is then modified by increasing or decreasing the pulse width by a percentage of the base map value based on system voltage, coolant temperature, inlet air temperature, and AFR when operating in closed loop mode. These modification maps are referenced continuously and additional maps including start extra, start decay, acceleration, deceleration, are also referenced during very specific points of engine operation and are a necessity to provide correct engine starting and operation [10].

The EMS manufactured by Advanced Engine Management Inc. is capable of multipoint sequential fuel injection, an improvement over the previously installed Haltech EMS and allows for increased engine performance and more control of engine operation and diagnostics. For testing and data, collection purposes the EMS communicates with a laptop computer using a serial link, which is useful for diagnostics, and logging information about
engine operation. When connected to the EMS, the laptop can also be used to display sensor information in the form of a virtual dashboard [10].

*Catalytic Igniter Boost Converter*

Cold-starting on ethanol-water fuel mixtures containing as much as 30 percent water required increased power to the catalytic igniters. Two DC-DC boost converters shown in Figure 4 were installed to enhance cold starting. The boost converters produced by Kenne Bell racing products are designed to increase the flow rate of 12-volt DC fuel pumps by supplying a higher voltage. This causes the fuel pump motor to turn at higher speeds necessary for achieving higher fuel flow rates. For this application, the boost converters are supplied 12-14 volts DC and increase the supply to the catalytic igniters to 12-18 volts DC at a maximum continuous current of 20 amps per boost converter. The catalytic igniters are controlled using dash mounted dials, one for each bank of four igniters. Although not currently connected

![Image of boost converters](image)

**Figure 4 Boost Converters for Catalytic Igniter Control**

*Exhaust After Treatment*

The exhaust system has been modified to accept removable sections containing 3-way catalytic converters located on both the right and left exhaust header collectors. To conduct vehicle testing exhaust test sections were also constructed to replace the catalytic converter sections during tuning and initial testing. Exhaust test sections became necessary because of catalytic converter failure during previous testing due to wet stacking of fuel. This condition occurred during engine warm-up and periods of fuel rich operation [11]. During
dynamometer testing and engine tuning, the catalytic converters were temporarily replaced with exhaust test sections capable of taking temperature and emissions data at the same location but not providing emissions after treatment. Sensor ports were also installed near the center of the test section for data collection purposes.

**Air/Fuel Ratio Measurements**

Previous testing has been preformed using standard automotive or narrow band oxygen sensors. The installation of a wide band oxygen sensor enabled the measurement of rich and lean mixtures accurately. The air/fuel sensors used are AEM universal exhaust gas oxygen (UEGO) sensor capable of measuring air/fuel mass ratios from 11:1 to 16.75:1. Accurate measurements in the lean region are important because for final engine operation a lean condition is desired, increasing fuel economy and reducing harmful emissions. For testing the air/fuel sensor was installed downstream of the exhaust test sections after both exhaust streams have merged. For flexibility, the air/fuel sensor can also be installed in either exhaust test section if individual cylinder bank air/fuel ratio data is desired [10].

The air/fuel sensor is also connected to the EMS allowing for closed loop engine operation. When the EMS is in closed loop mode the feedback from the air/fuel sensor is used to make immediate but temporary corrections to the base fuel map, allowing the engine to run at a predetermined and constant AFR under varying conditions not accounted for during initial fuel map calibration. Using the air/fuel sensor closed loop operation at conditions other than stoichiometric are possible. The EMS only operates the engine in closed loop mode when the engine has reached operating temperature. Until operating temperature is reached, the engine control unit (ECU) ignores the air/fuel sensor input and bases fuel injection on the preset fuel maps. The closed loop setting has two user defined parameters to customize operation, including a lean and rich limit which is a maximum percent by which the fuel maps can be modified and a feed back rate which controls how often the EMS samples the air/fuel sensor the default setting. In addition to closed loop mode, the EMS also allows the air/fuel sensor to modify the base fuel map by self-tuning the engine. This feature is very useful for building base fuel maps for different fuel mixtures once the engine is in a running condition. The self-
tuning function is only active above 5 percent throttle to allow for rich operation under idling conditions [10].

**Exhaust Gas Temperature Measurements**

To determine optimum catalytic converter placement and aid in thermal efficiency comparisons of different fuels exhaust gas temperature (EGT) sensors where installed in the vehicles exhaust system. The sensors are thin film platinum resistive temperature detectors (RTD) capable of measuring exhaust gas temperatures ranging from -40 to 1000 C°. The sensors feature a response time of 4.5 seconds when the engine has reached operating temperature. The sensors are connected to the EMS for monitoring and recording of EGT. The sensors are placed in the three locations: 1) the driver side exhaust header at the exhaust outlet, giving an approximate EGT at the entry to the catalytic converters; 2) immediately after the driver side catalytic converter; and 3) at the collector where both exhaust streams recombine. All three sensors provide valuable information, most importantly for catalytic converter placement to maximize catalyst life and effectiveness. If excess fuel present in the exhaust is causing above normal temperatures to form in the catalytic converters elevated EGT’s will be measured [10].

**Fuel Injection**

The entire fuel injection system has been modified to be compatible with alcohol-based fuels incorporating polyethylene tanks, stainless steel lines and fittings. The MSD 13.5 (kg/hr) alcohol compatible fuel injectors are regulated at 300 (kPa). The fuel injectors are sized correctly for operation on gasoline and operate between 20 percent and 80 percent of the available max pulse-width time. This setup is currently undersized for operation at maximum power on ethanol-water fuel mixtures because of the increased fuel flow requirements. The fuel metering equipment is unable to measure flow rates above 108 kg/hr, preventing data collection of high engine speed and load points while operating on ethanol-water fuel.

**METHODOLOGY**

Previously developed test protocols allowed vehicle testing that mimics the federal test protocol driving cycles. These tests would allow comparisons of fuel economy and specific
emissions between the test vehicle and other federally certified vehicles. These tests are useful when trying to make comparisons between previously published engine performance and emissions data, but for comparisons between different fuels on the same vehicle these tests can be very time consuming and unnecessary [8]. Extensive engine tuning must occur over the entire range of engine operation for each fuel composition before emission testing can be performed accurately.

Data was collected at constant throttle settings between 5 percent and 30 percent in 5 percent increments at engine speeds ranging from 1000 rpm to 2500 rpm. Due to the ethanol-water fuel chemistry, and size of the throttle body, throttle positions above 30 percent showed no change in engine operation and were therefore not included. At lower throttle settings, higher engine speed-readings were not taken because a sufficient load could not be applied accurately. Data recorded included Engine Speed, Power, Torque, Fuel Flow Rate, Air/Fuel Ratio, and Concentrations of NOx, CO, CO2, and HC. This was done to determine the bsfc, indicated efficiency, and specific emissions of each species over a wide range of operating conditions. Test conditions include gasoline at \( \lambda =1 \), ethanol-water at \( \lambda =1 \), ethanol-water at \( \lambda =1.2 \), and ethanol-water at \( \lambda =1 \) with only 7 functional catalytic igniters. The data set using only 7 functional igniters shows how harmful vehicle emissions increase dramatically even though from a operator’s standpoint the engine runs nearly identical. Although the catalytic igniters have become much more robust, they are still easily damaged during testing and maintenance. The fuel flow rate was determined using a MAX Machinery 710 series, positive displacement fuel flow meter, and exhaust species were measured using an EMS five-gas analyzer.

For consistent data collection, all measurements were taken by having the engine idle in gear until a set coolant temperature of 83 °C was reached. The throttle was then increased to the desired setting while the load was also increased at the rear wheels. When the desired load and engine speed was reached, all data collection equipment was allowed time to stabilize. Once the system had reached, a steady state operating point data was then collected.
Chassis Dynamometer Measurements

The chassis dynamometer applies a force that holds the entire power train at a constant speed. This force is measured and recorded as an average value and represents the torque output of the engine. A SuperFlow SF-602 chassis dynamometer setup in a single rear axle configuration was used for engine testing. The SF-602 uses a water-brake dynamometer capable of measuring 410 (kw) at maximum test speeds of 80 mph. The rear wheel torque is measured using a precision load cell and displayed in ft/lb [8]. Multiplying this torque value by the wheel speed as measured by optical sensors on the dynamometer rollers in rpm and dividing by 5252 results in the brake horsepower values used for calculations [11]. These values are all measured and displayed in Figure 5 on the dynamometer display.

![SuperFlow Dynamometer Display](image)

Figure 5 Display of Chassis Dynamometer Data

All measurements recorded by the dynamometer are measured at the rear wheel and therefore include all mechanical losses including transmission, torque converter, drive shaft, rear differential, rolling resistance and wheel slip. To ensure a constant value of wheel slip and rolling resistance the tire pressure was maintained at a constant 241 kPa and checked periodically during testing. Vehicle drive train losses typically range between 15 percent and 25 percent of engine power output. However, these loses are difficult to determine and change based on the drive train speed and gear in which the vehicle is operating [8]. These losses make it difficult to compare chassis dynamometer results to engine dynamometer results like those published in reference manuals, which indicate a 1986, 8-cylinder, 5.75
(liter) engine produces 157 (kw) at 4000 (rpm) and 42 (kg/m) of torque at 2800 (rpm) [12]. Since the change in fuel system and ignition are the only changes directly related to vehicle performance, test data recorded from the operation on different fuels can be made. The chassis dynamometer results will simulate results for actual on road vehicle operation excluding wind resistance and terrain, compared to simply having the engine on a test stand and recording data.

Figure 6 Model of all Equipment Installed in Vehicle Test Facility

Drive train inefficiencies reduce the rear wheel horsepower but, multiply the torque output resulting in rear wheel torques much higher than the 42 (kg/m) engine rating. The torque output of the engine is increased at three locations: the transmission, rear differential and wheel diameter. To make calculations to estimate drive train operation and performance the specifications for the drive train were used. The transmission, a C-6 three speed automatic transmission, has gear ratios of first 2.46:1, second 1.46:1, and third 1.00:1, while the rear differential ring and pinion gear provide for an additional 4.11:1 and the 235/85R-16 tire
diameter of .7874 (m) provides a final 1.292:1 [12]. Combining this system results in final drive train ratios of 13.031 in first gear, 7.734 in second gear, and 5.298 in third gear.

**FINDINGS**

Test results are displayed in the form of contour plots for each exhaust specie. On all of the contour plots the x-axis represents engine speed (rpm) and the y-axis represents bmep (bar).

To determine the accuracy of the results, a root sum square analysis was conducted. Table 1 summarizes the error associated with a single data point collected at 1900-rpm, 15.7 (kw), 19.5 kg/hr fuel mass flow rate, and at a torque of 28.9 (kg/m). This point is approximately the median of the results based on engine speed and brake mean effective pressure.

Table 1 Summary of Root Sum Square Analysis

<table>
<thead>
<tr>
<th>Variable</th>
<th>Nominal Value</th>
<th>Nominal Error</th>
<th>Percent Error</th>
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</thead>
<tbody>
<tr>
<td>bsfc (kg/MJ)</td>
<td>0.3459</td>
<td>+/- 0.0071</td>
<td>2</td>
</tr>
<tr>
<td>Indicated Efficiency %</td>
<td>16.61</td>
<td>+/- 1.47</td>
<td>9</td>
</tr>
<tr>
<td>BSE HC (g/kW-hr)</td>
<td>11.33</td>
<td>+/- 0.55</td>
<td>5</td>
</tr>
<tr>
<td>BSE CO (g/kW-hr)</td>
<td>293.9</td>
<td>+/- 13.9</td>
<td>4</td>
</tr>
<tr>
<td>BSE CO₂ (g/kW-hr)</td>
<td>2531</td>
<td>+/- 119</td>
<td>5</td>
</tr>
<tr>
<td>BSE NOₓ (g/kW-hr)</td>
<td>1.643</td>
<td>+/- 0.351</td>
<td>21</td>
</tr>
</tbody>
</table>

A detailed description of the error contribution from each piece of equipment has not been included, but the results indicate the largest source of error to be the accuracy of the ethanol-water fuel mixture. The error associated with determining the ethanol content of the mixture was found to be 2-5 percent by volume. This uncertainty accounts for approximately 95 percent of the error in all measurements except for BSE of NOₓ. The measured BSE of NOₓ is small compared to the absolute measurement uncertainty of 25 ppm. The uncertainty from the 5-gas analyzer at these low levels represents 95 percent of the total error associated with BSE of NOₓ.
**Brake Mean Effective Pressure**

The brake mean effective pressure is determined using Equation 1, where power is measured in rear wheel horsepower, $V_s$ represents the swept volume of the engine in $\text{in}^3$ and $\text{Engine speed}$ in $\text{rad/s}$ [13]. Using the calculated values of bmep and measure values of engine speed rpm, all of the data collected from either fuel at any air/fuel ratio can be shown on contour plots and compared.

$$\text{bmep} = \frac{\text{Power}}{V_s \cdot \frac{\text{Engine speed}}{2}} \quad (1)$$

**Brake Specific Fuel Consumption**

Brake specific fuel consumption (bsfc) is an important factor when comparing engines because it directly relates to overall engine efficiency. Since the reduced energy content of ethanol-water compared to petroleum-based fuels is not taken into account, direct comparisons cannot be made. The importance of bsfc for this comparison is to show the increased fuel flow rate required when changing between fuels. Values of bsfc kg/MJ are calculated using Equation 2, where $m_{\text{fuel}}$ is the mass flow rate of each fuel kg/hr and power is measured in kW [13].

$$\text{bsfc} = \frac{m_{\text{fuel}}}{\text{Power}} \quad (2)$$

Bsfc for gasoline in Figure 7 shows a minimum of 0.13 kg/MJ at 1700 rpm and 6 bar, while the bsfc of ethanol water shown in Figure 8 has a minimum of 0.23 kg/MJ at 1600 rpm and 4 bar. The lean ethanol water mixture shown in Figure 9 has a minimum bsfc of 0.20 kg/MJ at 1600-rpm and 4 bar. The bsfc under lean conditions is lower than stoichiometric conditions at most operating points. The lean mixture has a much wider overall speed and load range near minimum bsfc. This is desirable because the engine could operate as efficiently over most conditions indicating the potential of lean mixtures. The 1-cylinder misfire for the ethanol-water fuel shown in Figure 10 increases the bsfc significantly and reduces power output.
Figure 7 Bsfc kg/MJ, Gasoline, $\lambda = 1$

Figure 8 Bsfc kg/MJ, Ethanol-Water, $\lambda = 1$
Figure 9 Bsfc kg/MJ, Ethanol-Water, $\lambda = 1.2$

Figure 10 Bsfc kg/MJ, Ethanol-Water, $\lambda = 1$, 1-Cylinder Missfire
Vehicle Efficiency

Vehicle efficiency is used to compare operating conditions taking into account combustion and mechanical inefficiencies because power measurements are taken at the rear wheel. Using the values of bsfc, vehicle efficiency is calculated using Equation 3 where LHV\textsubscript{fuel} is the lower heating value of the fuel [13]. The heating value of the ethanol-water fuel mixture is calculated at 17.41 MJ/kg, making it 61 percent lower than gasoline at 44.65 MJ/kg [13]. Because of this, comparisons of vehicle efficiency are used to compare the same engine operating on different fuels.

\[
\eta = \frac{1}{bsfc \cdot LHV_{fuel}}
\]  

(3)

The vehicle efficiency plots show only minimal variation with changes in engine speed and are entirely dependant on bmep. Lean operation of ethanol-water shown in Figure 13 shows slightly higher efficiency over stoichiometric operation of ethanol-water shown in Figure 12. A maximum efficiency 25 percent for stoichiometric conditions and 28 percent for lean conditions was observed. It is not surprisingly that leaning out the mixture can slightly increase vehicle efficiency. However, vehicle efficiency with ethanol-water seems unrealistically high compared to stoichiometric operation on gasoline. A calibration of the fuel flow meter and a more accurate fuel energy content analysis needs to be conducted to rule these out as possible sources for error. Previous tests using ethanol-water mixtures in diesel engines yielded efficiency improvements ranging from 3 percent to 5 percent [2]. Similar improvements on this platform would have resulted in a vehicle efficiency of 21 percent for stoichiometric operation and 23 percent for lean operation.
Figure 11 Vehicle Efficiency percent, Gasoline, $\lambda = 1$

Figure 12 Vehicle Efficiency percent, Ethanol-Water, $\lambda = 1$
Figure 13 Vehicle Efficiency percent, Ethanol-Water, $\lambda = 1.2$

Figure 14 Vehicle Efficiency percent, Ethanol-Water, $\lambda = 1$, 1-Cylinder Missfire
Brake Specific Emissions Analysis

Evaluating ethanol-water as an alternative fuel requires comparisons of the exhaust emissions over a wide range of operating conditions. The brake specific emissions of each species needed to be calculated, because an increase in emissions between different fuels as indicated by the five-gas analyzer does not take into account differences in fuel chemistry, the rate at which species are produced, and the power output of the engine at each operating point. The molecular weight of the ethanol-water mixture is calculated at 36.21 (kg/kmol) compared to gasoline approximated using iso-octane at 114.23 (kg/kmol) [13]. The ethanol-water mixture is prepared based on volume; this requires property calculations to be modified based on the difference in densities between ethanol and water.

Using the molar flow rate the brake specific emissions of each exhaust species can then be calculated using Equation 4 where $M_{\text{W,Species}}$ is the molecular weight of the species measured. To display the entire sets of brake specific emissions data for each species contour plots of each species from both fuels are shown. The contour plots represent the BSE (g/kW-hr) of each species at a corresponding engine speed and load.

$$BSE_{\text{Species}} = \frac{\eta_{\text{Species}} * M_{\text{W,Species}}}{\text{Power}}$$

(4)

Carbon Monoxide Brake Specific Emissions

Carbon monoxide (CO) emissions were highly dependent on engine operating conditions. Gasoline emissions shown in Figure 15 were consistently higher ethanol-water emissions. Under stoichiometric conditions, ethanol-water emissions shown in Figure 16 were significantly higher than gasoline at idle conditions. Lean ethanol-water mixtures shown in Figure 17 indicate low CO formation at all speeds and loads. CO emissions for the 1-cylinder misfire shown in Figure 18 indicate that the presence of unburned fuel significantly increases CO production.
Figure 15 BSE CO g/kW-hr, Gasoline, $\lambda = 1$

Figure 16 BSE CO g/kW-hr, Ethanol-Water, $\lambda = 1$
Figure 17 BSE CO g/kW-hr, Ethanol-Water, $\lambda = 1.2$

Figure 18 BSE CO g/kW-hr, Ethanol-Water, $\lambda = 1$, 1-Cylinder Missfire
Hydrocarbon Brake Specific Emissions

Hydrocarbon (HC) emissions indicate unburned fuel in the exhaust usually from incomplete combustion. HC emissions are very dependant on combustion chamber design and only slightly dependant on engine operation [7]. Previous testing using ethanol water mixtures have indicated substantial increases in HC emissions. New developments in igniter design appear to have improved this condition. The HC emissions for ethanol-water Figure 20 compared to gasoline Figure 19 show only slight increases in HC emissions occurring primarily near idle conditions. For the ethanol-water fuel operating under lean conditions Figure 21 the emissions are at the minimum resolution of the five-gas analyzer. Discriminating emissions at these low levels would require more precise exhaust analysis equipment. The elevated HC emissions for the 1-cylinder misfire shown in Figure 22 are expected because of unburned fuel present in the exhaust with no after treatment installed. Another concern when analyzing HC emissions from alcohol based fuels is the formation of aldehydes. Aldehydes register as HC’s on the 5-gas analyzer and require FTIR emissions equipment to separate individual HC species. HC’s are simple to clean up with modern after treatment, but improvements made to reduce in-cylinder HC emissions can extend the life of after treatment systems. [7].
Figure 19 BSE HC g/kW-hr, Gasoline, $\lambda = 1$

Figure 20 BSE HC g/kW-hr, Ethanol-Water, $\lambda = 1$
Figure 21 BSE HC g/kW-hr, Ethanol-Water, $\lambda = 1.2$

Figure 22 BSE HC g/kW-hr, Ethanol-Water, $\lambda = 1$, 1-Cylinder Missfire
Nitrogen Oxides Brake Specific Emissions

Previous studies at the University of Idaho have shown water present during combustion for ethanol water fuel mixtures keeps in-cylinder temperatures down leading to decreased thermal NO\textsubscript{x} formation [14]. NO\textsubscript{x} emissions for the ethanol-water fuel Figure 24 show improvement at lower loads, but higher emissions at higher loads compared to gasoline Figure 23. For lean operation on ethanol-water shown in Figure 25, increased engine operating temperature appeared to counteract the cooling effect of the water, leading to a slight increase in nitrogen oxide emissions. The 1-cylinder misfire data shown in Figure 26 produced the highest level of NO\textsubscript{x} emissions. It is likely that combustion in the exhaust manifold was responsible for this behavior. In several cases, the measured NO\textsubscript{x} levels were too small for the 5-gas analyzer to measure accurately.

Figure 23 BSE NO\textsubscript{x} g/kW-hr, Gasoline, $\lambda = 1$
Figure 24 BSE NO$_x$ g/kW-hr, Ethanol-Water, $\lambda = 1$

Figure 25 BSE NO$_x$ g/kW-hr, Ethanol-Water, $\lambda = 1.2$
CONCLUSIONS

Catalytically assisted combustion of ethanol-water mixtures represents a possible alternative to conventional ignition of gasoline fuel in a spark ignition engine. While the clearest finding has been increased in vehicle efficiency, additional testing and a more in-depth analysis of the emissions model is required to verify these findings. The possibility of increased vehicle efficiency is substantiated in the literature [15]. Increasing the compression ratio, taking advantage of the high knock resistance of ethanol-water fuels, could further increase vehicle efficiency and reduce specific emissions.

Catalytic igniters allow ignition of fuels not possible with conventional ignition sources. Instead of reducing pollutants with after-treatment systems at the expense of engine performance and increased system cost, the formation of pollutants is controlled at the source by chemical and gas dynamic modifications of the in-cylinder combustion process. Specific emissions data for ethanol-water fuels indicates significant potential for reducing carbon
monoxide and nitrogen oxide emissions. Detailed studies of vehicle operation with after treatment technology is necessary to determine whether a similar impact on hydrocarbon emissions is possible.

The vehicle test results presented here are for only one ethanol-water fuel mixture, 70 percent ethanol and 30 percent water. Testing with mixtures of both higher and lower water content would give a more complete understanding of how the presence of water affects power output and emissions.

RECOMMENDATIONS

The vehicle emissions testing capabilities at the University of Idaho have increased substantially in recent years. This has resulted from purchase of new diagnostic equipment as well as an expanding knowledge base of students and faculty. This section outlines potential improvements in engine testing infrastructure that will benefit future vehicle research.

Fuel Flow Meter

Accurate and consistent fuel flow rate measurements are vital in determining efficiency, specific fuel consumption and specific engine emissions. The fuel flow meter used in this research is extremely accurate as seen in the uncertainty analysis, however it lacks the capacity to supply ethanol-water fueled at high engine loads and speeds. This prevented testing at speeds above 2500-rpm and brake mean effective pressures above 5 bar. The fuel system is capable of measuring flow rates up to 200 kg/hr at 2 psi, but the integrated fuel pump is significantly less than 200 kg/hr at the 40 psi input required by the fuel injection system.

Data Collection

For data collection to be accurate and repeatable, all information must be simultaneously recorded. Several measurements must be recorded from various pieces of equipment located at disparate locations in the vehicle test bay. The addition of a central computer connected to every data display would allow much more accurate data collection. The current method requires the engine to reach a steady state operating point for up to several minutes. This
allows devices like the five-gas analyzer to draw in a new exhaust sample and reach a steady and accurate reading. Although an automated data collection system would not speed up the transient response time of the five-gas analyzer, it would allow several data points to be recorded at this steady state point for the purpose of comparison.

**FTIR Analysis**

A recent addition to the University of Idaho engine test facility is a Fourier transform infrared (FTIR) spectrometer. This device will be used in research currently under way to provide a redundant exhaust species measurement and provide a complete breakdown of all hydrocarbon species produced by the combustion of ethanol most importantly aldehydes.

**Fuel Preparation**

The ethanol-water fuel prepared for vehicle testing consisted of small batches approximately 5 gallons of fuel mixed to 70 percent ethanol and 30 percent water. Previous research using ethanol-water as a test fuel has been mixed by a PR-111 refractometer and control tank apparatus shown in Figure 27. Currently the PR-111 refractometer is not operating correctly and an alternative method of measuring ethanol-water concentration was developed.

![Figure 27 PR-111 Refractometer and Ethanol-Water Mixing Station](image-url)
By measuring the volume and mass of a fuel sample, the concentration of the ethanol in the sample was determined using a table of fuel mixtures created using Engineering Equation Solver. The fuel sample volume was measured using a 2000 ml graduated cylinder with an accuracy of +/- 10 ml, while the mass was measured using a Sartorius QS32A digital scale with an accuracy of +/- .002 kg. The fuel measurement apparatus is shown in Figure 28. The mass in kilograms per 2000 ml gives the percentage of water and ethanol based on the mass of ethanol and water combined to give a total mass and volume of mixture. This method resulted in a 2 percent error in measuring the sample concentration, which is more accurate than the previously used apparatus. The only drawback of this method is the sample must be measured, adjusted, and then re-measured for accuracy. For future testing and vehicle operation, the ethanol-water fuel mixing station needs to be improved or at least repaired.

**Figure 28 Ethanol-Water Fuel Measuring Apparatus**

*Fuel Injectors*

The first set of fuel injectors used were MSD, stainless-steel, 38 lb/hr fuel injectors. These injectors are described as being compatible with ethanol, methanol and other alcohol-based fuels [8]. After 2 years of operation, the MSD injectors began to display signs of corrosion and began leaking when pressurized. Upon inspection by Rich Jensen from Cruzin
Performance, the injectors were found to have: serious internal corrosion, plugged pre-filters from fuel corrosion, solenoid valves not seating properly, leaking when pressurized, and poor spray pattern. Figure 29 shows the test results for each of the MSD fuel injectors tested, every fuel injector was found to be beyond repair and required replacement. A set of Ford Performance Racing products fuel injectors were purchased, these injectors are described as being compatible with alcohol fuels, but require purging when not in use for extended time periods. After these injectors have been used for an extended period of time they should be sent to Rich Jensen for inspection. Additional research may also yield a better suited fuel injector for this application.

![Cruzin Performance Logo]

**Figure 29** Test results After Long Term Operation on Ethanol-Water Fuel

**Particulate Emissions Testing**

Current testing equipment and procedures lack the ability to determine the particulate emissions produced during testing. An exhaust sample is taken just before the exhaust exits the tailpipe and is removed from the test facility using a suction system. To accurately measure particulate emission the vehicle exhaust would need to pass through a dilution tunnel where additional ambient air would be mixed with the exhaust allowing tapered element oscillating microbalance (TEOM) instrument to accurately measure the particulate emissions.
levels and compositions at steady state operating points. This method is very accurate and would allow the corresponding particulate emissions to be measured from changes to fuel chemistry.
REFERENCES – PART I


PART II – YANMAR DIESEL CONVERSION

INTRODUCTION

Analyzing differences in emissions and power for fuels with dissimilar composition requires an emissions model that takes into account fuel chemistry and engine power output. The model presented in this section takes into account the carbon content of each fuel when reducing emissions data. Specific emissions are useful when comparing different grades of gasoline, but can lead to confusion when comparing very different fuels such as ethanol-water and gasoline.

DESCRIPTION OF PROBLEM

The converted Yanmar engine described in this paper was formerly a CI engine. The specifications for this engine are listed in Table 2. Modifications to the converted Yanmar include milling the head to maintain the same compression ratio as the stock engine because of the additional volume of the catalytic pre-chamber. The cylinder head was also modified to incorporate pressure sensors. The final modification was replacing diesel injectors with catalytic igniters, requiring the fuel injectors be relocated to the intake manifold. A low-pressure fuel rail, controlled by a programmable fuel injection computer supplied the alcohol compatible fuel injectors.

Table 2 Original Specifications for the Yanmar Diesel Engine

<table>
<thead>
<tr>
<th>Cylinders</th>
<th>3</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>7.5 cm</td>
</tr>
<tr>
<td>Stroke</td>
<td>7.5 cm</td>
</tr>
<tr>
<td>Displacement</td>
<td>998 cc</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>17.5:1</td>
</tr>
<tr>
<td>Rated Power</td>
<td>15 kW</td>
</tr>
<tr>
<td>Maximum Speed</td>
<td>3000 rpm</td>
</tr>
</tbody>
</table>

Catalytic ignition conversions have been completed on both spark ignition and compression ignition engines. Engines converted from SI platforms control load by throttling an air/fuel mixture. The air/fuel ratio remains nearly constant across all operating conditions. As such,
converted SI platforms differ only in the ignition mechanism. In these engines, catalytic ignition allows the use of fuels not normally sustained by spark ignition. Engines converted from CI platforms are not throttled. In these engines, load is controlled solely by changing the amount of fuel delivered to the engine. The volume of air/fuel mixture in the cylinder remains nearly constant for all conditions, but the air/fuel ratio itself varies. However, fuel is injected in the intake manifold and allowed to mix upstream of the combustion chamber. Consequently, converted CI platforms operate in a homogeneous charge mode over a wide range of air/fuel equivalence ratios.

**Experimental Apparatus**

The experimental apparatus consisting of a water brake dynamometer, fuel flow meter, five-gas analyzer, and in-cylinder pressure transducers is shown in Figure 30. The apparatus is used to collect power, torque, in-cylinder pressure data, fuel flow rate, equivalence ratio and emissions data from the converted Yanmar engine.

![Figure 30 Experimental Apparatus for Yanmar Engine Testing](image)

**Engine Dynamometer**

To apply and measure a steady state torque a Land and Sea water brake absorber capable of measuring a 200 ft-lb load up to 160-hp was used. Computer control of the dynamometer allowed for a steady state torque to be applied and changed remotely while power measurements were recorded at each test point. This type of engine dynamometer works well...
for lower output engine but requires continuous calibration as the absorber temperature changes.

**Emissions Analysis**

A five-gas emissions analyzer EMS Model 5001 shown in Figure 31 provides a primary air/fuel ratio measurement and a reading of the exhaust flow based on the percent contribution of species. The five species measured include HC, O2, CO, CO2, and NOx. For the five-gas analyzer, the species measurements assume the sample is dry. Previous emissions analysis using the five-gas analyzer determined an insignificant amount of exhaust products condensed with the water according to a Henry’s law analysis, as a result the exhaust species is assumed dry [2]. A 4-channel wide band air/fuel sensor was used for measuring the equivalence ratio of the mixture in the engine. Three of the sensors monitor individual cylinder mixture, while the fourth measures the average of all the cylinders after the exhaust collects. This fourth sensor acts as a redundant measurement of the air/fuel ratio, verifying proper operation of the five-gas emissions analyzer.

![Five-Gas Emissions Analyzer](image)

**Figure 31 Five-Gas Emissions Analyzer**

**Fuel Metering System**

Engine fuel consumption was performed by a MAX Machinery 710 series positive displacement fuel flow meter shown in Figure 32. The meter chosen for its compatibility with both petroleum and alcohol fuels also provides the desired fuel pressure and displays the current rate of fuel consumption in kg/hr, and the total amount of fuel used in liters. The meter has fuel specific gravity compensation and temperature compensation.
In-cylinder Pressure Measurement

The head on the conversion engine was modified to accept PCB model 112M275 piezoelectric pressure transducers to obtain in-cylinder pressure readings for each cylinder. The pressure transducers were flush mounted in the head, but were installed through sleeves in the head cooling passages. Special purpose transducers with full Envar bodies capable of undistorted operation at the lower combustion temperatures were required to provide accurate measurements associated with water-ethanol/air mixtures were used. A 1000 pulse/revolution optical encoder was used to determine crankshaft position and trigger readings from the pressure transducers. This gives a reading every 0.36° of crank angle. Pressure and crank angle were recorded with a 200 kHz data acquisition system and fed to a PC for post processing. Monitoring three cylinders allows each cylinder to collect data at 66 kHz, which is adequate for sampling up to 4000-rpm. The encoder / pressure transducer setup provides instantaneous pressure as a function of crank angle and air/fuel equivalence ratio measurements. In-cylinder pressure measurements are useful when comparing differences between fuels in a similar engine because the maximum pressures developed can be measured and compared.
APPRAOCH/METHODOLOGY/FINDINGS

To evaluate engine performance and emissions of catalytically ignited ethanol-water fuel, two Yanmar diesel engines were rebuilt and tuned for operation on a specific fuel. Neither engine was equipped with exhaust after-treatment devices. One engine was left in stock condition, while the other was converted to catalytic operation as previously described. The stock engine was not designed for operation over 3000-rpm, which is controlled by mixing limitations in the combustion chamber. However, the homogeneous mixture in the converted engine combuts faster than the direct injection of the stock engine and higher speeds are possible. For safety reasons, the engine management computer was programmed to cut fuel to the converted engine beyond 3000-rpm. Rotating assemblies would require modification to operate safely at elevated speed.

Both engines were tested using the same protocol. Data was collected at constant throttle settings between 50 percent and 100 percent in 5 percent increments at engine speeds ranging from 1500-rpm to 3000-rpm at 250-rpm increments. At lower throttle settings, higher engine speed readings were not taken because a sufficient load could not be applied accurately. Data recorded included the following: Engine speed, power, torque, fuel flow rate, air/fuel ratio, and concentrations of O$_2$, NO$_x$, CO, CO$_2$, and HC. This was done to make comparisons of the two engines under a multitude of operating conditions.

Comparisons of brake mean effective pressure (bmem) and the power output of each engine are shown in Figures 33 and 34. Faster pressure rise as a result of increased flame speed from combustion of a homogeneous mixture, enables the converted engine to increase in bmem over all engine speeds [3]. Improvements over the stock configuration at full load ranged from 9 percent and 33 percent, in bmem and maximum power. Since the bmem is higher for the converted engine all graphs using bmem on the y-axis will have a different scale.
It seems counter-intuitive that the converted engine would produce more power because its fuel is of lower energy content. This is possible because the flow rate of fuel is much greater for ethanol-water than diesel and the increased volume of air/fuel mixture present during
At full power, the diesel engine operates lean, but because of the slow diffusion burn, there is not adequate time to combust any additional fuel. The homogeneous charge ethanol-water engine, on the other hand, produces peak power operating slightly rich (equivalence ratio of 1.1). The bmep is determined using Equation 4 where Power is measured in kW, \( V_s \) represents the swept volume of the engine in \( m^3 \), Engine Speed in rad/s, and the pressure values are converted to kPa for comparisons [4].

\[
\text{bmep} = \frac{\text{Power}}{V_s \times \text{Engine Speed}} \times \frac{1}{2}
\]  

(4)

**Brake Specific Fuel Consumption**

Brake specific fuel consumption (bsfc) is an important factor when comparing identical engines operating on the same fuel because it directly relates to overall engine efficiency. Since the reduced energy content of ethanol-water compared to diesel fuel is not taken into account, direct comparisons of bsfc cannot be made. The importance of bsfc for this comparison is to show the increased fuel flow rate required for the modified fuel system compared to the stock fuel system. Values of bsfc kg/MJ are calculated using Equation 5 where \( m_{\text{fuel}} \) is the mass flow rate of each fuel kg/hr and Power is measured in kW [4].

\[
\text{bsfc} = \frac{m_{\text{fuel}}}{\text{Power}}
\]  

(5)

The stock diesel engine shown in Figure 35 has minimum bsfc of .06 kg/MJ at 2750-rpm and between bmep of 600 and 800 kPa. The minimum bsfc of the ethanol-water engine shown in Figure 36 is 0.15 kg/MJ at a bmep of 700 to 1000 kPa over the entire speed range. The bsfc of the ethanol-water engine is 2.5 times that of the diesel but remains relatively constant over the entire engine speed range. A desirable effect because the modified engine could operate more efficiently across speed the range compared to stock operation.
Figure 35 Bsfc kg/MJ of Diesel

Figure 36 Bsfc kg/MJ of Ethanol-Water
**Thermal Efficiency**

Thermal efficiency $\eta$, is used to compare the engines allowing mechanical losses to be isolated, taking into account only the differences in fuel energy content. Using the values of bsfc, thermal efficiency can be calculated using Equation 6 where $LHV_{fuel}$ is the lower heating value of each fuel. The heating value of the ethanol-water mixture is calculated at 17.4 MJ/kg, making it 59 percent lower than diesel fuel at 42.64 MJ/kg. Because of this, comparisons of thermal efficiency are used to compare two engines instead of simply bsfc [4].

$$\eta = \frac{1}{bsfc \times LHV_{fuel}}$$ (6)

The stock diesel engine shown in Figure 37 has a maximum thermal efficiency of 37 percent around 2750-rpm and between bmep of 600 and 800 kPa. At lower pressures, the net thermal efficiency is in the range of 24 percent to 28 percent. The ethanol-water engine shown in Figure 38 has a higher maximum efficiency of 38 percent, this occurs over a wider speed range but between a narrower pressure range of 800 to 900 kPa. Under full load conditions the efficiency of the ethanol-water engine ranges from 32 percent to 35 percent. Thermal efficiency for the converted engine is lower than diesel under low-load conditions where the air/fuel mixture is very lean and the flame is easily quenched. Poor combustion efficiency is likely the cause of lower engine efficiency and higher emissions in this operating range.
Figure 37 Indicated Efficiency percent, Diesel

Figure 38 Indicated Efficiency percent, Ethanol-Water
Brake Specific Emissions Analysis

Evaluating ethanol-water as an alternative to diesel fuel requires comparisons of the exhaust emissions from each engine. The brake specific emissions of each species needed to be calculated, because an increase in emissions between different fuels as indicated by the five-gas analyzer does not take into account differences in fuel chemistry, the rate at which species are produced, and the engine power output at the desired point. To calculate the brake specific emissions of each exhaust species, a molar balance of reactants to products for NOx, CO, and HC was performed as shown in Equations 7 through 9. The molar flow rate \( n_{fuel} \) (mol/hr) shown in Equation 7 was calculated by dividing the mass flow rate by the molecular weight of each fuel. The molecular weight of the ethanol-water mixture is calculated at 36.21 (kg/kmol) compared to diesel of 193.89 (kg/kmol). The ethanol-water mixture is prepared based on volume; this requires property calculations to be modified based on the difference in densities between ethanol and water.

\[
\dot{n}_{fuel} = \frac{\dot{m}_{fuel}}{MW_{fuel}} \tag{7}
\]

The carbon content of each fuel is also required to determine a molar flow rate of reactants to products. For the ethanol-water mixture \( \text{C}_2\text{H}_5\text{OH} - \text{H}_2\text{O} \) the carbon content is 2 and for diesel of \( \text{C}_{16}\text{H}_{34} \) the carbon content is 16 [4]. The molar flow rate of each exhaust species can then be determined using Equation 8 where \( X_{\%} \) is the total percent of the individual exhaust species as indicated by the 5-gas analyzer and \( X_c \) represents the carbon content of each fuel. \( X_{\text{CO}\%}, X_{\text{CO}_2\%}, \) and \( X_{\text{HC}\%} \) are the measured percent of exhaust flow for each of the species, \( X_{\text{HC}\%} \) is multiplied by 6 because the five-gas analyzer assumes all measured hydrocarbons to be \( \text{C}_6\text{H}_{12} \).

\[
n_{species} = \frac{X_{\%} * X_c * n_{fuel}}{X_{\text{CO}\%} + X_{\text{CO}_2} + 6 * X_{\text{HC}\%}} \tag{8}
\]
Using the molar flow rate, the brake specific emissions of each exhaust species can then be calculated using Equation 9 where $MW_{Species}$ is the molecular weight of the species measured. To display the entire sets of brake specific emissions data for each species, contour plots of each species from both fuels are shown. The contour plots represent the BSE (g/kW-hr) of each species at a corresponding engine speed and load.

$$BSE_{Species} = \frac{\eta_{Species} * MW_{Species}}{Power} \quad (9)$$

**Carbon Monoxide Brake Specific Emissions**

Carbon monoxide (CO) emissions were highly dependent on engine operating conditions. The stock diesel engine shown in Figure 39 has a distinct minimum in the mid-load, high-speed range. CO emissions increase significantly at lower speed ranges but remain nearly constant with varying load. The ethanol-water engine shown in Figure 40 produces the lowest CO at higher loads independent of engine speed. The increased CO emissions at low load are likely due to incomplete combustion of the lean mixtures. At high loads extra water present during combustion from the rich conditions helps reduce CO emissions due to the water-gas shift mechanism [5].

![Figure 39 BSE CO g/kW-hr, Diesel](image-url)
Hydrocarbon (HC) emissions typically indicate unburned fuel. Due to the nature of a direct injection diesel, low HC emissions are expected, particularly at high-load and high-speed conditions. As mixtures become closer to stoichiometric, diesel engines tend to produce greater HC emissions. The stock diesel engine shown in Figure 41 never emitted more than 30 ppm HC resulting in .05 g/kW-hr at its minimum brake specific output.

HCs from the ethanol-water engine shown in Figure 42 are very high, even for an HCCI engine. Peak values were nearly two orders of magnitude greater than the diesel engine at 2700 ppm resulting in HC emissions of 30 g/kW-hr. This would suggest that there is a significant amount of unburned fuel leaving the combustion chamber. One possible cause is quenching from the excess water present during combustion. The pistons in both engines are inverted bowls designed for direct injection diesel combustion. This design is not optimized for HCCI, or catalytically ignited operation. The top of the piston that is not bowled gets closer to the head than the quench distance, and does not allow the air/fuel mixture in this region to ignite. Another concern is the formation of aldehydes that are byproducts of ethanol combustion. Ethanol byproducts register as HC’s on the five-gas analyzer and require FTIR
emissions equipment or gas chromatography to separate HC species. Hydrocarbons are simple to clean up with modern after treatment, but significant improvements could be made to the cylinder possibly reducing HC emissions. Changing piston design in the converted engine to one more typical of a homogeneous charge engine shows promise in lowering HC emissions before exhaust cleanup.

![Figure 41 BSE HC g/kW-hr, Diesel](image)

Figure 41 BSE HC g/kW-hr, Diesel
Figure 42 BSE HC g/kW-hr, Ethanol-Water

Nitrogen Oxides Brake Specific Emissions

The high levels of NO\textsubscript{x} emissions shown in Figure 43 are typical of a high compression diesel engine. A minimum value of 3.883 g/kW-hr at 2500-rpm and 761 kPa was the lowest emission per unit power measured in the test for the diesel engine. Studies at the University of Idaho have shown water present during combustion helps keep in-cylinder temperatures down leading to decreased thermal NO\textsubscript{x} formation [1]. NO\textsubscript{x} emissions for the converted engine, shown in Figure 44 show a significant reduction over all ranges of load and engine speed. In the ethanol-water engine, this is associated with a significant reduction in the exhaust gas temperature. The minimum NO\textsubscript{x} concentration of 0.2583 g/kW-hr was measured at 3000-rpm and 760 kPa. This improvement in NO\textsubscript{x} emissions results in a 93.4 percent reduction without using exhaust after-treatment devices.
Figure 43 BSE NOx g/kW-hr, Diesel

Figure 44 BSE NOx g/kW-hr, Ethanol-Water
FINDINGS/CONCLUSIONS/RECOMMENDATIONS

Catalytically assisted combustion of ethanol-water mixtures represents a possible alternative to conventional ignition of diesel fuel in compression ignition engines. Catalytic igniters allow ignition of fuels not possible with conventional ignition sources. Instead of reducing pollutants with after-treatment systems at the expense of engine performance and increased system cost, the formation of pollutants is controlled at the source by chemical and gas dynamic modifications of the in-cylinder combustion process. The original goal of reducing NOx in lean burn, high compression engines has been realized in the current conversion, but there is still work necessary to curb HC and CO emissions.

An interesting trend in a majority of the contour plots shows a lack of speed dependence while operating on ethanol-water fuels. The engine efficiency and the emissions produced remain relatively constant regardless of engine speed. This is not the case with the diesel engine, which operates most efficiently at approximately 75 percent maximum engine speed as well as 75 percent maximum engine brake mean effective pressure. The speed insensitivity of ethanol-water fuels suggests that these fuels are better suited to applications requiring variable range of operating conditions.

While the initial drive was for reduced emissions, an increase of indicated thermal efficiency and brake mean effective pressure are possible using this alternative fuel technology. These improvements are possible “because of the superior internal cooling” which improves the charging and, hence, performance” [3]. This cooling effect increases the total air-fuel mixture present during compression, which increases in-cylinder pressures during combustion. However, increased fuel flow and storage capacity are required for operation on water-ethanol. Modifications to further increase combustion efficiency and take advantage of ethanol’s ability to resist engine knock by replacing the bowl pistons with flat top pistons and increasing the compression ratio could reduce HC emissions. In this research, it is important to remember that no after treatment was deployed to clean the exhaust. The goal was to measure and control emissions at the source.
REFERENCES – PART II


PART III – STATE-OF-THE-ART LITERATURE REVIEW

INTRODUCTION/PROBLEM DESCRIPTION

Research has been conducted at the University of Idaho over the past several years relating to homogeneous charge, catalytically assisted compression ignition (HCCCI) of aqueous ethanol (Aquanol, a blend of 35 percent water and 65 percent ethanol). A catalytic plasma torch (CPT) igniter or Smartplug® was developed [Cherry and Elmore, 1990; Cherry, 1992; Cherry, et al., 1992] that uses an in-cylinder platinum/rhodium catalyst to ignite an aqueous fuel.

The purpose of this paper is to refresh our knowledge and stay abreast of work being conducted on the catalytic combustion of aqueous fuels and other topics that might aid in the understanding of our research. From an extensive literature review it was found that, outside of our group, very little work is currently being pursued on the catalytic ignition of aqueous fuels at the high temperatures needed for combustion. Research was found on the effect of adding steam or water to gas-phase hydrocarbon combustion, and on the decomposition of hydrocarbons on catalytic surfaces.

The benefits and deficiencies of using platinum or palladium as a catalyst will be reported on. Attention is also given to the catalytic ignition of hydrocarbon fuels in air over platinum, from an analysis of surface reactions to the adsorption/desorption parameters that could affect the ignition temperature, since this is of primary interest to our current research. The combustion of various hydrocarbons over platinum in a plug-flow reactor was found to be an active area of research and has provided a platform on which we can further our understanding of HCCCI.

Research found relating to the heterogeneous/homogeneous reactions of a hydrocarbon fuel with steam and a catalyst will be discussed. These conclusions will aid in the foundation for new research related to the catalytic ignition of aqueous ethanol. This research was all very closely tied together and seemed to point to the same conclusions; therefore it was not
necessary to perform an in-depth comparison of each article's discussion. We will instead provide an overview of past and present research that is related to HCCI.

Most of the work done thus far at the University of Idaho has been quantitative, by application of CPT igniter technology in various internal combustion engines. Although this research has yielded tremendous insight into the capabilities of igniting an aqueous fuel, the intricacies of improving this ignition process are not yet fully understood. To begin to create a better understanding of this process a plug-flow reactor has been built [Patel, 2004], and testing done on the combustion of propane-air mixtures over platinum to aid in the understanding of the combustion of aqueous fuels over platinum.

**APPROACH/FINDINGS**

It is a well-known fact that platinum (Pt) and palladium (Pd) are two of the best-known active catalysts for hydrogenation/dehydrogenation or total oxidation of hydrocarbon fuels [Masel, 1996; Turns, 2000]. Most of the work we found was related to exhaust cleanup from natural gas fuelled vehicles using various catalysts. However, this information is useful to our research since a percentage of steam was added to the process and the temperature was incrementally increased to determine light-off temperatures of the hydrocarbons over the catalyst. Light-off temperatures are comparable to the ignition temperature of an aqueous fuel over a catalyst.

Veser, et al., [1999], Pocoroba, et al., [2000], Velasco, et al., [2000], and Gelin, et al., [2003] showed that palladium is the preferred catalyst to use in the ignition of fuel-lean mixtures of hydrocarbons in air with or without steam. It was shown that Pd was able to ignite larger spectrums of equivalence ratios for various hydrocarbon fuels than platinum.

Of particular interest to our work is the work done by Veser, et al. [1999] where a plug-flow reactor was built parallel to that of Patel [2004] and the ignition temperature determined for various hydrocarbon fuels and catalysts. Their conclusion was that Pd showed the lowest ignition temperatures over the lowest equivalence ratios, which is where our research is
leading: there are issues cold starting Aquanol. Using a catalyst that reduces the ignition
temperature would help alleviate this problem. This would seem to contradict the choice of
Pt and Rhodium (Rh) as the catalyst in the Smartplug® application. However, the knowledge
that the melting point for Pd is 1555°C and the melting point for Pt is 1768°C and knowing
that combustion temperatures in an engine cylinder may exceed the melting point for Pd, we
wanted to choose the most durable catalyst. So, although Pd shows the capability of igniting
the leanest mixtures and the lowest temperatures it may not be able to withstand combustion
temperatures. Therefore, the next best choice is a Pt catalyst. This decision is supported by
the work of Velasco, et al. [2000] who found that the light-off temperatures and percent
conversions at 500°C for CO, NO, and C₃H₆ in an oxidizing-reducing environment were
lowest for fresh or aged Pt-Rh, as compared to Pd, Pt, or Pd-Rh. Although there seem to be
definite concerns with the effect of ageing on the different catalysts, they actually conclude
that Pd is the best ageing catalyst.

Fuel lean mixtures have the advantage of creating fewer pollutants, particularly relating to
the emission of NOₓ and CO [Turns, 2000]. Catalysts promote ignition of these fuel-lean
mixtures. Heterogeneous ignition occurs at much lower temperatures than standard spark
ignition (SI) or compression ignition (CI) engines, and also permits the lower homogeneous
combustion temperatures of fuel-lean mixtures and their advantageous emissions
characteristics [Masel, 1996]. The three main ways to control NOₓ listed by [Turns, 2000] are
time, temperature, and oxygen availability. He concludes that NOₓ can be reduced by
reducing the maximum combustion temperature, which is one of the effects of initiating
combustion of a fuel-lean hydrocarbon mixture by a catalyst.

Pocoroba, et al., [2000] and Gelin, et al., [2003] furthered catalytic combustion research to
determine which catalyst ages the best while maintaining catalytic activity. The research of
Pocoroba, et al., [2000] involved the effect of ageing for 30 days at 1000°C with 12 percent
steam on Pd- and Pt-impregnated materials with H₂, CO, and CH₄. Of interest to our research
is the effect of ageing with CH₄. They concluded that Pd showed the least amount of ageing
and higher reactivity in the temperature range 0°C to 500°C while above that temperature Pt
proved to be more active. Since homogeneous combustion temperatures are generally known to be higher than 1000°C [Veser, et al., 1999], this confirmed our decision to use Pt as our catalyst.

The research of Gelin, et al. [2003] was extremely helpful in comparing the effects of ageing with or without steam addition as this information has the potential to provide further documentation as to the durability of combustion catalysts used with Aquanol fuel. The temperature to a percentage of CH$_4$ conversion is given for both a dry and wet (10 percent water) feed. The temperature for the dry feed at 50 percent conversion of CH$_4$ with fresh Pd is 593K and 622K for aged Pd, while for fresh Pt the temperature is 698K and 843K for aged Pt. A 145K increase in the temperature was required to achieve 50 percent conversion between fresh and aged Pt and only a 29K increase for Pd. For the wet stream the temperature for 50 percent conversion of Pd is 698K and 693K for aged Pd, and for fresh Pt the temperature is 738K and 843K for aged Pt. This is quite amazing, that with the addition of 10 percent by vol. H$_2$O the aged temperature for Pd actually decreased by 5K, but the aged Pt increased by 105K. Even though there was a higher increase for the Pt catalyst, the temperature still decreased with the addition of water. These findings indicate that the addition of water to a hydrocarbon fuel has the potential of decreasing ignition temperatures.

Coinciding with the studies conducted on the determination of a noble-metal catalyst based on the requirements for fuel-lean ignition and low ignition temperature, there has been extensive research done on the parameters that affect ignition and ignition temperature [Cho, et al., 1986; Schwartz, et al., 1970; Veser, et al., 1999; Patel, 2004; Dupont, et al., 2001; Veser, et al., 1996; Bruno, et al., 1983; Dogwiler, et al., 1999; Aghalayam, et al., 2003; Li, et al., article in press; Perger, et al., 2005; Reinke, et al., 2005; Reinke, et al., 2005]. The parameters under investigation cited in these studies include: flow velocity, fuel type and concentration, and oxygen concentration. Research conducted on the catalytic ignition of hydrocarbon fuels is in agreement with Veser, el al. [1996] and Cho and Law [1986]. These researchers found a clear distinction between paraffins and olefins. The former showed a low ignition temperature at very fuel-lean conditions that peaked then decreased with an increase
in equivalence ratio up to a minimum ignition temperature near the fuel-rich flammability limit. In contrast, the latter ignited at lower equivalence ratios and reached a minimum ignition temperature under fuel-lean conditions.

Cho and Law, [1986], Vlachos, [1996], Veser, et al., [1996], Veser, et al., [1999], and Dogwiler, [1999] concluded that this was due to the surface adsorption/desorption properties of each fuel type. They determined that differences in ignition temperature were due to the platinum surface being covered by oxygen molecules or hydrocarbon molecules. This phenomenon of surface adsorption/desorption was further investigated by Perger, et al. [2005], where analytical equations were used to predict the heterogeneous ignition temperature, if you knew the adsorption/desorption properties of the gases used and had information about the test apparatus. This was accomplished by performing a heat balance of the system and from the Frank-Kamenetskii condition [Perger, et al., 2003].

Although it seems that a reliable method has been established to predict the heterogeneous ignition temperature, the homogeneous ignition temperature and distance has been much harder to predict [Reinke, et al., 2005a]. Reinke, et al. [2005a] created a 2D elliptic Computational Fluid Dynamics (CFD) program to predict the homogeneous ignition distance compared with what they found experimentally. Using the C$_1$/H/O schemes of Warnatz, et al. [1996] they were able to predict the homogeneous ignition distance quite accurately. These predictions and experimental results will be closely followed to ensure that our experimental apparatus is properly set up and is therefore consistent with the literature found in this review.

Research at the University of Idaho started with a Yanmar three-cylinder, direct-injection (DI) diesel engine by Morton [2000]. The engine was modified for homogeneous charge combustion using CPT igniter technology. Advances in the design of the CPT igniter were made, such that durability and effectiveness were increased. A 10 percent increase in maximum power output and a 10 percent reduction in NO$_x$ emissions were found while using Aquanol as compared to diesel.
This work was followed by Clarke [2001] who obtained two 20 hp Yanmar engines, a dynamometer, and a 5-gas analyzer enabling him to more easily compare the performance of Aquanol to diesel. One engine was converted to run on Aquanol and the other was left in the stock condition to be used for comparison. Clarke [2001] modeled ignition timing and focused on parameters that affect ignition timing which in turn affect the design of the CPT igniter. He found that ignition timing was controlled by the igniter length, core temperature, and compression ratio and that water concentration had little affect. With this knowledge more adjustments were made to the engine and igniter and a 50 percent increase in peak power and a 12 percent decrease in NO\textsubscript{x} was obtained. There was an increase in hydrocarbon and carbon monoxide emissions, but oxidative exhaust catalysts can control these pollutants.

The natural progressions of this research lead to the desire to test an engine in real world situations. Valley Transit of Lewiston, Idaho donated a 1985 Ford Econoline Van to the University of Idaho. Cordon [2001] determined the intricacies of converting the van to a dual-fueled vehicle, capable of operating on Aquanol or gasoline, creating a van that could be switched between fuels in less than an hour. Due to this conversion even the gasoline fuel economy and emissions were greatly improved. Test protocols for the dynamometer, fuel handling systems, and emissions analyzer were also established, so that future work could continue per a standard basis.

Williams [2006] continued by mimicking an urban driving cycle with the newly modified dual-fueled vehicle to compare the emissions and power of the two fuels. As was found with the previous research by Morton [2000] and Clarke [2001], there was a significant decrease in NO\textsubscript{x} emissions and an increase in peak power. Significant advances were made in resolving cold starting issues by adding an engine coolant heater and by boosting the igniter amperage. These additions increased the engine block temperature and igniter temperature aiding in the catalytic ignition of Aquanol under cold-starting conditions.

All of the research to date at the University of Idaho has been based on a comparison of emissions between engines fueled by Aquanol, diesel, or gasoline. One extremely important design difference to this comparison is that the gasoline engines used exhaust cleanup, i.e. a
catalytic converter. However, due to the lower combustion temperatures of Aquanol and hence the lower exhaust temperatures, the light-off temperatures of the catalytic converters were too high. Therefore, they were not useful in oxidizing any of the exhaust products from Aquanol combustion. Even without exhaust cleanup there was a decrease in NO\textsubscript{x} from all test apparatus while combusting Aquanol as compared to gasoline.

All of these analyses concluded that there were several advantages of Aquanol combustion, but cold starting remained to be an issue. The next step was to begin some qualitative work, this lead to the construction of a catalytic plug-flow reactor by Patel [2004] to investigate heterogeneous reactions and ultimately determine the ignition temperature of Aquanol with variations in flow velocity, equivalence ratio, and igniter temperature. The first step in this process however, was to use a hydrocarbon fuel and compare the experimental results with theoretical calculations of prior research [Cho, et al., 1986; Schwartz, et al., 1970; Veser, et al., 1999; Patel, 2004; Dupont, et al., 2001; Veser, et al., 1996; Bruno, et al., 1983; Dogwiler, et al., 1999; Aghalayam, et al., 2003; Li, et al., article in press; Perger, et al., 2005; Reinke, et al., 2005a, 2005b]. Patel [2004] was able to replicate this research providing confidence that experimental methods were satisfactory. His work will permit continued research using aqueous fuel under catalytic ignition in a plug-flow reactor.

CONCLUSION

It was determined from the research gathered that platinum is one of the best-known catalysts to aid in the ignition of lean hydrocarbon mixtures and potentially the ignition of aqueous-ethanol-air mixtures, at the higher temperatures needed for homogeneous combustion. Although there are some discrepancies associated with whether or not platinum catalysts are subject to thermal ageing deficiencies, it was shown that platinum is still a very well known catalyst to aid in the ignition of fuel-lean mixtures. The addition of water to a hydrocarbon fuel has been shown to decrease ignition temperatures. Also it has been shown by University of Idaho researchers [Olberding, et al., 2005; Cordon, et al., 2002; Morton, et al., 1999; Morton, 2000; Clarke, 2001; Williams, 2006] that the addition of water to ethanol has a great potential to decrease exhaust pollutants.
A clear distinction was made with respect to the ignition of paraffins as opposed to olefins due to whether or not the platinum surface was saturated with oxygen or hydrocarbons. From this discovery it was determined that the heterogeneous/homogeneous ignition temperatures are largely controlled by the adsorption/desorption parameters of the combustion mixture. Knowing these parameters the heterogeneous ignition temperature could be predicted, while the homogeneous ignition distance could be predicted.

The HCCCI of Aquanol has been shown to be a unique research topic to the University of Idaho. A great deal of progress has been made in understanding the complexities of this system. Aquanol combustion has been shown to lower NO\textsubscript{x} pollution and increase the peak power of both diesel and gasoline engines. The platinum/rhodium catalyst used in the CPT igniter technology has been very effective at igniting lean aqueous fuel mixtures. From the research at the University of Idaho, Aquanol has performed well as a renewable replacement fuel to gasoline. A foundation has been formed on which future research will thrive.

The expert research that has been gathered on the heterogeneous/homogeneous ignition of various fuels over platinum, combined with the research performed at the University of Idaho, will enable us to continue this advanced research on internal combustion transportation engines fueled with Aquanol under homogeneous charge, catalytically assisted compression ignition (HCCCI).
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