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1) Executive summary

The University of Minnesota is pleased to submit the final report for “Determining the Optimal Performance in Adapting Onsite Electrical Generation Platforms to Operate on producer Gas from Fuels of Opportunity, Concept Definition through Technology Development.”

The University of Minnesota Morris campus was selected for this study primarily due to the advanced work the Morris campus has done in developing local fuels as a replacement for conventional fossil fuel supplies. Morris collaborated within the U of MN organization to bring the best research capabilities to the project. In addition, a core group of leading private partners was brought into the project based on their knowledge and expertise in portable energy platforms.

The following report will illustrate the design and development process of comparing a nonconventional stationary biofueled combined heat and power plant with the state of the art biofueled portable internal combustion generator. The common denominator in this comparison is the focus on fuels of opportunity, which in this case is locally grown corn cobs. Both platforms used local sustainable fuel supplies to conduct the research and measure the outcomes.

Conceptually the projects are quite different in the size and scale of the intended operating environments. The biofueled combined heat and power plant is a stationary platform with significant capital investments and an operating expectation measured in decades. The stationary platform requires a reasonably predictable and continuous load. Also, the biofuels must be transported to the site.

We asked the question, what about those areas that do not have significant continuous loads and do not have access to the capital investments to produce usable biofueled energy? Could we impact the energy independence of this country by looking at other options? Since biofuels generally have a low energy density, moving biofuels has economic constraints on how far they can travel while still making collection economical. Paradoxically, there is often an inverse relationship between biofuel production areas and energy demand. Could we use a platform that could convert site based low density energy in a distributed generation scenario to electrical energy which can be moved to where it is needed? Is there a biofuel energy producing platform that could be taken to the source of the fuel? If there were such a platform, what would its energy costs be compared to the conventional stationary systems? Would the emissions be better or worse for the environment? Could these platforms be dual fueled to take advantage of either fossil or biofueled options? This grant provided the roadmap to answer these questions.

Very significant design questions had to be answered to understand the feasibility of a biofueled portable reciprocating engine generator. There have been many attempts to use standard producer gas in internal combustion engines. We asked the U of MN Center for Diesel Research

to take the lead in exploring how we might use producer gas fumigation to mitigate some of the well-known problems of using producer gas in diesel engines. Their pioneering research on producer gas fumigation and the corresponding reductions in nitrous oxides and soot production pointed the way forward. At the same time we engaged the technical expertise of All Power Labs, who have been working on the application of advanced computer controls to manage the gasification process in an experimental down draft gasifier. This jump to new gasifier designs and more accurate controls set the stage for the application of the next generation gasifier. We asked the engineer of record for our state of the art CHP biomass plant, Hammel Green & Abraham (HGA), to assist with technical coordination and energy analyses for the two systems.

Finally, we turned to a world recognized leader in portable generator design and manufacture. Cummins Power Generation agreed to work with the grant team to provide the state of the art capability in diesel generation platforms.

This grant provided for the convergence of content experts in the area of biofuels (U of MN Morris), diesel research (U of MN Center for Diesel Research), leading edge gasification systems (All Power Labs), and power generation experts (Cummins Power) to come together in an environment to explore and answer the questions raised in the above paragraph. To answer these questions, each content provider had to think differently about their product or process. The process of discovery lead to the confirmation of some assumptions and the rethinking of others. The research simply required a different thought process or a wholesale changing of conventional understanding. There were lessons learned and knowledge gained.

On the economic front, we were surprised at how inexpensively we could produce biofueled power from our stationary combined heat and power unit. We were not surprised to see the cost of the portable generation exceed most rates provided by utilities.

On the environmental front, emissions from biofueled sources had significant improvements in CO, NO_x, and soot production over fossil fuels.

On the operational side, we were pleased to see the new generation gasifier produce clean gas that would not foul the turbocharger in the diesel genset. We were also pleased to see that there was no significant derating of power using up to 80% producer gas. The details of these discoveries and challenges are included in the attached final report.

The University of Minnesota and its partners appreciate the opportunity to conduct this research through the funding provided by this grant award.

2) Goals and accomplishments

Goal: Test operation of a bench engine on syngas with conventional injection and HCCI

The Center for Diesel Research conducted extensive experimental work on the use of syngas in a diesel engine. Two published papers have come out of that work:

Bika, A, Franklin, L, and Kittelson, D. 2012. Homogeneous charge compression ignition engine operating on synthesis gas. Intl. J. Hydrogen Energy 37:9402-9411. Elsevier.

Fang, W, B Huang, D. Kittelson, W. Northrop. 2012. Dual-Fuel Diesel Engine Combustion With Hydrogen, Gasoline and Ethanol as Fumigants: Effect of Diesel Injection Timing. Proc. ASME ICEF 2012. ASME.

These papers are attached in their entirety to this report. Among the significant findings:

A single cylinder compression ignition engine was modified to run as a homogeneous charge compression ignition engine. The engine was operated on blends of hydrogen and carbon monoxide which varied from 100% H₂, 75/25 H₂/CO ratio, and 50/50 H₂/CO ratio, by volume. Two equivalence ratios were investigated; 0.26 and 0.30. The combustion characteristics and efficiencies were measured for each condition. The following conclusions were drawn from this work:

For all of the conditions tested, increasing in-cylinder peak pressure, in-cylinder peak temperature, and increasing peak HRR were seen with increasing intake air temperature, which led to an advancing start of combustion.

The 50/50 H₂/CO ratio conditions required a roughly 20°C higher intake air temperature to achieve stable engine operation, compared to the 100% H₂ conditions. The 75/25 H₂/CO ratio conditions exhibited similar trends and required a roughly 10°C increase in intake temperature to achieve stable engine operation, compared to the 100% H₂ conditions.

The peak in-cylinder temperatures for all of the conditions ranged from roughly 1200 K to 1500 K depending on intake air temperature, mixture concentration, and fuel composition.

Increasing CO fraction in the mixture increases its autoignition temperature so that a higher intake temperature is required compared to pure H₂.

Increasing CO fraction in the fuel does not have a significant effect on the rapid burn angle. All conditions were within 2 CAD of each other.

The combustion efficiency for all conditions tested was between 83% and 88% and increased with increasing H₂ fraction.

A substantial fraction of fuel energy leaves the system as heat transfer to the cylinder walls, caused by low load engine operation and high intake air temperatures.

These and other such findings played a major role in the adaptation of engine controls to run the PowerTainer on producer gas.

Goal: Develop gasifier of sufficient capacity with innovative heat recovery system

Picture 2.1: PowerTainer Internal Layout and External View.

Gasifier Type and Architecture

The PowerTainer 100 kW gasifier is a highly modified fixed bed downdraft gasifier, with a multistage, multipoint heat recycling system. Waste heats from the outgoing syngas and engine exhaust are captured and returned to appropriate process zones in the reactor. Drying and pyrolysis thermal loads are handled via these “external” heat sources, not parasitically off the combustion zone.

This waste heat recycling and gasifier-engine integration results in higher reactor temperatures for improved tar conversion, increased tolerance for high moisture fuels, and meaningful increases in total system efficiency. These wins mean the gasifier solves the tar problem in the reactor, instead of a large downstream filtering system. Similarly the usual add-on gas cooling system is eliminated by cooling the gas with incoming air and biomass fuel.

These improvements to the total gasifier system significantly reduce the size, cost and complexity of needed components, as well as the resulting installation footprint.

Core System Components

- Gasifier - All Power Labs 100 kW downdraft gasifier
- Gasifier Datalogging and Control - All Power Labs Process Control Unit (PCU)
- Engine Control - Cummins PowerCommand 1.1
- Genset - Cummins QSB7-DSGAA genset with 6.7L Turbo Diesel and 3-phase generator rated to 100 kW stand by on diesel

Fuel Feed System

The shipping container enclosed hopper holds approximately 400 cu/ft of biomass, usually adequate for a 24+hr run at full load. Material is fed into the gasifier from the hopper via a chain-drag conveyor, feeding a rotary air lock, through a heated auger system, and into the gasifier. Use of an airlock for feed to the gasifier allows the hopper to operate at ambient pressure, and therefore use the warm air from the genset radiator to dry incoming fuel.

With proper configuration, the dual capacitive sensors were found to perform effectively for fuel input sensing and control.

Fuel Characteristics and Performance

During operation onsite at the Morris campus, one fuel was used - corn cobs. Testing with this fuel showed an angle of repose that was higher than that for which the system was designed, which caused bridging within the hopper. This required installation of agitators to promote flow of fuel. Corn cobs generated a structurally stable charcoal, important for maintaining gas flow without blockage in the gasifier hearth.

Figure 2.1: Photo of the chopped corn cob fuel used during testing.

Gasifier Performance

Extensive testing of the combined gasifier and genset system was conducted with tests on December 5th and 6th, 2012.

The gasifier was found to maintain stable temperatures and very good gas flow through the hearth across the gasifier load range. By reviewing data by the number of kWe offset by producer gas we can understand the dependence on reactor conditions with load.

Producer gas had a mean HHV of 6.70 MJ/m^3 (SD=0.37) during test periods on 12/5 and 12/6 (Table 2.1).

Gasifier producer gas composition was seen to have little dependence on test conditions.

	HHV [MJ/m^3]	H ₂ [%]	CO [%]	CO ₂ [%]	CH ₄ [%]
Mean	6.70	19.15	24.98	9.61	1.95
SD	0.37	1.88	2.23	2.67	0.70

Table 2.1: Gasifier gas composition across all test conditions

Substitution [kWe]	T _{rest1} [°C]	T _{red1} [°C]	T _{red5} [°C]
0-20	843	818	720
20-40	867	842	727
40-60	868	842	737
60-80	874	836	739
80-100	881	835	753

Table 2.2: Gasifier Temperatures vs. Substitution [kWe]

Substitution [kW]	P _{comb} [Pa]	P _{reactor} [Pa]
0-20	-420.88	-592.01
20-40	-831.33	-1190.08
40-60	-996.25	-1496.62
60-80	-1140.5	-1661.79
80-100	-1659.6	-2452.07

Table 2.3: Gasifier Pressures vs. Substitution [kWe]

In figures 2.2 through 2.6, dot = median value, lower/upper box bounds = 1st/3rd quartile.

Figure 2.2. Producer Gas Higher Heating Value vs. Load and Producer Gas Substitution.

Figure 2.3: Producer Gas Carbon Monoxide Concentration vs. Load and Producer Gas Substitution

Figure 2.4: Producer Gas Hydrogen Concentration vs. Load and Producer Gas Substitution

Figure 2.5: Producer Gas Methane Concentration vs. Load and Producer Gas Substitution

Figure 2.6: Producer Gas Carbon Dioxide Concentration vs. Load and Producer Gas Substitution

Gas composition saw no significant shifts across changes in gasifier loading during tests. Peaks in HHV and H_2 content were seen when system was temporarily shut down for engine hand-over outside the test periods.

Goal: Gasifier and Genset Integration

System integration used original diesel governor control to maintain speed control. Producer gas was manually fumigated through valving into the intake air stream, with the governor control compensating for added fuel value from producer gas fumigation.

The diesel injection timing and turbo boost control was not optimized for testing.

Goal: Demonstrate and test the fully functional PowerTainer

Raman Laser Gas Analyzer – ARI system 124A (RLGA), ~2-3 minute time resolution

NOx Analyzer – Greenline 8000l, 1 second time resolution

Soot Analyzer – Artium LI-300 - Laser Induced Incandescence (LII)

Load Bank - Make/Model – Mosebach 100 kW, 1 second time resolution

Figure 2.7: Experimental Design (Solid Dot - Nominal Treatment Value, Cross - Measured PG Substitution Data per Test, Open Circle - Diesel)

Figure 2.8: Time series showing data from primary testing sequence on 12/5/2012.

Labels show treatment and test (P[Load]-D[Rate] [Test #]). Vertical lines show time data was sampled for test statistics (solid - start, dashed - end).

Test design on 12/5 (shown in following figures) consisted of a warm up phase of the engine and gasifier. Tests across loads and fuel rates consisted of ramp down in load bank loading from 75 kWe, 50 kWe, 25 kWe, and 12.5 kWe, and up again. For each load, fuel rate started at the lowest test level and ramped to highest below the pure diesel fuel rate, at rates of 1, 1.5, 2, 3.25, and 5 gal/hr. Substitution rate was controlled by the operator.

Data Collection Handling

Data was collected from separate instruments to local computers. Time resolution of data was 1 second (PCU, LII, NOx, PowerCommand), and 1.5-3 minutes (LGA) depending on number of active gas sampling ports. Engine ECU data (fuel rate, boost, speed) was transferred over CANbus and logged by the PCU.

All data from 12/5/12 and 12/6/12 were merged to a common 30 second time base. LGA data was linearly interpolated. NOx, LII, and PCU data was filtered with a constant weight 30 second window and then sampled at the common time base.

Treatment statistics were taken 2 minutes after the test start time until test end time.

The resistive load bank provided a consistent load, although measured load was up to 2.3% below the nominal setting for the given load (see Table 2.4 below).

Nominal Load [kWe]	25.0	50.0	75.0	100.0
Measured Load, Mean [kWe]	24.43	49.12	73.46	97.93
Measured Load, SD [kWe]	0.01	0.02	0.03	0.08

Table 2.4: Nominal vs. Measured Load provided by load bank

Gasifier Conditions

The following figures demonstrate the gasifier temperature profile and producer gas composition over the course of a stable run.

Figure 2.9: Gasifier Temperature Profile

Figure 2.10: Producer Gas Composition for Carbon Dioxide, Carbon Monoxide, and Hydrogen

Figure 2.11: Producer Gas Composition for Methane, Oxygen, Water, and Higher Heating Value

Producer gas composition was stable over testing. Peaks in composition were seen during low or no gasifier load when hot during engine hand-over or temporary shutdown.

Genset Operation with Producer Gas Substitution

Figure 2.12: Exhaust Gas Composition over a Sample Stable Run

Figure 2.13: Exhaust Gas Composition over a Sample Stable Run (cont.)

Figure 2.14: Fuel Substitution Rates over Five Load Rates

Figure 2.15: Engine Boost Rates over Producer Gas Substitution Ratios for Five Load Rates

Figure 2.16: Oxygen Exhaust over Producer Gas Substitution Ratios for Five Load Rates

Figure 2.17: Carbon Dioxide Exhaust over Producer Gas Substitution Ratios for Five Load Rates

Figure 2.18: Soot Exhaust over Producer Gas Substitution Ratios for Five Load Rates

Figure 2.19: Carbon Monoxide Exhaust over Producer Gas Substitution Ratios for Five Load Rates

Stable genset operation was found to occur down to 1 gal/hr diesel with up to 75 kW load. Unstable operation was characterized by high soot emissions and unstable fuel rate and generally occurred at high substitution. Engine operation at 100 kW above 40% substitution was unstable, with suspected causes of mixer/fumigation dynamics leading to unstable mixtures with temporary very high substitution levels (<1 gal/hr).

A maximum of 85% stable fuel substitution was shown (75 kW, 1 gal/hr diesel) with relatively low soot and CO emissions.

The following figures illustrate exhaust gas composition profiles over fuel substitution ratios for five different load rates. We see that the O_2 and CO_2 measured with the RLGA respond as expected, with the O_2 decreasing and the CO_2 increasing as the load is increased.

Figure 2.20 (5 panels): Exhaust Gas Composition Profiles over Producer Gas Substitution Ratios for Five Load Rates

Oxygen and carbon dioxide levels are clearly inversely correlated, with O_2 decreasing with increasing load, and CO_2 increasing with increasing load and substitution. CO measured 0% on diesel, suspected to be below the resolution of the RLGA. Mean soot levels on diesel across loads were measured at 18.5 g/m^3 ($SD=4.1$).

Emissions testing showed that optimal emissions were achieved on either pure diesel or maximum substitution at higher loads. The diesel cycle is fuel governed with constant air intake. At low loads, low CO and H_2 content may be below flammability limits, allowing high slip and high exhaust CO content. As load and substitution increases, CO and H_2 concentrations allow completion of combustion. Free oxygen on diesel was as low as 12% at 100 kW, with substitution free oxygen decreased to a low of 5%. Low free oxygen was found to correlate to

high soot output, but CO and H₂ emissions were found to continue decreasing under decreasing oxygen content. Soot formation increases with decreasing free oxygen, and should decrease with decreasing diesel fuel rate (seen clearly at 50 kW load) which implies that maximizing achievable stable substitution levels should decrease soot production levels.

The soot concentration measured with the LII stays fairly constant at around 20 mg/m³, but does spike at each load increase as expected. The NO readings from the electrical cell are somewhat sporadic as the instrument auto zeroed during this time interval and then returned to a lower level, but the trend is upward with increasing load as expected. These results are summarized in Table 2.5 and this data used as a baseline for comparison with operation with produced gas substitution. This data consists of a three minute average and standard deviation taken at the end of the time at each mode.

Load SP		Load	LII M. C.	NO	NO _x	% CO	% CO ₂	% O ₂
12.5	Avg.	12.3	21.6	95.8	121	0.0	2.97	16.5
	S.D.	0.012	0.25	0.39	0.13	0.00	0.00	0.01
25	Avg.	24.8	23.1	120	148	0.0	3.75	15.4
	S.D.	0.010	0.31	0.47	0.42	0.00	0.01	0.02
50	Avg.	49.2	21.7	159	180	0.0	4.90	13.7
	S.D.	0.012	0.52	1.61	1.62	0.00	0.01	0.01
75	Avg.	73.6	19.0	209	226	0.0	5.68	12.6
	S.D.	0.021	0.34	2.56	2.62	0.00	0.00	0.01
100	Avg.	98.0	18.5	294	314	0.0	6.42	11.6
	S.D.	0.013	0.33	1.83	1.67	0.00	0.02	0.03

Table 2.5: Exhaust Gas Composition Profiles over Five Load Rates

In figure 2.21 the results shown in Table 2.5 are extended to cover the entire run. In this figure we can see how unsteady the operation is once the producer gas is introduced into the engine.

Figure 2.21. Load and emissions vs. time over entire run.

The producer gas substitution rate was varied at each of the steady state load points. In figure 2.22 only the time period with the producer gas substitution is shown. Generally the substitution rate was increased over the time at each steady state load. This can be inferred here from the torque reading logged from the ECU, which is based on the fuel rate. The ECU believes that the load is reduced as it is providing less fuel to the engine as the amount of producer gas is increased by the operator.

Figure 2.22. Load and emissions vs. time with producer gas

Load SP		Load	LII M. C.	NO	NO _x	% CO	% CO ₂	% O ₂
12.5	Avg.	12.3	8.2	18	63	2.6	5.1	13.5
	S.D.	0.010	1.01	4.1	4.1	0.24	0.07	0.10
25	Avg.	24.8	6.4	47	101	2.0	6.7	12.0
	S.D.	0.027	0.62	2.0	1.5	0.04	0.05	0.06
50	Avg.	49.2	20.5	--	--	1.0	9.3	10.1
	S.D.	0.054	1.78	--	--	0.12	0.11	0.17
75	Avg.	73.7	24.29	175	247	0.4	11.7	8.1
	S.D.	0.042	2.24	7.3	8.0	0.22	0.55	0.74
100	Avg.	98.0	80.5	143	178	0.6	10.2	8.2
	S.D.	0.744	22.2	12.0	11.2	0.14	0.78	0.79

Table 2.6: Emissions vs. Load with Maximum Producer Gas Substitution

In figures 2.23 - 2.25, the data from Tables 2.5 and 2.6 are compared graphically. The CO data is not graphed due to the fact that there was no CO measured for the diesel only modes.

Figure 2.23. LII exhaust soot concentration vs. load comparing diesel only to max producer gas substitution

Figure 2.24. Exhaust NO concentration vs. load comparing diesel only to max producer gas substitution

Figure 2.25. Exhaust CO₂ and O₂ concentration vs. load comparing diesel only to max producer gas substitution

In figure 2.26, a similar data set is shown with the addition of CO and H₂ in the producer gas as well as the engine fuel rate. We can see that the fuel rate is identical to the torque just with a different scaling factor. The variation in the CO and H₂ concentrations in the producer gas adds an uncontrolled factor to the testing which limits the ability to directly interpret the data as a function of substitution rate.

Figure 2.26. Load and emissions vs. time with producer gas

3) Project Narrative

Synopsis of the project

Overall purpose and expected outcomes

The overall goal is to develop an energy platform combining biomass gasification and a diesel engine-powered generator into a transportable source of 100 kW power. This energy platform will be easy to install and will use almost any local biomass fuel source. It could be deployed in industrial, agricultural, educational, municipal, and other local applications. This novel approach to onsite power generation includes: adaptation of the diesel engine to run on producer gas, a fuel normally used only in spark-ignition engines; development of advanced gas cleaning technology to prevent damage to critical engine parts (e.g., the turbocharger); improvement of homogeneous charge compression technology for use with producer gas in the diesel engine; and the development of a compact gasifier that will produce exceptionally clean gas from a variety of biomass fuels. Ultimately, the goal is to develop a robust system that will expedite the deployment of distributed renewable electrical generation, thus reducing greenhouse gas emissions, reducing dependence on fossil fuels, and improving the nation's energy security.

Q1, 2010

Representatives of the principal partners met at the Cummins Power Generation (CPG) headquarters in Fridley, MN. Attendees included Jim Barbour and Dave Aronson from the University of Minnesota, Morris (UMM); David Kittelson and Anil Bika from the Center for Diesel Research (CDR) at the University of Minnesota, Twin Cities; John Pendray, Brooks Simning, Madhukar Mahishi, Charles Vesely, and Paul Plahn from CPG; and Jim Mason from All Power Labs (APL). The group discussed partners' roles in the project and began to plan the development of a producer gas that is sufficiently clean to fuel a turbodiesel engine driving a generator to produce 100 kW of electricity. The partners also discussed the project timeline and communication among the partners.

By the end of 1Q2010, APL had begun bench testing of component designs for the gasifier. They maintained regular contact with UMM. By this time, DOE and UMM had not yet reached a contract agreement. Since the partners were risk averse regarding spending before the formal agreement was complete, little apparent progress was made. Although the project was delayed, we expected rapid progress once the final contract was in place.

Q2, 2010

During a project meeting on May 6, 2010, at the offices of HGA Architects and Engineers (HGA), the principal partners and team members reviewed each of their roles in developing the project and continued discussion of the various possibilities for developing a producer gas that could be used in a diesel engine to run a generator set to produce electricity. APL continued development, bench testing of components, and some fabrication on the new gasifier. Jim Mason at APL continued integration work on a smaller scale in anticipation of receiving a diesel engine and generator from Cummins in order to build the larger integrated system. Professor Arne Kildegaard (UMM) and his student assistant began conducting background research to provide a document that summarizes and identifies prior biomass research on a national level, prior

research studies at the state/regional level, and candidate industries in our region with potential flows of biomass waste that could be considered fuels of opportunity for the sake of this project. The official modification of the original grant award was received on June 15, 2010, which poised the group to move from primarily preparatory and planning work to project implementation. A dramatic increase in activity by all parties was thus expected and the initial delays were documented in the Timeline and Management Plan.

Q3, 2010

The staff at APL focused on continued refinement of the automation system for the gasifier. CPG worked with the other partners to select an appropriate diesel genset for the project. The decision was taken to use a 90 kW genset with 208 VAC output reconfigurable to 480 VAC. CDR modified 2 engine test stands for testing syngas in a variety of operating regimes: spark ignition, compression ignition, and homogeneous charge compression mode.

Q4, 2010

The UMM staff worked with all partners individually to address any issues specific to a partners' role. UMM also placed purchase orders for the Raman Scattering Laser Gas Analyzer and the Laser Induced Incandescence Carbon Analyzer. Both pieces of equipment were being built and were scheduled to arrive in the next quarter (Jan.-March). Hammel Green & Abraham (HGA) staff monitored project progress and began design of the equipment layout and installation process at the UMM test site.

The staff at APL focused on continued refinement of automation systems and reliability tests for the prototype scale gasifier/genset system. This work included modification of the grate, exploratory work on waste heat capture to enhance the process efficiency, and auger boot development and testing. All the solutions developed were transferable to the full scale unit.

CPG primarily focused on resolution of outstanding issues to allow scheduling of the production of the diesel genset that the team agreed upon for the project. There were initial plans to use a 2.4 liter diesel powered genset in the full scale unit. However, as the project moved forward it was determined that the project would be better served by replacing the 2.4 liter genset with a 6.7 liter genset for several reasons. The larger genset uses electronic controls instead of the mechanical controls used in the 2.4 liter genset. Electronic controls give greater flexibility in controlling critical engine performance parameters such as fuel rail pressure (and hence the injection pressure), fuel injection timing, duration of injection, pilot and post injection fuel quantities, etc. The ability to control these parameters (either independently or together) is vital for the success of the project. Another important project requirement was the ability to connect the genset to the utility grid. This required a special controller (PCC3300) that had the necessary grid paralleling features. Also, the 6.7L engine provides a higher power output (100 kWe).

CDR modified an Isuzu engine to operate on blends of syngas and diesel fuel. CDR developed specifications for a diesel fuel injection software/hardware system to enable the multiple injection strategy needed for the high efficiency clean combustion of syngas and diesel blends.

There was also much work on revising the project timeline, given both the delayed start because of finalizing the grant award and the very complicated process of getting the genset ordered,

developed and fabricated. This process delayed a conversation we were hoping to have with our NETL program and contracting officers in this quarter, but plans were made to postpone that conversation until the next quarter.

Q1, 2011

CPG built the genset per the specification mutually agreed upon by all project partners. The genset has a 6.7L diesel engine and a PCC 3300 controller that can parallel the utility grid. The rated power output of the genset is 100 kWe. The genset was shipped to All Power Labs in March 2011.

APL mostly paused during the quarter while waiting for the arrival of the diesel genset from CPG. APL did engineering work laying out components for a container based system. In early March, HGA received the genset manuals from Cummins and identified interface points for power and control on the equipment. In late March, HGA visited the UMM site and evaluated potential locations for the container system near the boiler plant. Consideration was given to fueling the system, traffic circulation, and the technical requirements for connecting the system to the campus grid. A site at the northeast corner of the existing boiler plant nearest the campus grid tie-in point was deemed the best option. The second location that appeared viable was at the southeast corner of the walking floor fuel handling system. While this location is viable, it would make traffic circulation somewhat more difficult and the distance to the grid interface would require considerable expense for trenching a cable.

CDR modified a 4-cylinder 5.2L Isuzu engine to operate on blends of syngas in HCCI mode. The engine was modified to operate in a single cylinder mode with the remaining three cylinders deactivated. A new diesel fuel injection system was installed on the engine. Performance, gaseous emissions, and particulate emissions data were collected with the engine using hydrogen and various blends of simulated syngas. As UMM waited for the arrival of the laser gas analyzer and the carbon analyzer, personnel shipped a ton of corn cobs to APL so they could begin testing the gasifier on the fuel that will be used at UMM.

Q2, 2011

At the end of March, the Cummins genset arrived at APL. APL began final layout work on the container to accommodate the genset and began modification of the container. APL also delivered a 10 kW Power Pallet (a small gasifier/genset unit with a spark ignited engine) to CDR. The Power Pallet will be shared by CDR and UMM for demonstrations, teaching, and research following the project conclusion. Jim Mason of APL traveled to CDR for the final assembly and initial test runs of the unit. During the testing, problems with the air/fuel mixture (λ) control were discovered and solved. This experience went directly back to APL and the new λ control system was implemented on what had come to be called the “PowerTainer.”

On 17 and 18 May, representatives from CDR, UMM, HGA, CPG, and APL observed the operation of the Power Pallet at CDR. The team then met to discuss lessons learned from the Power Pallet that would transfer to the PowerTainer.

Due to limitations on connection to the Raman Scattering Laser Gas Analyzer (RLGA), a new location for the PowerTainer was chosen at the southeast corner of the Heating Plant. UMM received the RLGA during the quarter. Installation was scheduled for Q3.

The project PI and the Sponsored Projects Officer at UMM worked with DOE/NETL personnel regarding a proposed Budget Period 1 extension and resultant timeline modifications. The extension and new timeline were approved by DOE/NETL in Q2. A request for revised budgeting of some items was submitted in April. By the end of Q2, the project team was awaiting final approval.

Q3, 2011

The revised project management plan was approved in Q2 and the modified budget was approved in Q3.

Project PI Lowell Rasmussen sent Jim Barbour to APL in Berkeley, CA, in early August 2011 to review the status of the construction of the gasifier/genset and shipping container and to coordinate design features with other team members. Project PI Lowell Rasmussen and Dave Aronson consulted frequently with Jim Mason at APL and with Doug Maust and Scott Wheaton at HGA and other staff at UMM regarding the design of the PowerTainer and site preparation issues at UMM. We confirmed that APL would have the PowerTainer delivered to UMM by late November 2011.

In Q3 APL focused on fabrication and assembly for all systems relating to the shipping container gasification genset (PowerTainer). This included modifications to mount the genset, gasifier and hopper within the container. APL also made container modifications for a broad side door access with robust ventilation. Work continued on fabrication, testing and modification of the gasifier reactor and filtration system. APL also assembled control/automation electronics and designed and fabricated a fuel feed system including multiple auger stages and rotary airlocks and completed interconnection of plumbing and valving.

While work on the PowerTainer continued in Berkeley during Q3, APL continued collaboration with other project partners to refine the siting and grid interconnection solutions for the UMM installation.

Engineers from HGA worked with UMM staff and code review officials to select the best site for the PowerTainer and reviewed all code issues. HGA staff prepared construction documents for the electrical work needed to connect the PowerTainer to the campus grid. HGA prepared shop drawings and submitted them for review to the code office staff and made adjustments as recommended by code officials.

At the UMM campus, preparations began for the installation of the RLGA which was on site. The LII 300 Laser-Induced Incandescence Spectrophotometer also arrived at UMM. Set up and training was scheduled for October 2011. UMM staff accumulated about 200 hours of baseline testing on the steam turbine in preparation for head-to-head testing of the PowerTainer and the steam turbine.

During Q3 APL and HGA staff worked on system integration of the gasifier and the genset and planned for the power connections at UMM. Initially, CPG staff were to provide support for controls integration. In the modified plan it was mutually agreed at the biweekly meetings during the quarter that it would be more valuable to have CPG staff support for control integration once the PowerTainer was in position at UMM. CPG had previously shipped the 6.7 L diesel engine to APL for integration with the gasifier and during Q3 were largely in a holding pattern until the PowerTainer was to be shipped to UMM in November.

During Q3 CDR continued tests on the Isuzu 4HK1-TC engine to determine combustion characteristics and PM emissions over the operating range for various fuels. This is in preparation for using synthesis gas as the main fuel. The first objective of the tests in Q3 was to examine the performance of the diesel fuel injection control system that was installed in June and July. Initial challenges in achieving stability resulted in CDR personnel adding a low pressure fuel pump to the fuel supply system which solved the problem such that the injection parameters could be controlled to the set point values. A series of tests with diesel fuel were conducted to set up a baseline for the following fumigation and reactivity controlled compression ignition (RCCI) experiments.

The exhaust gas recirculation (EGR) system was modified since the engine was changed to the single cylinder version. EGR rate sweep and fuel injection timing sweep tests were performed to investigate the effects of EGR on combustion and emission characteristics. Low temperature combustion was achieved by the utilization of high EGR rate and very late injection timing, which led to very low NO_x and soot emissions, but efficiency was sacrificed. The second objective was to realize premixed charge compression ignition (PCCI) and reactivity controlled compression ignition (RCCI) combustion strategies with ethanol and gasoline as primary fuel and small amount of diesel as ignition source. This set of experiments is important and valuable for the future work based on syngas and diesel dual-fuel low temperature combustion modes. Tests were conducted to determine the effects of varying injection pressure, injection timing, and diesel fuel percentage. Tests were also conducted to determine how variations affected emissions and engine efficiency.

The proposed extension and timeline modifications were approved in Q2 and the request for the budget revision received final approval in Q3. Budget period 1 (stage 3) would now end December 31, 2011. Budget period 2 (Stage 4) would begin January 1, 2012, and run through December 31, 2012.

Q4, 2011

With the assistance of the NETL program office, we completed and submitted our application for continuation into Stage 4 of the project.

Cables were installed for the interconnection of the PowerTainer genset to the grid. Other site preparation was completed.

The genset, named the “PowerTainer” by APL, arrived in Morris in December. Corn cobs were procured from a local farmer. The cobs will fuel both the genset and the steam turbine.

The LII was taken to the CDR for setup, testing, and calibration against CDR's two different soot analysis instruments. The LII performed flawlessly in its operation, but the vendor representative was not satisfied with its calibration. The instrument was returned to the factory for recalibration.

We also continued the preliminary testing of the steam turbine to understand how it works and how well we can collect data from it. The steam turbine was operated a total of 509 hr during Stage 3. Of this time, installation, break-in and training took 144 hours. There were 365 hours of normal operation. Monitoring and collection of performance data were done over a period of 247 hours, which is 67.7% of the available operating time. The raw data are kept in the Operator's Logs at the UMM Heating Plant. During the performance testing 286,430 pounds of fuel were consumed, producing a total of 25,597 kWh of electricity. In our application, we use only a portion of the available energy in the steam for power production, reducing the steam pressure from boiler pressure down to 18 psi, the operating pressure of our campus heating system.

CDR made major progress. During this quarter, a series of experiments was conducted running the engine in a fumigation mode with lab-generated synthesis gas (syngas) as the primary fuel and diesel as the secondary fuel. These experiments were performed to investigate the feasibility of utilizing syngas in a modified diesel engine. The results were compared with the previous experiments where the engine was run in a fumigation mode with gasoline as the primary fuel at the same load and speed conditions. Fumigation is an early-stage form of the reactivity controlled compression ignition (RCCI) strategy. RCCI has been recently reported to achieve both very high efficiency and low emissions over a wide operational range using gasoline as the primary fuel. In the fumigation mode, a primary fuel of lower reactivity such as hydrogen, gasoline or ethanol is injected into the intake manifold to form a homogeneous fuel-air mixture, and a high reactivity fuel like diesel is injected directly into the cylinder to act as an ignition source. Previous investigations on fumigation were mostly done in the region where the pilot diesel fuel is injected close to top dead center (TDC). In this set of experiments, it was found that by advancing the injection timing of the diesel fuel, very low NO_x and soot emissions, as well as relatively high efficiency and stable operation, could be achieved.

In conclusion, stable and efficient operation of syngas in fumigation mode has been realized in our modified Isuzu diesel engine. The NO_x and soot emissions could be simultaneously reduced to very low levels by advancing the pilot diesel injection timing, while keeping relatively high efficiency. A promising prospect was shown for utilizing syngas in diesel engines with a small amount of diesel fuel. This is one of the first demonstrations of syngas operation in the RCCI mode that we know of. An instrumented injector was purchased to allow the measurement of the actual start of injection (SOI) of the diesel fuel, but it was not functional when received. The present data for SOI is based on the signal sent to the injector controller. The goal is to base future testing on the actual SOI.

HGA and UMM completed a small generator interconnect application for the 100kW generator. The application is complete except for the documentation that the generator is certified for continuous parallel operation. Cummins checked on whether this generator set meets this requirement.

During this quarter the electrical installation was reported to HGA as complete and in conformance with HGA's specification and the unit was sited according to its drawings. The project partners completed all Stage 3 work and were ready to begin Stage 4.

Q1, 2012

The PowerTainer went through preliminary testing and some extensive modifications to the fuel handling system. These modifications produced a much more efficient fuel delivery to the gasifier. Upgrades were also made to the electronic control system.

The LII was recalibrated at the factory and returned to CDR where its performance was confirmed.

The steam turbine ran under normal operating conditions. There was a scheduled maintenance shutdown in May. Following the shutdown, we began data collection for the head-to-head comparison between the steam turbine and the PowerTainer.

CDR completed all of its work as budgeted in Stage 3. Results were reported to all partners. Major accomplishments included achieving successful Reaction Controlled Compression Ignition (RCCI). Operating the engine in RCCI mode on a mixture of syngas and diesel fuel produced very low NOx and soot emissions while running at high efficiency.

HGA and UMM personnel continued to work with Otter Tail Power on the interconnection agreement for the PowerTainer. It was expected that the interconnect agreement would be signed in April.

Q2, 2012

The PowerTainer, in continued operational testing, developed a serious ignition problem while running on producer gas. Modifications were made to the air intake system and the control system. Backfire safety devices were installed. The engine now runs smoothly and reliably. A round of testing was performed in late June with APL, UMM, Cummins, and CDR personnel on site at UMM. That testing culminated in a public demonstration day on 28 June. The event was attended by over 40 people representing academe, business, government, farmers, entrepreneurs, and a few who were just curious. The PowerTainer ran flawlessly, and the demonstration day was deemed a great success.

Task 6 was completed except for emissions measurements, which were undertaken when the PowerTainer was running under load in Q3.

In continued testing, ignition problems arose in the PowerTainer diesel engine. Although having completed its work as budgeted in Stage 3, CDR continued to be an active partner, including consulting with APL on solutions to the ignition problems. Cummins also provided assistance. These ignition problems were unexpected and posed a serious threat to the timely completion of the project deliverables. The project partners worked very hard to solve this problem so that the work could proceed.

The LII, which was recalibrated at the factory in Q1, developed a data communications issue. CDR personnel, working in communication with the factory, were able to solve the problem via a software patch.

HGA and UMM personnel continued to work with Otter Tail Power Company on the interconnect agreement for connecting the PowerTainer to the grid. The interconnect agreement was completed early in Q2.

Q3, 2012

The PowerTainer continued in testing. Modifications were made to the fuel handling system inside the container, with special attention paid to reducing bridging of the fuel. The engine continued to run smoothly on producer gas with relatively small proportions of diesel fuel. We were interrupted in September by a problem with the grate in the gasifier. A new grate was being made at APL as the Quarter ended.

In attempts to connect to the grid, it was determined that the generator was not equipped with control equipment for parallel connection. The project team discussed several options at length, finally deciding that the least expensive and most expeditious solution was to use a load bank. A suitable load bank was purchased and put into service.

We have achieved power production of 97 kW on a blend of producer gas and diesel fuel. Our earlier concerns that we might lose up to 20 kW generating capacity when using producer gas were unfounded.

Testing at UMM in September, achieved 70% substitution at 50 kW. It was seen that stable operation occurs down to the idle diesel consumption of 1.9 gal/hr. Higher fuel substitution can occur at higher loads

The gas analysis and soot test equipment were connected to the PowerTainer and were fully operational.

Q4, 2012

The PowerTainer was operated under load with full instrumentation with APL personnel on site in Morris for several weeks. These fully instrumented runs of the blended producer gas/diesel fuel mixtures have provided performance data that are outside of the expected outcomes. We had expected to see a derating of the power produced from the blended fuel operations. While we had achieved engine operation with 75% producer gas under no load, we had not yet operated under load with high substitution of producer gas for diesel. In this quarter we demonstrated that the genset will operate for extended periods (repeatedly) under load (75 kW) on a mixture of 80% producer gas and 20% diesel. Under these conditions, the measured output of the genset was 74-75 kW. The results demonstrate that there is no derating of the genset capacity when operating at high substitutions.

Analyses of the producer gas and the engine exhaust gave us some surprises. For example, when running under load we occasionally saw a sudden spike in hydrogen concentration in the producer gas to more than twice the normal levels. A spike would produce a symmetrical peak over a period of 5-6 minutes. These spikes were infrequent and occurred with no discernible pattern. As of the end of December, we had not formed a working hypothesis to explain this behavior. These anomalies are intriguing, but the observation most piquing our interest is the excellent performance of the gasifier and genset working together.

We had planned to do the fully instrumented runs and then switch to production runs to complete the grant timeline. With the better than expected performance of the unit, we wanted to go back to the fully instrumented runs to try to understand what was causing the difference from expected performance. To allow this to happen, we asked for a three month no cost extension of the grant to allow for the investigative research on the thermal conversion chemistry, with the production runs after completion of the data analysis. The extended time to allow work on both the thermal chemistry and the operational characteristics would provide the best outcomes for the NETL grant deliverables. We were notified in late December that our request had been approved, but that final processing would not occur until after the new year.

Q1, 2013

The request for a no-cost extension was granted. In early January, the PowerTainer was shipped to APL in California to allow this testing to proceed. The weather in Minnesota had turned quite frigid, causing operational problems with our instruments. Also, the cold and the blowing snow made it very difficult for personnel to work outdoors. Testing continued once the unit was back in Berkeley. Data analysis moved into full swing.

4) Product(s) Developed and Publications

Commercial Potential

Steam turbine operation

Electrical production data were gathered for the steam turbine for 2144 hours over the period from January 2011 through March 2013. At times, the turbine was not running because of insufficient steam demand. The turbine requires at least 5,000 lb/hr steam flow to be put into service. Periods during which there was no electrical production were discarded from the dataset. Also excluded were periods of intermittent production caused by hourly variation in demand. Ultimately, a total of 504 hr of production time were used for analyses.

To explain the results, the following example calculation is presented. The average electrical output was 70.3 kW, and the average hourly fuel consumption was 1148 lb to produce steam. Thus, each pound of fuel produced $70.3 \text{ kW} / 1148 \text{ lb} = 0.062 \text{ kWh/lb}$. The average HHV of the corn cobs is 7582 Btu/lb, or 2.221 kWh/lb. The conversion efficiency of the corn cobs to steam is about 80% for the gasification and 88% for the boiler. So the 7582 Btu/lb in the cob becomes $7582 * 0.80 * 0.88 = 5338 \text{ Btu/lb}$ into steam. One pound of saturated steam at 150 psi contains 1196 Btu. Thus, each pound of cobs could produce $5338 / 1196 = 4.46 \text{ lb}$ of steam at 150 psi. The steam pressure is reduced by the turbine to 18 psi, the operating pressure of the campus heating system. Since saturated steam at 18 psi contains 1165 Btu, from each pound of steam, 31 Btu are available for generation of electricity. Thus $4.46 \text{ lb steam} * 31 \text{ Btu/lb} = 138.3 \text{ Btu}$ potentially converted to electricity. That is, of the 7582 Btu in one pound corn cobs, 138.3 Btu are available to be converted to electrical energy. That is 0.0405 kWh , and $0.0405 / 2.221 = 1.82\%$. At the price currently being paid by UMM for corn cobs, \$85.00/ton, 1.82% of that cost goes to produce 40.5 kWh/ton at 50% turbine efficiency. So, $85 * 0.0182 = \$1.547$, and $\$1.547 / 40.5 = \$0.038 / \text{kWh}$.

Financial model for assessing viability of the CHP and the PowerTainer

In this section we compare the three platforms in terms of their O&M costs per unit of power, as well as in terms of the *levelized cost of energy* (LCOE) produced over their projected lifetimes.

System 1 is the system currently in place on the campus of the University of Minnesota, Morris. It consists of the updraft inclined-grate close-coupled atmospheric gasifier, a high pressure steam unit from English Boiler, and a Skinner back pressure turbine rated at 315 kW @ 15k lb/hr. Exhaust steam exits the turbine at low-pressure, sufficient to supply the campus district heating system. System 2 is the PowerTainer, an experimental, shipping container-based syngas genset, well characterized elsewhere in this grant report. System 3 is a mature technology, widely available commercially, which is likely to be the chief competition for the PowerTainer in many applications.

The capital costs and contingent labor and maintenance costs associated with the UMM biomass gasification and high pressure steam system, not including the back pressure cogeneration unit, were reported and analyzed in a previous study (see endnote 1). Tables 4.1 through 4.3 below report these figures.

Clearly the capital costs of this system are quite high, for reasons discussed in that study. These figures are not directly relevant to the question examined here, however. The current task is to assess the incremental cost of generating electricity from an existing biomass gasification- steam system.

The logical approach for estimating the contingent cost of power generation is to consider the heating load as a hard constraint – i.e. as an obligatory target. The marginal O&M cost of the electricity, then, is the extra cost associated with producing high-pressure steam (capable of spinning the turbine), over and above the cost of generating the low-pressure steam necessary to meet the campus thermal load. Since capital costs are a contingent cost from the perspective of evaluating the new investment, an appropriate per-kwh capital cost must also be calculated and added in.

The *levelized cost of energy* (LCOE) is a standard methodology for calculating the value of a generation asset. Effectively, it calculates the price that a generator's output must be paid, in today's dollars, over the lifetime of the asset, in order for the project to have a positive net present value. For projects like this one, where capital costs are high and operations costs are low, the LCOE is highly sensitive to the level of output. For low capital-cost, high operations-costs projects (such as System 2, and especially System 3), the co-variance of output levels and input costs reduces this sensitivity. Because of this feature, operational assumptions about the frequency and intensity of the asset's use will affect the estimated LCOE.

Even beyond this, the complicating factor for determining the LCOE of this cogeneration technology is that the technical efficiency of the power plant varies with the flow of steam. System use is generally not binary. Steam production (by assumption here) will follow the thermal load, and when that load is below the full 10K lb/hr that the system is capable of, the ratio of Btu-in-to-kw-out will be below the optimal level (though still economically compelling, over a wide range of loads). We will therefore need to conduct scenario analyses below, in order to calculate LCOE under differing operational assumptions.

2003 Cost Estimate			\$5,100,000
2007 Approved Budget			\$8,900,000
	Building	\$2,030,000	
	Mechanical & Electrical	\$3,700,000	
	Gasification Equipment	\$1,270,000	
	Nonconstruction costs	\$1,900,000	
Costs through 2011 - estimated			\$10,400,000
	Budget	\$8,900,000	
	Additional Construction & Start up	\$400,000	
	Steam Turbine and Installation	\$1,100,000	

Table 4.1: Capital Costs for Biomass CHP Plant

Source: "Biomass Gasification: A Comprehensive Demonstration of a Community-Scale Biomass Energy System" Final Report to the USDA Rural Development, Grant 68-3A75-5-232

Grounds Crew			\$1,500
	Hours per Week	50	
	Salary per Hour	\$30	
Mgmt. and Acctg			\$450
	Hours per Week	15	
	Salary per Hour	\$30	
Mechanical & Boiler Staff			^600
	Hours per Week	20	
	Salary per Hour	\$30	
		Total per Week	\$2,550
		Total per Year	\$132,600

Table 4.2: Labor Costs for Biomass Operations

Source: "Biomass Gasification: A Comprehensive Demonstration of a Community-Scale Biomass Energy System" Final Report to the USDA Rural Development, Grant 68-3A75-5-232

Predicted Annualized Auxiliary Eqpt Costs			\$26,000
	Equipment Capital	18,000	
	Eqpt. Operational	8,000	
Predicted Additional Maintenance			\$27,000
	Labor	15,000	
	Supplies	12,000	
Operating Supplies			\$20,000
	NaOH	15,000	
	Water	5,000	
ANNUAL TOTAL			\$73,000

Table 4.3: Operations and Maintenance Cost for Biomass Plant

Source: "Biomass Gasification: A Comprehensive Demonstration of a Community-Scale Biomass Energy System" Final Report to the USDA Rural Development, Grant 68-3A75-5-232

We use the following assumptions to obtain an estimate of the marginal cost of power:

Assumptions		
1)	Steam flow	<i>follows the thermal load</i>
2)	Btu content/lb fuel (corn cobs)	7582
3)	Conversion efficiency of fuel (extraction/content)	0.8
4)	Combustion efficiency of fuel	0.865
5)	Btu/lb saturated steam @ 280 psi	1202.3
6)	Btu/lb saturated steam @ 18 psi	1166
7)	(5)-(6) = marginal Btu/lb high-pressure steam	36.3
8)	Cost/ton biomass (delivered)	85
9)	Feed water temp. (°F)	180
10)	Btu/lb feed water	148
11)	kWh=f(lb/hr steam)	$(-86)+.028*(\text{lb/hr steam})$ [5k lb/h MINIMUM]

Table 4.4: Key Parametric Assumptions for System 1

Note that assumption (11) is based on personal correspondence with engineers from the Skinner Corporation. Below 5k lb/hr steam throughput, the turbine produces no power.

The UMM physical plant has achieved maximum steam output of approximately 10k lb/hr, with the limiting factor being the flow of syngas from the gasifier unit. The experience to date suggests that a larger gasifier may be necessary to meet the full 15k lb/hr that the boiler is capable of. For the time being, the boiler could potentially be operated up to 15K lb/hr by cofiring with syngas and natural gas. In what follows, we model operation at 10k lb/hr on syngas alone.

Based on these assumptions, the marginal *fuel* cost of power (from the biomass-based system alone) is a function of the thermal load, as follows:

lb steam/hr	5000	6000	8000	10000
kWh	0	82	138	194
marginal fuel cost	N/A	\$0.022	\$0.017	\$0.015

Table 4.5: Marginal Fuel Costs of Power, as a Function of Steam Through-put

When the thermal load is below 3072 lb/hr, the full Btu charge of the high-pressure steam (as well as incremental labor and maintenance charges, not yet included) must be attributed to the power, and the economics quickly become prohibitive. The third row figures may be compared with a reference retail price of approximately \$0.08/kWh in Minnesota, and \$0.125/kWh nationally. When thermal load is present, the incremental fuel cost of the power is very low.

The actual thermal load at UMM varies significantly over the course of the year, with heavy winter (steam heat) and summer (absorption chiller AC) loads, bracketed by “shoulder seasons,” during which demand falls off to the much smaller domestic hot and cold water loads. It would be difficult and not particularly useful to focus on the historic time-variant operations of this specific physical plant. Instead, we consider two implementation scenarios. In the first, we model an application where the thermal load perpetually exceeds 10k lb/hr, so that the gasifier-boiler-back pressure turbine-steam system may be operated at capacity at all times (and the remaining thermal load followed by a low pressure backup boiler). In the second scenario, we model a system that operates 25% of the time at each of four output levels, as indicated in Table 4.6 below. Since the output efficiency of the turbine increases with steam volume (see the formula above), operating scenario 2 will not only produce significantly less power, but the O&M cost per kWh will also be higher. The key assumptions common to both scenarios are given in Table 4.7.

First, we present a back-of-the-envelope calculation which ignores discounting.

	Scenario 1	Scenario 2
Steam Flow (lb/hr)		
10k	100.00%	25.00%
8k	0.00%	25.00%
6k	0.00%	25.00%
<5k	0.00%	25.00%

Table 4.6: Scenario Analysis for Operating Levels of the Back Pressure Turbine

Assumptions	Scenario 1 & 2
Expected Lifetime (years)	40
Capital Cost of BP Turbine	\$620,220
Maintenance costs (annual)	\$2,000
Labor costs (annual)	\$0.00
Scrap value	\$0.00

Table 4.7: Common Scenario Assumptions

Scenario 1:

Operating continuously at 10k lb/hr steam volume (280 psi), the annual kWh amount to 1,700,410, at a marginal fuel cost of \$25,772.56 (over and above the cost of producing the low-pressure steam to meet the thermal load), or \$.0152/kWh. Annual labor, maintenance, and straight-line capital costs amount to \$17,505.50 or \$.0103/kWh. Thus the back-of-the-envelope cost/kWh is the sum of these: \$.0255/kWh.

Scenario 2:

Operating 25% of the time at each of the steam output volumes identified above, the annual kWh amount to 907,178 at a marginal fuel cost of \$.017/kWh. Annual labor and straight-line capital costs amount to \$15,463.54, or \$.0193/kWh. Together these total \$.0363/kWh.

	Annual kWh	Fuel cost/kWh	Non-fuel cost/kWh	Overall Cost/kWh
Scenario 1	1,700,410	\$0.0152	\$0.0103	\$0.0255
Scenario 2	907,178	\$0.0170	\$0.0193	\$0.0363

Table 4.8: Scenario Comparison, w/o Discounting

The Levelized Cost of Energy

The *levelized cost of energy* (LCOE) is a more accurate measure, which takes into account the time value of money.² It may be interpreted as the real price, in current dollars, that a generator must receive for its output, over the course of its entire lifetime, in order to break even. The formula is as follows:

$$LCOE = \frac{\frac{\sum_{t=1}^n I_t + M_t + F_t}{(1+r)^t}}{\frac{\sum_{t=1}^n E_t}{(1+r)^t}}$$

where

LCOE = Average lifetime levelized electricity generation cost

I_t = Investment expenditures in the year t

M_t = Operations and maintenance expenditures in year t

F_t = Fuel expenditures in year t

E_t = Electricity generation in year t

r = Discount rate

n = Life of the system

In order to apply this formula to the scenario analysis above, we must make two additional assumptions: 1) the real cost of fuel, as well as O&M, is constant through time (i.e. increasing at exactly the economy-wide inflation rate); 2) the electricity output of the system is constant over time (no decay in productivity). Applying these assumptions to the scenarios developed above, we arrive at the true LCOEs, reported in Table 4.9, which also allows the discount rate to vary.

Discount rate	Scenario 1 LCOE	Scenario 2 LCOE
0.00%	\$0.0255	\$0.0414
3.50%	\$0.0276	\$0.0518
7.00%	\$0.0363	\$0.0726

Table 4.9: Levelized Costs of Energy of the UMM Cogeneration Plant

System 2: The PowerTainer

The PowerTainer has been successfully tested at an output level of 74 kW, running on a blend of syngas and diesel. Table 4.10 reports the relevant inputs, outputs, and prices.

	Diesel	Biomass	Electricity
Hourly input	1.5 (gal)	73.19 (kg)	-
Price/unit	\$4.00	\$85.00	-
% of Btu inputs	22.90%	77.10%	-
Output (kW)	-	-	74
Input cost per unit output	\$0.0811	\$0.0927	-
Total cost per unit output		-	\$0.1737

Table 4.10: Fuel Inputs, Power Outputs, and Fuel Unit Costs of the PowerTainer

Availability	70.00%
Annual labor cost	\$75,000.00
Engine TBO (hr)	15000
Annual overhaul set-aside	\$2,045.17
Other annual maintenance	\$10,000.00
Capital Cost	\$200,000.00
Expected lifetime (yr)	20

Table 4.11: Design, Operating, and Cost Assumptions for the PowerTainer

On the basis of these parameters and assumptions, and using straight-line capital costs (no discounting), we arrive at the back-of-the-envelope calculation in the final row of Table 4.12.

Annual O&M Costs	\$165,930.93
Annualized Straight-Line Capital Cost	\$10,000.00
Annual kWh	454027
Cost per kWh	\$0.39

Table 4.12: PowerTainer Costs, Output, and Unit Output Cost

As before, this back-of-the-envelope calculation suffers from failing to consider the time value of money. In order to correct this, we apply the LCOE formula to the input/output analysis above, making two additional assumptions (as in the previous LCOE analysis): 1) the real cost of fuel, as well as other O&M, is constant through time (i.e. increasing at exactly the economy-wide inflation rate); 2) the power output of the system is constant over time (no decay in productivity). In this way we arrive at the LCOEs, reported in Table 4.13, which also allows the discount rate to vary.

Discount Rate	LCOE
0.00%	\$0.3875
3.50%	\$0.3965
7.00%	\$0.4070

Table 4.13: PowerTainer Levelized Cost of Energy

Note that because the ratio of capital costs to O&M costs is so much lower than in System 1, and (relatedly), the covariance of input costs and output revenues is so much higher, the LCOE for System 2 is relatively insensitive to changes in the discount rate.

System 3: The Diesel Genset

For the sake of comparison, we develop the above calculations for a similarly-sized diesel genset, which is likely to be the competing technology platform for the PowerTainer in many applications. For the sake of this set of calculations, we have used the parameters for the genset embedded in the PowerTainer, substituting pure diesel for the 77.1% of the Btu derived from syngas above.

	Diesel	Electricity
Hourly input (gal)	6.5502	-
Price/unit	\$4.00	-
% of Btu inputs	100.00%	-
Output (kW)	-	74
Total fuel cost per unit output (\$/kWh)	\$0.3541	

Table 4.14 Unit Costs of the Diesel Genset: Fuel Inputs, Power Outputs, and Fuel

On the basis of these parameters and assumptions, and using straight-line capital costs (no discounting), we arrive at the back-of-the-envelope calculation in the final row of Table 4.15.

Availability	90.00%
Annual labor cost	\$10,000.00
Engine TBO (hr)	15,000
Annual overhaul set-aside	\$2,629.50
Other annual maintenance	\$2,000.00
Capital Cost	\$50,000.00
Expected lifetime (yr)	25

Table 4.15: Design, Operating, and Cost Assumptions for the Diesel Genset

Annual O&M Costs	\$221,315.09
Annualized Straight-Line Capital Cost	\$2,000.00
Annual kWh	583,749
Cost per kWh	\$0.3826

Table 4.16: Diesel Genset Costs, Output, and Unit Output Cost

As before, this back-of-the-envelope calculation suffers from failing to consider the time value of money. In order to correct this, we apply the LCOE formula to the input/output analysis above, making two additional assumptions (as in the previous two LCOE analyses): 1) the real cost of fuel, as well as other O&M, is constant through time (i.e. increasing at exactly the economy-wide inflation rate); 2) the power output of the system is constant over time (no decay in productivity). In this way we arrive at the LCOEs, reported in Table 4.17, which also allows the discount rate to vary.

Discount factor	LCOE
0.00%	\$0.3826
3.50%	\$0.3843
7.00%	\$0.3865

Table 4.17: Diesel Genset Levelized Cost of Energy

In Table 4.18 we present in a unified format comparable LCOE data for the three technologies evaluated, at each of three discount rates.

	System 1: UMM Cogeneration Unit	System 1: UMM Cogeneration Unit	System 2: The Powertainer	System 3: Diesel Genset
Discount Rate	<i><u>Scenario 1</u></i>	<i><u>Scenario 2</u></i>		
0.00%	\$0.026	\$0.041	\$0.388	\$0.383
3.50%	\$0.028	\$0.052	\$0.397	\$0.384
7.00%	\$0.036	\$0.073	\$0.407	\$0.387

Table 4.18: LCOEs for Each System

End Notes

1 For a full description see: “Final Report to USDA Rural Development, Grant 68-3A75-5-232.”

2 Note that the previous estimate characterizes the special case where the discount rate is zero.

Commercial development Plan

Although only 5 years old, APL has developed a world-wide distribution and service network for its Power Pallet, a small gasifier/genset integrated onto a 4 ft x 4 ft pallet skid. It is available in 10, 15, and 20 kW versions. Sales have exceeded expectations and APL has expanded its production capabilities.

Market interest for the PowerTainer has been strong. Specifically, interest is coming from three distinct areas:

- 1) Rural, off grid remote settings, where access to stable energy is either unavailable or is predicated on the use of volatily priced carbon based fuel sources. Given the scale of energy the PowerTainer is capable of producing, there has also been a significant demonstrated interest in coupling energy production from a PowerTainer with the energy needs of telecom towers, so they could get both energy and communications. Such an arrangement would likely take place in the context of an Energy Service Company (ESCO) purchasing the PowerTainer to operate, then selling the energy to local users under some form of a Power Purchase Agreement.
- 2) Agricultural materials producers and handlers who have a “ready-made” source of fuel on hand. They could supplement their existing energy infrastructure with a PowerTainer, either on a micro-grid or as a grid-tied production platform. Given their expertise in materials handling and production, it would not be difficult for many to adapt their processes to ensure an appropriate source of biomass. Such users could include sawmills, lumber yards, and commercial agricultural producers of food.
- 3) Grid interconnection to markets with established Feed In Tariffs (FITs). In several European Union countries there is an extant, and robust, FIT market for the production of renewable energy. In those countries that have already established a qualification for biomass based energy production, there has been a consistent demand for a mid-size, “farm scale” platform such as the PowerTainer.

In order to have a market ready product, further refinement is required, in both operation and instrumentation. The PowerTainer is currently configured to be operated by a trained technician who specializes in biomass gasification. In order to have a product adaptable to a wider market, more automation and more rigorous operational guidance materials would need to be generated. Additionally, further testing would need to be performed on a variety of feed stocks and configurations, to ensure the widest possible understanding of what constitutes a viable source of fuel.

All Power Labs is currently planning on next generation model testing to explore a refined reactor, as well as various fuel handling scenarios. APL is also planning future units utilizing a spark-fire engine. Funding for further testing of optimized compression ignition modes is being pursued.

That said, there are no significant barriers to widespread adoption, and with further refinement the current platform should achieve substantive market penetration.

Future R&D needs/opportunities

Lessons learned and next R&D needs

1. Soot is low on pure diesel, goes up at high and mid ranges of power and low to mid substitution, typically decreasing as the substitution level increases, however at the 100 kWe load case, the soot output level remained very high perhaps due to other issues such as the intake line collapsing due to the high vacuum levels. The soot went down again at the lower power, 25 and 12.5 kWe, loads and high substitution levels at loads of 75 kWe and below.
2. CO and H₂ slip through the engine is low at low substitution, high in mid-range of power and substitution, and lower again at high power and high substitution.

The emerging theme seems to be the conflicting conditions needed for good diesel cycle (no air throttling, lean burn) combustion vs. good Otto (premixed, near stoichiometric) combustion. When primarily in one mode or the other, emissions are reasonable as that mode is operating properly. When in significant mix between the modes, the needs of each are fighting the other, and poor emissions follow. The dual fuel mostly makes emissions worse, but with potentially a recovery to better at high substitution amounts.

Diesel engines rely on significant excess oxygen to have a diffusion flame combustion regime happen tolerably in a very short period of time. Diesels will go into soot runaway with less than around 30% free oxygen. If you start to dilute the air available with other inert gasses, which is the case with mixing in syngas, you are reducing the available oxygen at the diffusion flame front at injection. The diesel combustion event starts to fall apart.

Similarly, the syngas air portion needs to be reasonably mixed to get good combustion of the syngas. There are flammability limits within which CO and H₂ will burn, particularly CO. If you have too much air, as when at low load, you are more likely to fall below the flammability limits of CO and it will slip through the engine. The excess free oxygen is actually a problem for the syngas portion of the dual fuel mixture, whereas the excess oxygen is critical to the diesel working correctly.

Thus the needs of the diesel cycle and the Otto cycle are most in conflict in the mid ranges of the dual fuel mixtures. We get the worst soot performance and CO and H₂ slip in the mid ranges of power and substitution. At the higher end of power and substitution, things return a bit to order, as we are moving primarily into an Otto mode, and there is very little diesel to soot (even though the diesel coming in may produce high soot levels - there just isn't that much of it). At the same time, CO and H₂ slip go down as the syngas air mixture is near enough to stoichiometric (tolerable lean), that we stay within the flammability limits of the gases and thus they convert.

Similarly, at the bottom end of power or substitution we get good emissions as we are primarily in diesel mode. Soot is reasonable due to desired full air, and there is little CO and H₂ anyway to pass through the combustion chamber unburned. To the degree there is any, you want to be at

higher power on the diesel so the diesel combustion event propagates most widely and thus gets to where the CO and H₂ reside. At low power, most of the diesel combustion is in the piston combustion bowl and goes to completion before much gets out to the edges and skirt where CO and H₂ are already mixed.

Thus to the degree you are dual fuelling in an un-throttled regime, it seems you want to be at high load so the diesel combustion event is most likely to envelope the entire chamber, as it needs oxygen from the entire chamber.

In summary, dual fuelling diesels with syngas would greatly benefit from a hyper low injection amount, as the Reactivity Controlled Compression Ignition (RCCI) work is attempting, however even with this we'll still get lots of slip of CO and H₂ at low loads, as the excess oxygen is still there and challenging the flammability limits of the far off stoichiometric syngas/air mixture. Modifying the engine geometry such as operating with a different piston design may also limit the amount of CO and H₂ slip through the engine.

This could be fixed with a stratified charge regime, added to the partial compression ignition regime currently explored at the Center for Diesel Research lab.

You could just convert the diesel to an Otto cycle with spark plugs in the injector holes and add a throttle. This would allow you to maintain the large benefits of compression ratio increase possible with syngas, and to run the engine to varying excess air amounts to minimize throttling losses at partial load.

Other R&D opportunities

Further work on emissions reduction:

- Modified injection timing control should allow improved emissions and a decrease in diesel fuel rate
- Conversion to a low temperature combustion mode should allow lowered emissions
- Diesel engine cylinder geometry could be modified to provide cleaner combustion
- Testing of performance on other fuels
- Characterization of thermal loads through the system
- Continued automation to minimize operator attention required

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Homogeneous charge compression ignition engine operating on synthesis gas

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ABSTRACT

Mixtures of hydrogen and carbon monoxide were used to simulate the fuel component of synthesis gas and operate a single cylinder engine in homogeneous charge compression ignition (HCCI) mode. The engine was originally an air-cooled direct injection (DI) compression ignition (CI) engine. The original diesel fuel injection system was removed and a port fuel injection (PFI) system with intake air heating was added. The engine speed was maintained at a constant 1800 RPM.

Three synthesis gas fuel compositions were tested, which comprised of 100% H₂, 75/25 H₂/CO ratio, and 50/50 H₂/CO ratio, by volume. These compositions were investigated at an equivalence ratio (EQR) of 0.26 and 0.30. In-cylinder pressure and H₂/CO emissions measurements were made at all conditions.

To achieve peak indicated mean effective pressure (IMEP) at a given equivalence ratio, the intake air temperature had to be increased with increasing CO fraction in the synthesis gas mixture. For the EQR = 0.26 conditions, the intake air temperature required for best IMEP was 78 °C, 84 °C, and 98 °C, for 100% H₂, 75/25 H₂/CO ratio, and 50/50 H₂/CO ratio, respectively. For the EQR = 0.30 conditions, the intake air temperature requirements were 62 °C, 71 °C, and 81 °C, for the same respective H₂/CO proportions. The peak in-cylinder temperatures for all conditions tested ranged from roughly 1200 K–1500 K depending on intake air temperature, mixture concentration, and fuel composition. The combustion event was short, with the rapid burn angle ranging from 9.5 CAD to 11.5 CAD for all conditions tested, and the synthesis gas mixture composition did not change this significantly. The combustion efficiency was between 83% and 88% for the peak IMEP conditions tested.

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1. Introduction

A concept called homogeneous charge compression ignition (HCCI) is an internal combustion engine operating regime that is being investigated by many researchers, because of its very low NO_x emissions characteristics. Like an SI engine, an HCCI engine typically relies on external mixture formation (PFI system) to produce a homogeneous mixture which is inducted

into the engine cylinder during the intake stroke. This mixture is then compressed, and ignition (and subsequent burning) occurs solely due to the increase in temperature and pressure during the compression stroke. An HCCI engine is similar to a CI engine, because it is usually operated at wide open throttle (typically boosted) and is compression ignited.

HCCI engines typically exhibit diesel-like efficiency without suffering from high engine out NO_x or PM emissions,

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and are operated globally fuel-lean, which facilitates a low temperature combustion (LTC) environment. LTC means that the NO_x forming high temperature combustion regions are avoided. The lean mixture is also homogeneously distributed which means that there are no locally rich pockets of fuel and air, essentially eliminating soot formation.

There are issues with operating an engine in the HCCI regime. There is no direct control of ignition timing, such as a spark in SI engines or the fuel injection event in CI engines. One method of controlling the ignition timing is by increasing or decreasing the intake air temperature. This, however, makes transient control cumbersome. There are also difficulties with controlling in-cylinder pressure rise rates during high load engine operation. To achieve higher load operation, large amounts of exhaust gas recirculation (EGR) are required. These approaches, and the HCCI operating regime in general, are not thought to be production viable yet, and a significant amount of research is being devoted on choosing the ideal HCCI fuel and control scheme.

Onishi et al. performed what are now viewed as the first HCCI experiments and named it active thermo-atmosphere combustion (ATAC) [1]. Compared to SI engine operation, they reported a substantial decrease in NO_x emissions along with reduced combustion noise. They also noticed that there was no discernable flame propagation, but instead combustion occurred spontaneously at multiple points. In the same year as Onishi et al., Noguchi et al. investigated HCCI, but named it Toyota-Soken (TS) combustion [2]. They reported crank angle dependent radical formation, which was different from conventional SI engine operation.

Najt and Foster investigated four-stroke HCCI operation in a Cooperative Fuels Research (CFR) Octane engine and gave it the name compression ignited homogeneous charge (CIHC) [3]. They used primary reference fuels and focused on evaluating the influence of varying compression ratio, intake air heating, and EGR on HCCI combustion characteristics. They found that HCCI combustion could be described by the global hydrocarbon kinetics, where the low temperature kinetics controlled the ignition of the fuel, while the high temperature kinetics controlled the heat release rate and combustion duration.

In more recent years, several researchers have investigated HCCI engine operation with hydrogen. Most researchers have focused on using small amounts of hydrogen or hydrogen rich gases [4–6], while few have investigated HCCI engine operation using pure hydrogen [7–9]. Caton et al. investigated pure hydrogen HCCI combustion in a single cylinder indirect injection (IDI) diesel engine [9]. They tested engine operation at various compression ratios, air/fuel ratios, and intake air temperatures. They reported significant unburned hydrogen in the exhaust (0.5–1%) and an engine efficiency that was lower than when operating in CI mode. The lower engine efficiency was attributed to higher heat transfer to the cylinder walls and lower combustion efficiency. Gomes Antunes et al. investigated hydrogen HCCI combustion in an engine with a compression ratio of 17:1 [8]. The ignition timing was adjusted by heating the intake air. It was found that the engine could be operated at an excess air ratio (λ) from 3 to 6, which was limited by knock and misfire, respectively. The maximum brake thermal efficiency was reported as 45%. Stenlaas et al. investigated the efficiency, combustion phasing and emissions

from a hydrogen HCCI engine [7]. Similar to Gomes Antunes et al., they also found that the operating regime could be varied from $\lambda = 3$ to 6. They found that as the intake temperature was increased, the start of combustion advanced, and for the loads tested, HCCI engine operation exceeded the thermal efficiency of SI engine operation. Despite this work using hydrogen in HCCI engines, little work has been published on the utilization of synthesis gas in the HCCI regime.

Synthesis gas, also known as syngas, producer gas, or reformer gas, is a mixture of gases containing primarily hydrogen, carbon monoxide, carbon dioxide, and nitrogen along with other trace gases. Syngas can be produced from a variety of feed stocks ranging from fossil coal to renewable biomass. This allows for syngas production to be tailored to a localized resource. The problem with having a variable feedstock and variable production methods is that the syngas composition can vary significantly.

There have been investigations of synthesis gas supplemented diesel and spark ignition engines and even synthesis gas HCCI applications [4–6,11–13]. However, there is only one known work regarding operating an HCCI engine on straight synthesis gas, which only briefly investigated a single blend of synthesis gas [10].

Stenlaas et al. investigated reformed methanol gas (RMG) as a straight HCCI engine fuel, which has a 67/33 H_2/CO ratio if thermally reformed. [10]. They compared their findings against an HCCI engine operating on straight hydrogen, as well as a spark ignition engine operating on RMG. The main objective the Stenlaas et al. paper was to determine if the RMG HCCI engine concept was possible at a system level, while also investigating the combustion characteristics and emissions. They found that there was not sufficient energy in the exhaust gas to drive the catalytic reformation of methanol while operating in HCCI mode with either RMG or H_2 . They were able to extend the load range beyond H_2 operation alone, however the range was still limited to extremely lean operation from $\lambda = 3$ to 6. They found that syngas HCCI operation produced a rapid heat release rate and the start of combustion (SOC) was difficult to control. Considerable emissions of CO, along with some emissions of hydrocarbons were reported, with the latter thought to have originated from the lubricating oil.

The work presented in this paper differs from prior work in the literature; because this work was based on experiments conducted using multiple compositions of syngas in HCCI mode, whereas the only other known work was based on a single composition of syngas [10]. The focus of this work was on investigating the combustion characteristics and efficiency of an engine operating on varying blends of synthesis gas in HCCI mode. Key emphasis was placed on determining the effect that varying intake temperature and fuel composition has on the combustion characteristics and engine efficiency.

2. Experimental setup

The experiments in this study were performed on an overhead valve (OHV) Yanmar L100V single cylinder engine. The engine was originally an air-cooled, naturally aspirated, four-stroke direct injected (DI) diesel engine. The engine was modified by removing the diesel fuel injection pump and converted to

operate in port fuel injected HCCI mode. The intake manifold was modified with two gaseous fuel injectors and a 1 kW resistive intake air heater. The engine was coupled to an induction motor/generator, which maintained the engine speed at a constant 1800 ± 25 RPM during both motoring and firing conditions.

The original piston was a re-entrant style piston with a lip at the top of the bowl to increase toroidal turbulence, which is designed to enhance diesel-air mixing and burning. For these HCCI tests, however, the original bowl geometry and large squish area were deemed unnecessary. A more conventional HCCI piston style with a shallow dish shape was designed and fabricated, while maintaining the original clearance volume. The engine specifications are shown in Table 1.

The fuel injection system consisted of a Siemens 3RG-4021 inductive sensor which was mounted to the engine valve cover to sense intake valve motion. This sensor was used as a trigger input for a two channel BNC 555 pulse-delay generator which sent a signal to the two Quantum PQ2-3200 hydrogen fuel injectors, via National Semiconductor LM1949 injector drivers. The H_2 and CO were injected only when the intake valve was open to prevent any buildup of the gases within the intake manifold. Fig. 1 shows a schematic of the experimental setup.

The engine air flow rate was measured using a Merriam 50MW20-2 laminar flow element. The H_2 and CO fuel flows were measured independently using Alicat m-series mass flow meters. The intake air temperature was controlled using a PID temperature controller, which maintained the temperature to $\pm 0.3^\circ\text{C}$.

In-cylinder pressure was measured using a Kistler 6123 pressure transducer coupled to a Kistler 504E charge amplifier. A National Instruments 6062E data acquisition card was used with Labview 8.6.1 to acquire the high sampling rate in-cylinder pressure data. All parameters derived from the in-cylinder pressure data reported in this work, was based on a 100 cycle average. A US Digital HB6M rotary optical encoder with 1800 CPR resolution was used as an external clock for in-cylinder pressure data acquisition timing.

An Atmosphere Recovery Inc. laser gas analyzer was used to measure the hydrogen and carbon monoxide concentration in the exhaust, which were used for the combustion efficiency calculations (shown in a subsequent section).

3. Testing methodology

Three syngas fuel compositions were tested, comprising of 100/0, 75/25, and 50/50 H_2 to CO ratios (H_2 /CO by vol.%). These

1	Encoder	6	Fuel Flow Meters
2	Intake Valve Sensor	7	Air Flow Meter
3	In-cylinder Pressure	8	Intake Surge Tank
4	Intake Heater	9	Motor/Generator
5	Fuel Injectors		

Fig. 1 – Experimental setup.

compositions were investigated at equivalence ratios (EQR) of 0.26 and 0.30. These equivalence ratios were selected because they corresponded to a flow rate of 30 slpm and 35 slpm for the 100% H_2 conditions. Lower EQR testing was not possible, because of the limitations on the intake air heating system on the test rig for the CO blended conditions. Higher EQR testing was not possible, because the in-cylinder pressure approached the engine design limits.

The engine was first motored for 30 min to bring the oil temperature and intake temperature above 60°C and 90°C , respectively. Once these temperatures were reached, the hydrogen flow to the engine was increased to 30 slpm and the engine slowly moved from motoring to firing. The engine was operated at this condition until the oil temperature stabilized, at which point data were collected for the particular test conditions.

A sweep of intake temperatures using 2°C intervals was made at each fueling condition to determine the peak IMEP. It was determined that a sweep from high temperature to low intake temperature or low temperature to high temperature produced a hysteresis effect, due to slow cooling or heating of the cylinder walls. To minimize this effect, the test matrix was randomized and between each intake temperature test condition, the engine intake temperature was brought to 2°C higher than the highest intake temperature at the particular fueling amount, for 2 min. This was done because it produced a consistent starting condition for each test point. The same procedure was repeated for all of the test conditions.

Table 1 – Engine specifications.

Piston geometry	Dished piston
Bore (mm)	86
Stroke (mm)	75
Compression Ratio	21.2
Engine Speed (RPM)	1800 ± 25
Swept Volume (L)	0.435
Connecting Rod/Crank Radius	3.22

4. In-cylinder data analysis methods

The in-cylinder analysis was based on the precise knowledge of the in-cylinder pressure at known cylinder volumes throughout the engine cycle. The key equations are shown in the following sections; however a detailed derivation of the heat release rate analysis is given in [Appendix A](#).

4.1. Heat release rate and in-cylinder temperature analysis

The heat release rate analysis is based on the first law of thermodynamics shown in Equation (1).

$$\delta Q_{HR} = dU - \delta W - \delta Q_{HT} \quad (1)$$

In Equation (1), δQ_{HR} is the heat released by combustion, δQ_{HT} is the heat transferred to the walls, δW is the work done on the piston, and dU is the change in internal energy. Substitutions must be made into Equation (1) to get dU and δW in terms of parameters that can be measured, such as in-cylinder pressure (P), volume (V), and crankshaft position (θ). If the heat transfer to the walls is not known, and not estimated, then the net heat release rate can be calculated by combining the two heat transfer terms as shown in Equation (2).

$$\left(\frac{dQ}{d\theta}\right)_{net} = \left(\frac{dQ}{d\theta}\right)_{HR} - \left(\frac{dQ}{d\theta}\right)_{HT} = \frac{1}{\gamma-1} V(\theta) \frac{dP}{d\theta} + \frac{\gamma}{\gamma-1} P(\theta) \frac{dV}{d\theta} \quad (2)$$

In this work all rates of heat release are reported based on the net value, neglecting heat transfer to the walls, and the specific heat ratio (γ) was assumed to be 1.3.

The in-cylinder temperature was calculated based on the ideal gas law, assuming a closed system after intake valve closure (IVC). This requires the following: 1) known charge mass (after IVC), 2) known volume change as a function of crank angle, 3) known in-cylinder pressure as a function of crank angle, and 4) the assumption of no heat losses or exhaust residuals. From the measured quantities, a reasonable approximation of the cylinder temperature can be made based on Equation (3).

$$T_{cylinder} = \frac{P_{cylinder} \times V_{cylinder}}{m_{charge} \times R_{charge}} \quad (3)$$

4.2. Efficiency calculations

The combustion efficiency calculation was based on quantifying the fuel energy entering the system and the fuel energy leaving the system. This is shown in Equation (4).

$$\eta_{combustion} = 1 - \frac{\dot{m}_{fuel,exhaust} \times LHV_{fuel}}{\dot{m}_{fuel,intake} \times LHV_{fuel}} \quad (4)$$

For this study the mass flow rate of the fuel in the intake and exhaust consisted of H_2 and CO. The mass flow rates of the H_2 and CO in the exhaust were calculated based on fuel and air mass flow and exhaust composition measurements. The cycle efficiency calculation, based on the net indicated work, is shown in Equation (5).

$$\eta_{cycle} = \frac{\text{Indicated Work}}{\dot{m}_{fuel} \times LHV_{fuel}} \quad (5)$$

An analysis of the heat transfer to the cylinder walls, although neglected for the heat release rate calculations, was conducted based on the enthalpy/energy entering and leaving the system, shown in [Fig. 2](#).

The transfer of all enthalpies and energies, except for the leakage past the piston rings/valves and the heat transfer to the cylinder walls were directly measured since the engine was operating on a fuel of known composition (H_2 and CO). For the analysis an assumption of 2% leakage past the rings was made. This assumption is higher than would be expected, however a reduced leakage assumption can be used with the difference being added directly to the heat transfer value.

5. Results and discussion

The results and discussion section of this paper is divided into three sub-sections: 1) in-cylinder pressure and temperature, 2) combustion characteristics, and 3) engine efficiencies. The first section presents data on how the in-cylinder pressure and temperature plots vary with intake air temperature changes. The second section shows how the combustion characteristics are affected with varying H_2/CO ratio. The final section shows the variation of the combustion and cycle efficiency and describes how the energy leaves the system (heat transfer to walls, work output, exhaust, etc). For the last two sections, results are shown for the best IMEP conditions.

5.1. In-cylinder pressure and In-cylinder temperature

[Fig. 3](#) shows the in-cylinder pressure and heat release rate (HRR) plots for the engine operating with 100% H_2 and 50/50

Fig. 2 – Schematic of energy flow from combustion chamber.

Fig. 3 – In-cylinder pressure traces and heat release rates of the engine operating at varying intake air temperatures for: (a) EQR = 0.26 with 100% H₂, (b) EQR = 0.3 with 100% H₂, (c) EQR = 0.26 with 50/50 H₂/CO ratio, and (d) EQR = 0.30 with 50/50 H₂/CO ratio. All conditions show an advancing SOC with increasing intake air temperature.

H₂/CO ratio for an EQR of 0.26 and 0.30. The IMEP for all conditions tested ranged from 1.75 bar to 2.75 bar. The data was taken at 2 °C increments, however for clarity and conciseness only select intake temperature conditions are shown. Testing was also conducted with a 75/25 H₂/CO ratio at an EQR of 0.26 and 0.30. Data from those conditions are not shown, because they reveal similar trends to the 100% H₂ and 50/50 H₂/CO ratio conditions.

For all of the conditions tested, the plots in Fig. 3 show increasing in-cylinder peak pressure and increasing peak HRR with increasing intake air temperature. This trend was similar to what was shown by Stenlaas et al. [10]. For the 100% H₂ conditions, the intake air temperatures were varied from 72 °C to 84 °C for an EQR of 0.26 and 60 °C–72 °C for an EQR of 0.30. For the 50/50 H₂/CO ratio conditions, the intake air temperatures were varied from 92 °C to 104 °C for an EQR of 0.26 and 78 °C–90 °C for an EQR of 0.30. This corresponded to an operating range from high temperature to low temperature of 12 °C for all conditions tested. The high intake air temperature limit was quantified by the engine exhibiting in-cylinder pressure rise rates above 6–8 bar/CAD. The low intake air temperature limit was quantified by misfire, or the engine exhibiting a coefficient of variation of IMEP (COV_{IMEP}) exceeding 10%.

The 50/50 H₂/CO ratio conditions required a roughly 20 °C higher intake air temperature to achieve stable engine operation, compared to the 100% H₂ conditions. The 75/25 H₂/CO ratio conditions required a roughly 10 °C increase in intake air

temperature to achieve stable engine operation, compared to the 100% H₂ conditions. This shows that the autoignition temperature of the mixture increases with increasing CO fraction.

The HRR plots of Fig. 3 show that increasing the intake temperature leads to earlier combustion and higher HRRs. High intake temperatures lead to early combustion and high, ultimately limiting, peak pressures while low temperatures lead to late combustion lasting well into the expansion stroke and consequent low efficiency.

Fig. 4 shows the peak in-cylinder pressure for the best IMEP conditions of the engine operating with varying H₂/CO ratios. A trend of increasing peak cylinder pressure with increasing CO fraction is seen for an EQR = 0.30. The peak cylinder pressure increases from roughly 53 bar–60 bar, for the 100% H₂ and 50/50 H₂/CO ratio conditions, respectively. The peak cylinder pressure remained relatively constant at 59 bar for all conditions tested at an EQR of 0.26.

Fig. 5 shows the in-cylinder temperature plots for the engine operating with 100% H₂ and 50/50 H₂/CO for an equivalence ratio of 0.26 and 0.30, similar to the in-cylinder pressure plots of Fig. 3. For all conditions tested, the peak in-cylinder temperature increased with increasing intake temperature. This was likely due to the higher initial intake air temperatures, which caused higher compression temperatures, and led to combustion occurring earlier in the cycle. These occurrences likely produced both, higher cylinder temperatures, and higher cylinder pressures. The in-cylinder temperatures for all the

Fig. 4 – Peak in-cylinder pressure for the maximum IMEP conditions with the engine operating at an EQR of 0.26 and 0.30 using varying H_2/CO ratios.

Fig. 6 – Peak in-cylinder temperature of the best IMEP conditions of the engine operating at an EQR of 0.26 and 0.30 with varying ratios of H_2/CO .

conditions ranged from roughly 1200 K–1500 K depending on intake air temperature, mixture concentration, and fuel composition.

Fig. 6 shows a plot of the peak in-cylinder temperatures for the best IMEP conditions of the engine operating with varying H_2/CO ratios at an EQR of 0.26 and 0.30. For an EQR of 0.30, the peak in-cylinder temperature for the best IMEP conditions increase with increasing CO fraction. The peak temperatures are roughly 1340 K, 1395 K, and 1415 K for 100% H_2 , 75/25 H_2/CO ratio, and 50/50 H_2/CO ratio conditions, respectively. For an EQR of 0.26 the peak in-cylinder temperatures increase

slightly with increasing CO fraction from roughly 1350 K, 1360 K, and 1370 K for the same respective H_2/CO proportions.

Fig. 7 shows a plot of how the intake temperature must be increased with increasing CO fraction, for an EQR of 0.26 and 0.30 for the best IMEP conditions. For both equivalence ratios tested, the best IMEP intake air temperature was higher for a higher CO fraction in the syngas. For an EQR of 0.26 the intake air temperature required for best IMEP was 78 °C, 84 °C, and 98 °C, for 100% H_2 , 75/25 H_2/CO ratio, and 50/50 H_2/CO ratio conditions, respectively. For an EQR of 0.30 the same trend, but with lower temperatures, was seen with intake

Fig. 5 – In-cylinder temperature plots of the engine operating at varying intake temperatures: (a) EQR = 0.26 with 100% H_2 , (b) EQR = 0.3 with 100% H_2 , (c) EQR = 0.26 with 50/50 H_2/CO ratio, and (d) EQR = 0.30 with 50/50 H_2/CO ratio.

Fig. 7 – Intake air temperature requirement to maintain best IMEP for various H_2/CO ratios. The plot shows data points for EQR = 0.26 and EQR = 0.30.

Fig. 9 – Rapid burn angles for EQR = 0.26 and EQR = 0.30, with varying H_2/CO ratios.

temperatures ranging from 62 °C, 71 °C, and 81 °C, for the same respective H_2/CO proportions. This suggests that the mixture autoignition temperature increases with increasing CO fraction and corresponding higher intake temperature requirements. Stenlaas et al. similarly showed that the intake temperature required to achieve a certain 10–90% heat release angle was higher for RMG, compared to hydrogen [10].

5.2. Combustion characteristics

The three main combustion characteristics investigated for the various fuel blends in this study were; 1) crank angle at 10% cumulative heat release or CA10, 2) crank angle at 50% cumulative heat release or CA50, and 3) rapid burn angle or CA10 to CA90. The crank angle location of 10% of the cumulative heat release defines the SOC in this work. The CA50 parameter was selected because it is used by many in the literature to quantify combustion phasing.

For all of the conditions tested, the CA10 and CA50 locations remained relatively constant with little to no change with increasing CO fraction in the syngas mixture. This is shown in Fig. 8. For an EQR of 0.26 the CA10 remained at roughly 4 CAD ATDC and the CA50 remained at roughly 8 CAD ATDC, for all fueling proportions. At an EQR of 0.30, an

advancing CA10 and CA50 trend is seen with increasing CO fraction.

Fig. 9 shows a plot of the rapid burn angle varying with various proportions of H_2/CO for the best IMEP conditions. For an EQR of 0.26, the rapid burn angle increased slightly with increasing CO fraction, while there was no real trend for the EQR of 0.30 conditions. For all of the conditions tested, the rapid burn angles for the best IMEP conditions were between 9.5 CAD and 11.5 CAD. That is much faster than conventional SI and CI engine operation. The data also shows that the increasing CO fraction in the fuel does not have a significant role on the rapid burn angle, since all conditions were within 2 CAD of each other.

5.3. Engine efficiencies

As explained in the experimental setup section, the combustion efficiencies were based upon the precise measurement of the fuel flow into the engine and the exhaust composition and concentration, specifically hydrogen and carbon monoxide, out of the engine. Measurements of NO_x emissions were also made, but are not reported here because they were very low, which is a characteristic of low temperature combustion. The specific emissions of carbon monoxide and hydrogen are also not reported, because they are indirectly present in the

Fig. 8 – Plots showing positions for (a) CA10 and (b) CA50 for EQR = 0.26 and 0.30.

Fig. 10 – Combustion efficiency and cycle efficiency of the engine operating at an EQR = 0.26 and EQR = 0.30 with varying proportions of H₂/CO.

combustion efficiency plot, which is a more relevant and illustrative plot.

Fig. 10 shows the combustion efficiency and cycle efficiency of the engine operating at an EQR of 0.26 and 0.30 with varying proportions of H₂/CO. The combustion efficiency was between 83% and 88% for all the conditions tested. These combustion efficiencies are significantly lower than conventional compression ignition or spark ignition engine combustion efficiencies and are likely caused by the lower overall HCCI combustion temperatures which give rise to a larger quench volume between the bulk burning zone and the cylinder walls, leaving a significantly larger fuel/air mixture unburned. There was also a trend of decreasing combustion efficiency with increasing CO fraction in the fuel. For both equivalence ratios tested, the combustion efficiency went from roughly 88% and 87% for the 100% H₂ conditions, down to 84.5% and 83.5% for the 50/50 H₂/CO ratio conditions, respectively. The cycle efficiency plots show a different trend, where the lowest cycle efficiencies are observed for the 100% H₂ conditions. The trend of lowest cycle efficiency for the 100% H₂ conditions likely results from higher heat transfer at these conditions, caused by the high thermal conductivity of H₂ compared to CO. Further discussion is provided later in this section. For all the conditions, however, the cycle efficiencies

fall below 30% which is much lower than conventional CI engine operation.

Fig. 11 shows how the combusted fuel energy left the system for the various fuel compositions and equivalence ratios. Both equivalence ratios tested showed the 100% H₂ condition of having the highest fraction of combusted fuel energy leaving the system through heat transfer to the walls. The slightly higher percentage of energy leaving as heat transfer to the walls compared to conventional engines was likely due to the elevated intake temperatures and the relatively low load engine operation. The elevated intake temperatures mean that there will be more heat transfer to the walls during the compression stroke. The low load engine operation means that the heat transfer during the compression stroke, which will be relatively constant at varying loads, will have a larger impact compared to an engine operating at higher loads. Stenlaas et al. performed a similar analysis of energy leaving the system for an engine operating on H₂ and RMG in both SI and HCCI modes [10]. They too found that the proportion of fuel energy leaving as heat transfer to the cylinder walls was between 30 and 40% for the HCCI conditions, while it was only 15–30% for the SI conditions. They operated their HCCI conditions at lower speeds (800 RPM and 1200 RPM), compared to the SI conditions (1200 RPM and 1600 RPM, and attributed the increased HCCI heat loss to the lower engine speeds and not due to the different combustion regime. Their results also showed that the H₂ HCCI condition had less heat transfer to the cylinder walls, compared to the RMG HCCI condition (39% vs. 32%). From their results, a conclusion cannot be made on why there was a higher percentage of energy leaving as heat transfer operating on RMG compared to H₂, because a constant speed was not used.

The results in this paper show higher percentages of fuel energy leaving the system via heat transfer to the walls for the 100% H₂ conditions. A constant speed was used for all conditions tests, so a likely cause of this phenomenon may be the higher thermal conductivity of the hydrogen gas compared to carbon monoxide. Although the intake temperatures for the H₂/CO blended conditions were higher than the 100% H₂ conditions, which would lead to a higher thermal gradient during the compression stroke, the higher thermal conductivity of the hydrogen gas may contribute to a higher overall heat transfer rate to the cylinder walls.

Fig. 11 – The proportion of combusted fuel energy leaving the system as exhaust energy, piston work energy, and cylinder wall heat transfer energy, for the engine operating at an equivalence ratio of 0.26 and 0.30.

6. Conclusions

A single cylinder compression ignition engine was modified to run as a homogeneous charge compression ignition engine. The engine was operated on blends of hydrogen and carbon monoxide which varied from 100% H₂, 75/25 H₂/CO ratio, and 50/50 H₂/CO ratio, by volume. Two equivalence ratios were investigated; 0.26 and 0.30. The combustion characteristics and efficiencies were measured for each condition. The following conclusions were drawn from this work:

- For all of the conditions tested, increasing in-cylinder peak pressure, in-cylinder peak temperature, and increasing peak HRR were seen with increasing intake air temperature, which led to an advancing start of combustion
- The 50/50 H₂/CO ratio conditions required a roughly 20 °C higher intake air temperature to achieve stable engine operation, compared to the 100% H₂ conditions. The 75/25 H₂/CO ratio conditions exhibited similar trends and required a roughly 10 °C increase in intake temperature to achieve stable engine operation, compared to the 100% H₂ conditions
- The peak in-cylinder temperatures for all of the conditions ranged from roughly 1200 K–1500 K depending on intake air temperature, mixture concentration, and fuel composition
- Increasing CO fraction in the mixture increases its auto-ignition temperature so that a higher intake temperature is required compared to pure H₂.
- Increasing CO fraction in the fuel does not have a significant effect on the rapid burn angle. All conditions were within 2 CAD of each other
- The combustion efficiency for all conditions tested was between 83% and 88% and increased with increasing H₂ fraction
- A substantial fraction of fuel energy leaves the system as heat transfer to the cylinder walls, caused by low load engine operation and high intake air temperatures

Appendix A

The heat release rate analysis is based on the first law of thermodynamics shown in Equation (A1).

$$\delta Q_{HR} = dU - \delta W - \delta Q_{HT} \quad (A1)$$

In Equation (A1), δQ_{HR} is the heat released by combustion, δQ_{HT} is the heat transferred to the walls, δW is the work done on the piston, and dU is the change in internal energy. Substitutions must be made into Equation (A1) to get dU and δW in terms of parameters that can be measured, such as in-cylinder pressure and volume. Equation (A2) shows the definition of internal energy, or dU .

$$dU = mc_v dT \quad (A2)$$

In Equation (A2), c_v is the constant volume specific heat. The ideal gas law is shown in Equation (A3).

$$pV = mRT \quad (A3)$$

Differentiating the ideal gas law gives Equation (A4).

$$mdT = \frac{1}{R}(pdV + Vdp) \quad (A4)$$

Substituting Equation (A4) into Equation (A3) gives Equation (A5).

$$dU = \frac{c_v}{R}(pdV + Vdp) \quad (A5)$$

Substituting Equation (A5) into Equation (A1), and knowing that $\delta W = pdV$ we have Equation (A6).

$$\left(\frac{dQ}{d\theta}\right)_{HR} = \frac{c_v}{R} \left(p \frac{dV}{d\theta} + V \frac{dp}{d\theta} \right) + p \frac{dV}{d\theta} + \left(\frac{dQ}{d\theta}\right)_{HT} \quad (A6)$$

Equation (A6) can be further reduced by realizing that $R = c_p - c_v$ and the specific heat ratio $\gamma = c_p/c_v$, which gives Equation (A7).

$$\left(\frac{dQ}{d\theta}\right)_{HR} = \frac{1}{\gamma - 1} V(\theta) \frac{dP}{d\theta} + \frac{\gamma}{\gamma - 1} P(\theta) \frac{dV}{d\theta} + \left(\frac{dQ}{d\theta}\right)_{HT} \quad (A7)$$

If the heat transfer to the walls is not known, and not estimated, then the net heat release rate can be calculated by combining the two heat transfer terms as shown in Equation (A8).

$$\left(\frac{dQ}{d\theta}\right)_{net} = \left(\frac{dQ}{d\theta}\right)_{HR} - \left(\frac{dQ}{d\theta}\right)_{HT} = \frac{1}{\gamma - 1} V(\theta) \frac{dP}{d\theta} + \frac{\gamma}{\gamma - 1} P(\theta) \frac{dV}{d\theta} \quad (A8)$$

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**DUAL-FUEL DIESEL ENGINE COMBUSTION WITH HYDROGEN, GASOLINE AND
ETHANOL AS FUMIGANTS: EFFECT OF DIESEL INJECTION TIMING**

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ABSTRACT

Premixed compression ignition (CI) combustion has attracted increasing research effort recently due to its potential to achieve both high thermal efficiency and low emissions. Dual-fuel strategies for enabling premixed CI have been a focus using a low reactivity fumigant and direct diesel injection to control ignition. Alternative fuels like hydrogen and ethanol have been used as fumigants in the past but typically with diesel injection systems that did not allow the same degree of control or mixing enabled by modern common rail systems. In this work we experimentally investigated hydrogen, ethanol and gasoline as fumigants and examined three levels of fumigant energy fraction (FEF) using gasoline over a large direct diesel injection timing range with a single cylinder diesel engine. It was found that the operable diesel injection timing range at constant FEF was dependent on the fumigant's propensity for autoignition. Peak indicated gross cycle efficiency occurred with advanced diesel injection timing and aligned well with combustion phasing near TDC as we found in an earlier work. The use of hydrogen as a fumigant resulted in very low HC emissions compared with ethanol and gasoline, establishing that they mainly result from incomplete combustion of the fumigated fuel. Hydrogen emissions were independent of diesel injection timing and HC emissions were strongly linked to combustion phasing, giving further indication that squish and crevice flows are responsible for partially burned species from fumigation combustion.

INTRODUCTION

With increasing concern about fossil fuel depletion and environmental degradation, research in internal combustion engines has been directed towards improving thermal efficiency, lowering engine-out emission levels, and effectively using alternative fuels to reduce our dependency on non-renewable fuels. Due to their capability to run with higher compression ratio, leaner fuel-air charge and lower throttling

loss, compression ignition (CI) engines are more efficient than spark-ignited engines. However, the main challenge for CI engines is to meet increasingly stringent emissions legislation at an affordable cost while maintaining or improving their efficiency advantages.

One potential way to simultaneously reduce NO_x and soot emissions in CI engines without the complexity and expense of aftertreatment is to avoid conventional diffusive combustion by enabling highly premixed CI combustion [1]. For single-fuel operation, homogeneous charge compression ignition (HCCI) combustion is one such strategy and is generally used for fuels with high volatility like gasoline. NO_x and soot emissions in an HCCI engine are dramatically reduced due to low combustion temperature and a well premixed charge. However, issues like incomplete combustion at low-load conditions, relatively narrow operating range and difficulty of combustion-phasing control remain barriers to the widespread application of HCCI engines. For fuels with low volatility and high reactivity like diesel fuel, partially premixed CI combustion can be achieved by extending ignition delay using low intake temperature or applying high levels of EGR. Here, combustion phasing is more easily controlled by the fuel-injection event. However, the allowable operating range for this approach is limited to light to mid-load conditions [2] though some have achieved higher load using multiple injections, high EGR and boosted intake manifold pressure [3].

Dual-fuel premixed CI concepts like reactivity controlled compression ignition (RCCI) have been recently shown to reduce NO_x and soot emissions to very low levels while maintaining extremely high thermal efficiency over a large operating range [4-8]. The dual-fuel RCCI concept combines port fuel-injection, or fumigation, of a low reactivity fuel like gasoline and direct-injection of a high reactivity fuel like diesel to create an optimized fuel reactivity distribution. The difference between RCCI and traditional fumigation with pilot diesel injection close to TDC is that the single or multiple

injections of diesel fuel in RCCI occur early in the compression stroke to partially premix the direct-injected fuel with the surrounding gas. This allows optimum control of the combustion event and minimizes diffusion burning associated with the diesel injection.

In recent studies, RCCI using gasoline as the low reactivity fuel and diesel as high reactivity fuel were thoroughly investigated over a wide range of load conditions [4, 5, 8]. Over 50% gross indicated efficiency with extremely low emissions of NO_x and soot for most operating conditions were reported. Other groups, including work done by the authors have shown similar improvements in emissions for RCCI but with only moderate gains in efficiency [9, 10, 11]. Work in RCCI has also been extended to single-fuel strategies with ignition additives to enhance the reactivity of gasoline for use as the directly injected fuel [12, 13] and the use of alternative fuels like E85 [6] as the fumigant. E85-diesel operation exhibits significant combustion differences and requires higher quantities of diesel fuel to maintain optimal combustion phasing compared to gasoline-diesel operation under mid- to high-load conditions.

Using hydrogen as a fuel in CI engines has drawn increasing research attention due to its high auto-ignition temperature and wide flammability range, making it suitable for high-compression ratio lean-burn engines [14, 15]. Biomass gasification for power generation is one application area where a hydrogen-rich gas called syngas has traditionally been used in dual-fueled diesel engine. Dual-fueling with syngas has been thoroughly reviewed elsewhere [16]. Though it is not novel to use hydrogen as a fumigant in diesel engines, its use as the low reactivity fuel in RCCI modes has not been thoroughly investigated.

The purpose of this study is to compare hydrogen with ethanol and gasoline as fumigants in RCCI and to compare the performance and emissions characteristics of gasoline-diesel RCCI at different levels of fumigant energy fraction (FEF). A naturally-aspirated single-cylinder diesel engine was used for the experimental study. The first part of this paper discusses combustion and emissions using hydrogen, gasoline, and ethanol at a constant FEF over a large direct diesel injection timing range. In the second part, the effect of FEF on the operable diesel injection timing range with gasoline as a fumigant is examined in more detail.

EXPERIMENTAL

The experiments in this study were conducted using an Isuzu 4HK1-TC diesel engine with specifications listed in Table 1. Cylinder number 1 of the engine was isolated with separate intake and exhaust lines such that the engine could be operated in single-cylinder mode with the remaining three cylinders motored by a DC dynamometer. The stock turbocharger was removed and the engine was naturally-aspirated for the duration of the study. A laminar flow element was used to measure the intake air flow rate with a large volume surge tank downstream to dampen oscillatory flow resulting from single-cylinder operation.

Table 1: Specifications of engine used in the experimental study

Engine Type	4-stroke DI Diesel
Manufacturer/Model	Isuzu 4HK1-TC
Number of Cylinders	4, in-line
Bore x Stroke (mm)	115 x 125
Conn. Rod Length (mm)	198
Crank Length (mm)	62.5
Clearance Volume (liter)	0.0742
Displacement (liter)	1.298
Number of Cylinders	4, in-line
Total Displacement (liter)	5.192
Compression Ratio	18.5
Diesel Injection System	Common Rail, Solenoid
Gasoline/Ethanol/Hydrogen injection System	Port Fuel Injector
Rated Power	157 kW @ 2550 rpm
Rated Torque	597 N-m @ 1850 rpm

Two port fuel injectors were mounted in the intake runner of cylinder 1; one for liquid fuels like gasoline and ethanol, and the other for gaseous hydrogen. The stock common-rail diesel injector and piston bowl were retained and the compression ratio was not modified.

Table 2: Constant parameters for the investigated engine operating conditions

Condition	H-1	E-1	G-1	G-2	G-3
Fumigant	H ₂	Ethanol	Gasoline	Gasoline	Gasoline
Engine Speed (rpm)	1500	1500	1500	1500	1500
Nominal IMEP (kPa)	450	450	450	450	450
EGR (%)	0	0	0	0	0
Fumigant Energy Fraction (%)	80	80	80	70	90
Intake Temperature (°C)	35	35	35	35	35
Intake Pressure (kPa)	98	98	98	98	98
Diesel Injection Pressure (bar)	400	400	400	400	400

The constant engine operating conditions used in the experiments are listed in Table 2. For each of the five conditions, diesel injection timing was varied over the widest possible range without encountering unstable combustion as detected by a coefficient of indicated mean effective pressure (IMEP) variation greater than 4%. For each condition, a nominal IMEP was set at 450 kPa at a baseline timing setting. Fueling was then held constant for the injection timing sweep by maintaining the same injection duration and pressure, resulting in an IMEP variance of approximately ± 20 kPa. In the first set of experiments, gasoline, hydrogen and ethanol were used as fumigants, and the FEF was fixed to $80 \pm 1\%$.

These cases are denoted by G-1, H-1 and E-1. For the H-1 case, hydrogen was diluted with 60% nitrogen in part to simulate its typical concentration in syngas produced by biomass gasification [17] as well as to prevent excessively high cylinder pressure rise rate. In the second set of experiments, only gasoline was used as fumigant. The energy fractions of gasoline in total energy input were set as 70%, 80% and 90%, while the total energy input was kept constant. These three different gasoline fraction cases were denoted by G-2, G-1, and G-3.

Diesel and gasoline obtained from a local fuel station were used in the study. The gasoline had a pump octane number of 91. Physical properties of the fuels used were assumed to be equivalent to those found for light diesel, gasoline, ethanol and hydrogen in ref. [18] and were listed in Table 3.

Table 3: Physical properties of fuels used in this study

Fuel	Diesel	Gasoline	Ethanol	Hydrogen
Density @ 0°C and 1 atm (kg/m ³)	0.84-0.88	0.72-0.78	0.785	0.090
Heat of Vaporization (kJ/kg)	270	305	840	—
Lower Heating Value (MJ/kg)	42.5	44.0	26.9	120.0
Research Octane Number	—	92-98	107	130
Cetane Number	40-55	13-17	—	—

NO_x, CO and hydrocarbon emissions were measured using an AVL Fourier Transform Infrared (FTIR) analyzer on a wet basis with a hot sample line maintained to 190°C. Hydrocarbons (HC) were calculated using FTIR-measured concentrations of a collection of species using equation 1.

$$(1) \ x_{HC} = x_{CH_4} + 2 \cdot x_{C_2H_2} + 2 \cdot x_{C_2H_4} + 2 \cdot x_{C_2H_6} + 3 \cdot x_{C_3H_6} + 5 \cdot x_{IC_5} + 5 \cdot x_{NC_5} + 7.5 \cdot AHC$$

Here, x_i is the wet volumetric concentration of a species i in the exhaust gas where IC5 is isopentane, NC5 is normal pentane and AHC are total aromatics. The molecular weight of hydrocarbons was assumed to be 14 g/mol for calculation of gross-indicated specific HC.

A Raman Laser Gas Analyzer was used to measure hydrogen in the exhaust gas and an AVL Micro-Soot photo-acoustic analyzer was used for to measure soot. In-cylinder pressure data was collected with a Kistler 6056A piezoelectric transducer mounted in a glow plug adapter. An encoder with 0.36 CA° resolution was used to collect the high speed data. The start of injection (SOI) of the directly-injected diesel fuel was approximated by the 50% rise time of the measured current signal sent to the fuel injector. Figure 1 shows the overall engine experimental setup.

IMEP was calculated over the power portion of the cycle from -180 to 180 CA° and the gross apparent rate of heat release (RoHR) calculation was based on a first law analysis given in ref. [18]. Since the absolute value of heat released is not as important to this study as the bulk shape of the curve

with respect to crank angle, a constant $\gamma=1.3$ was assumed for the compression and expansion processes as opposed to derived polytropic exponents. CA05 and CA50 are the calculated crank angle locations of 5% and 50% cumulative heat release and are used to represent the ignition timing and combustion phasing, respectively. Ignition delay is defined as the period between the start of diesel injection and CA05 on a crank angle basis.

Figure 1: Schematic of Experimental System

RESULTS AND DISCUSSION

COMPARISON OF FUMIGANT TYPE

This section discusses the experimental results of changing direct diesel injection timing with gasoline, ethanol and hydrogen as fumigants with 80% FEF. Figure 2 shows how ignition delay, CA50 and indicated gross cycle efficiency ($\eta_{i,g}$) varied with diesel injection timing for the three different fumigants. The range of diesel injection timing over which stable combustion could be maintained was widest for G-1 while the range for E-1 was much narrower than the other two cases. One explanation for this narrow timing range is the very high heat of vaporization of ethanol which cools the incoming charge and causes lower in-cylinder temperature prior to the onset of combustion for E-1. A second consideration is ethanol's inhibition effect on low temperature combustion during the first-stage of heat release as demonstrated by Splitter et al. [6] where they found that a lower FEF was required when using E85 as the fumigant compared to a 95.6 RON gasoline to achieve similar engine performance. Hashimoto [19] has also shown the inhibition effect of ethanol on homogeneous ignition of n-heptane in a rapid compression machine using varying ethanol concentrations.

Figure 2: Ignition delay, CA50 and gross indicated cycle efficiency versus diesel injection timing for G-1, H-1, E-1 cases

A minimum point of CA50 as a function of injection timing exists for all three cases. Either advancing or retarding the diesel injection from this point will result in retarded combustion phasing. This trend has been seen in other studies of premixed CI combustion in engines [11]. With more advanced diesel injection, mixing time for diesel and the surrounding charge is extended, thus the local reactivity of the fuel-air mixture is decreased, leading to a late premixed combustion event. As the diesel injection timing is retarded, combustion becomes more coupled with the injection event due to higher local reactivity in the piston bowl thereby resulting in delayed ignition and combustion. The divergence of timing and combustion phasing with early diesel injection indicates primarily premixed combustion, i.e. the RCCI regime. In this region, the ignition delay and combustion phasing vary with fumigant types, with ethanol having the longest delay and the most retarded combustion phasing followed by hydrogen and gasoline. At retarded diesel injection timing, ignition delay and CA50 converge for the three fuels, suggesting that the directly-injected diesel fuel dominates ignition timing and combustion phasing like in traditional dual-fuel fumigation combustion.

The peak η_{ig} was nearly the same for all three fumigants but occurred at different diesel injection timings. For H-1 and

G-1, peak efficiency occurred in the region of more advanced diesel injection timing and corresponded well with CA50 located just after TDC as indicated by the shaded horizontal line in Figure 2. In our previous work [11], we showed that primarily premixed fumigation operation leads to peak thermal efficiency at combustion phasing near TDC since the closed portion of the cycle more closely approximates the ideal constant-volume Otto Cycle. For the E-1 case, peak η_{ig} occurred at more retarded diesel timing but also corresponded to CA50 near TDC.

Figure 3: Gross apparent rate of heat release for G-1, H-1, E-1 cases at 48°, 32°, 20° diesel injection timing

A more detailed look at combustion resulting from varying diesel injection timing for the three fumigants is shown in Figure 3 where the apparent gross RoHR is plotted versus crank angle at three timing settings. With early diesel injection at SOI=48° BTDC, the degree of diesel premixing is very high, and combustion of diesel and fumigant appears to occur simultaneously, resulting in Gaussian-shaped RoHR curves. The H-1 case shows a more delayed heat release event than G-1 but with a sharper shape of RoHR. Another interesting comparison between the two fumigants at early timing is that gasoline clearly shows an early low temperature heat release event commonly seen in premixed CI combustion whereas the H-1 case does not. Both cases used the same quantity of diesel but the absence of low temperature heat release when using hydrogen as a fumigant is further evidence that the fumigant

plays a key role in the ignition process. Similar observations have been shown and discussed by Guo et al. [20]. In a study of a diesel HCCI engine with hydrogen enrichment, it was shown that the participation of hydrogen in some reactions during low temperature heat release stage led to delayed combustion phasing as well as diminished low temperature heat release. Also, it was deduced from the data and previous work [21] that hydrogen enhances the rate of the reaction, $H + O_2 = O + OH$ which controls the high temperature heat release event and thus shortens the combustion duration.

At $SOI=32^\circ$ BTDC, the H-1 case still exhibited a largely Gaussian RoHR curve indicative of premixed combustion but the G-1 case had two distinct heat release peaks. This dual heat release is typically explained by combustion of the directly-injected diesel fuel providing ignition energy for premixed combustion of the fumigant [11]. Note from Figure 2 that at $SOI=32^\circ$ BTDC, the ignition delay for H-1 is larger than G-1 while the CA50 locations for the two cases are almost the same. This is most likely caused by the combination of inhibited low temperature heat release and promoted reaction rates during high temperature heat release by hydrogen as previously discussed. These effects also lead to the observation that although hydrogen does not allow as large of a diesel timing range as illustrated in Figure 2, it does result in a shorter premixed combustion event. The E-1 case shows very late combustion without a distinct diesel ignition curve which is indicative of ethanol's inhibition effect as previously discussed.

With late diesel injection timing at $SOI=20^\circ$ BTDC, two distinct peaks in RoHR are observed for all three fuels. Ignition occurs at nearly the same crank angle indicating that the injected diesel fuel plays a more dominant role in determining ignition. Hydrogen has the shortest combustion duration, again due to enhanced reaction rates during high temperature heat release. The E-1 case has the longest combustion duration due to the inhibition effect of ethanol extending the second heat release peak into the expansion stroke.

The emissions of NO_x , soot, CO and H_2 are given in Figure 4 for the three fumigant fuels on a gross-indicated specific basis plotted versus CA50 to examine their dependence on combustion phasing over the directly-injected diesel timing range. NO_x emissions are lower in the region of more advanced diesel injection timing and increase for the same combustion phasing as timing is retarded. This is most likely due to more heterogeneous combustion with later diesel injection. Although early injection timing lowered NO_x similarly for the three tested fuels, the H-1 case showed discernibly lower NO_x with retarded diesel timing. Here, it is possible that hydrogen had the effect of reducing diffusion burning by accelerating the combustion of injected diesel as observed by shorter combustion duration shown in Figure 3; however, more detailed conclusions about the nature of this trend is beyond the scope of this paper.

Soot emissions were low and for all three fuels over the tested timing range. This indicates that combustion was primarily premixed throughout the tested range with minimal high temperature fuel-rich regions in the cylinder. The H-1

condition exhibited the highest soot emissions with retarded diesel injection timing, possibly resulting from the high temperature burning of hydrogen combined with a late diesel injection.

Figure 4: Gross indicated specific NO_x , soot, CO and H_2 emissions for G-1, H-1, E-1 cases

Emissions of CO were extremely low for the H-1 condition indicating that the directly-injected diesel fuel does not contribute strongly to CO emissions in fumigation combustion. This corresponds well to other published studies that show CO mostly results from incomplete combustion in the squish volume for premixed modes of diesel combustion [22]. CO was nearly the same for E-1 and G-1 at retarded diesel timing settings and for G-1, retarded timing resulted in much higher CO than for advanced timing. Within the range of advanced timing for G-1, a minimum for CO was found at approximately the CA50 corresponding to the highest $\eta_{i,g}$ point in Figure 2 linking the overall thermal efficiency to combustion efficiency.

Hydrogen emissions were nearly constant over the timing range for the H-1 case. This independence from diesel injection timing and combustion phasing is evidence that hydrogen within the piston bowl region was always consumed but fumigated hydrogen in the squish and crevice regions remained. The short combustion duration for the H-1 case even with retarded combustion phasing shown in Figure 3 reinforces this idea and strengthens the argument that hydrogen enables enhanced premixed burning of both fumigant and the directly-injected diesel fuel.

Figure 5 shows a plot of detailed hydrocarbon emissions data measured using the FTIR analyzer as a function of CA50. The indicated specific emissions of HC, the sum of species given in Equation 1, shows a near monotonic dependence on combustion phasing regardless of early or late diesel injection timing. HC emissions for the H-1 and E-1 cases are extremely low because ethanol is not considered as part of the HC calculation and because the directly-injected diesel fuel is mostly consumed throughout the timing range.

The illustrative curve shown for the G-1 case in the HC plot in Figure 5 indicates that there is an asymptote in emissions as combustion phasing was advanced. This is evidence of the highest combustion efficiency occurring with phasing close to TDC. Overall, the HC emissions for H-1 and E-1 were more than ten times less than the G-1 case; therefore it is likely that the remaining hydrocarbons mostly consisted of partially burned gasoline located in the squish and crevice regions. As combustion phasing is retarded, unburned HC occur from late burning resulting in poorer combustion quality, a phenomenon shown in an optical study of premixed diesel combustion performed by Ekoto et al. [23].

Formaldehyde is also found in the exhaust emissions and has been shown to occur mostly from over-lean areas in the combustion chamber [24]. The trend shown for the G-1 case closely resembles the CO curve shown in Figure 4 where retarded injection results in higher formaldehyde emissions at retarded combustion phasing. A minimum in formaldehyde for G-1 is also seen in the advanced region establishing a linkage between formaldehyde and CO emissions for premixed CI combustion. Formaldehyde emissions for the E-1 case were the highest in the retarded diesel injection timing region and may have been due to the well-known linkage between alcohol fuels and aldehyde emissions for spark-ignited engines [25].

Ethanol emissions for the E-1 case were higher on an indicated specific basis than the overall HC emissions for the G-1 case. This is most likely due to the inhibition effect of the fumigated ethanol and later burning previously described resulting in poorer late cycle oxidation.

Comparing hydrogen with ethanol and gasoline as fumigants over a large range of diesel injection timing at constant FEF illuminates some interesting features of dual-fueled diesel modes in general. First, by using hydrogen as a fumigant, relatively low CO and HC emissions were found; therefore, the high HC and CO emissions shown in Figures 3 and 4 for the G-1 case can be assumed to originate from the fumigated gasoline. Also, based on the emissions of hydrogen

for the H-1 case, the unburned fumigant in the squish and crevice regions where cold over-lean conditions exist are essentially constant, a trend further evidenced by the shape of the HC versus CA50 curve for the G-1 case in Figure 5.

Figure 5: Gross indicated specific HC, HCHO, EtOH emissions for G-1, H-1, E-1 cases

COMPARISON OF FUMIGANT ENERGY FRACTION

The previous section showed how combustion proceeded differently when the reactivity of the fumigant mixture was changed. The reactivity of charge prior to injection of diesel can also be altered by changing the amount of fumigant used. Here to elucidate the differences between fuel and fumigant mixture three FEF levels of 70, 80 and 90 % are examined at the same nominal IMEP over a large range of diesel injection timing. The amount of diesel fuel injected was adjusted accordingly to keep the total fuel energy input constant for these three cases. Figure 6 shows the variation of ignition delay, CA50 and $\eta_{i,g}$ with diesel injection timing. As the FEF was reduced, the operational timing window became wider due to increased overall reactivity of the fuel-air mixture. At early diesel injection, the ignition delay increased for higher FEF levels, similar to reducing fumigant reactivity. At late diesel injection, nearly no difference is seen in ignition delay for all the three cases

because ignition is controlled by the injection event of diesel fuel as previously discussed.

As in the variation of fumigant type, the peak $\eta_{i,g}$ corresponded well with an overall CA50 occurring just after TDC in the advanced diesel injection region for changing gasoline energy fraction as shown in Figure 6. However, the magnitude was different as FEF was altered; higher diesel fuel content led to higher peak $\eta_{i,g}$. One possible explanation is that combustion duration is shortened as the overall reactivity of fuels is increased at the same combustion phasing, leading to more constant volume-like combustion.

Figure 6: Ignition delay, CA50 and gross indicated cycle efficiency versus diesel injection timing

To further examine combustion trends, Figure 7 shows RoHR curves at three diesel injection timing settings for the three tested FEF levels. At early injection timing of SOI=48° BTDC, combustion proceeded in a primarily premixed fashion for each case with higher gasoline fraction leading to later peak RoHR and overall longer combustion duration. It can also be observed from the RoHR at later injection timing shown in Figure 7 that the level of FEF has a predictable effect on the relative magnitude of the first and second stages of high temperature heat release. As more diesel fuel is added, the first stage of heat release becomes more dominant due to the increased reactivity of the injected charge near the location of injection.

Figure 8 shows the gross indicated specific emissions of NO_x, soot, HC, CO for at the three FEF levels. Like in Figure 4, two regions of NO_x emissions exist, one for advanced and one for retarded diesel injection timing. The NO_x emissions in the latter region are reduced with more gasoline which strengthens the argument that the diesel injection plays the key role in creating in-homogeneous regions in the combustion chamber where temperatures and mixture are sufficient for NO_x production. Soot emissions were near lower detectable measurement limits for the entire tested FEF and diesel injection timing range which indicates lean, premixed conditions throughout.

Figure 7: Gross apparent rate of heat release for G-1, G-2, G-3 cases at 48°, 32°, 20° diesel injection timing

Emissions of CO for the three FEF cases shown in Figure 8 have similar trends to the fumigant variation from Figure 4. In the advanced timing region, the combustion phasing of minimum CO coincides with the CA50 of best efficiency shown in Figure 6 again establishing a link between overall thermal efficiency and combustion efficiency. HC emissions had a strong dependence on combustion phasing independent of FEF, especially with CA50 near TDC. This confirms the earlier conclusion that HC resulting from dual-fuel combustion is mostly linked to the fumigated fuel but also indicates that the quantity of fumigated fuel is not as important. Hanson et al. [26], in recent work with an optimized piston bowl with a shallow bowl for RCCI found that HC emissions may be more linked to crevice flows than squish volume. With combustion

phasing occurring later in the expansion stroke, conditions are not favorable for oxidation of these crevice flows when they emerge, thus accounting for the strong dependence of HC emissions on CA50.

Figure 8: Gross indicated specific NO_x, soot, CO and HC, for G-1, G-2, G-3 cases

CONCLUSIONS

This work experimentally examined dual-fuel combustion in a single cylinder diesel engine at an engine load of 450 kPa IMEP. Three fumigant types; gasoline, ethanol and hydrogen and three FEF levels; 70, 80 and 90% with gasoline as a fumigant were investigated over a wide range of direct diesel injection timing. Very advanced timing led to primarily premixed CI combustion whereas more retarded timing resulted in more traditional fumigation combustion with increased diffusion burning.

One conclusion of the work is that a fumigant's propensity for autoignition has a discernible effect on ignition delay especially with early diesel injection. For example, ethanol

delayed ignition and combustion phasing, severely limited the operable diesel injection timing range and resulted in high unburned ethanol concentration in the exhaust. Hydrogen retarded the ignition but shortened the combustion duration compared with gasoline. Increasing FEF with the same fumigant delayed the ignition timing and combustion phasing as well as increased the combustion duration at early diesel injection timings.

The use of hydrogen as a fumigant allowed some interesting insights about the nature of HC and CO emissions in dual-fuel combustion. With hydrogen fumigation, CO and HC emissions were very low compared with ethanol and gasoline, establishing that they mainly result from incomplete combustion of the fumigated fuel. Hydrogen emissions were independent of diesel injection timing and HC emissions were strongly linked to combustion phasing, further indicating that squish and crevice flows are mostly responsible for partially burned species from fumigation combustion.

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NOMENCLATURE

Abbreviations	
AHC	Aromatic Hydrocarbons
BTDC	Before Top Dead Center
CA05	Crank Angle location of 5% Gross Heat Release
CA50	Crank Angle location of 50% Gross Heat Release
CI	Compression Ignition
EGR	Exhaust Gas Recirculation
FEF	Fumigant Energy Fraction
FTIR	Fourier Transform Infrared
IC5	Isopentane
IMEP	Indicated Mean Effective Pressure
HC	Hydrocarbon
HCCI	Homogeneous Charge Compression Ignition
NC5	Normal Pentane
RCCI	Reactivity Controlled Compression Ignition
RoHR	Rate of Heat Release
RON	Research Octane Number
SOI	Start Of Injection
TDC	Top Dead Center
Symbols	
$\eta_{i,g}$	Gross indicated thermal efficiency

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