ENGINE PERFORMANCE, COMBUSTION AND EMISSIONS
EVALUATIONS OF A DIESEL NATURAL GAS DUAL FUEL ENGINE

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ABSTRACT
Dual fuel engines are being utilized more due to stricter emission standards, increasing diesel fuel cost, and decreasing natural gas cost. These engines are originally sold as diesel engines. They are converted to dual fuel operation with an aftermarket dual fuel kit. Natural gas is blended with the intake air. As this occurs the amount of diesel used is reduced. The maximum natural gas substitution is limited by knock or emissions of carbon monoxide (CO) and total hydrocarbons (THC). In this work a John Deere 6068H diesel engine is converted to operate as a dual fuel engine. The engine is a tier II 6 cylinder, 6.8 liter, 4-stroke compression ignition engine with a compression ratio of 17:1 and a power rating of 168 kW at 2200 rpm. This work aims to evaluate emissions and efficiency of a diesel derivative dual fuel engine. A natural gas fuel system was installed to deliver fuel upstream of the turbocharger compressor. The engine was operated at 1800 rpm through five different load points in diesel and dual fuel operating modes. Fuel consumption and pollutant emissions were measured. Elevated carbon monoxide and hydrocarbon emissions were observed at low loads for dual fuel operation. Overall CO and THC emissions increased for dual fuel operation. However, particulate matter (PM) and nitrogen oxides (NOx) on average were significantly decreased.

Keywords: dual fuel engine, combustion, engine emissions, diesel engine.

INTRODUCTION
Diesel engine has become a favorable engine due to high fuel economy, reliability and power output. It powers most of the nation's trucks, buses, locomotive, ships, farm equipment and stationary power-generating engines. Diesel engine converts chemical energy of fuel into thermal energy and later transforms this thermal energy into mechanical energy in combustion chamber. Common exhaust emissions from a compression ignition (CI) engine are oxides of nitrogen (NOx), carbon monoxides (CO), total hydrocarbons (THC) and particulate matters (PM).

Many researchers have been searching for a way to overcome issues with high emissions with diesel engine while maintaining the standard original engine infrastructure. Various solutions have been proposed, including utilizing alternative fuels as a dedicated fuel in diesel pilot ignition engines, gas turbines, dual fuel and bi-fuel engines. Among these applications, one of the most promising options is the diesel derivative dual fuel engine with natural gas as the supplement fuel. The diesel derivative dual fuel engine is one of the concepts used to overcome problems with diesel engines. A dual fuel engine is a diesel engine with an aftermarket or OEM kit that allows for utilization of natural gas during combustion. No major modification to the diesel platform is required in order to convert the diesel engine to dual fuel operation. In dual fuel systems, air and natural gas are mixed before being inducted into the intake manifold. The mixture is ignited when diesel fuel is injected near top dead center of the compression stroke, acting as an ignition source. Natural gas is the primary fuel in this system, and thus it controls the total power output from the engine once dual fuel operation is commenced. A significant advantage of dual fuel is its flexibility; these units can operate with natural gas substitution (diesel displacement) or solely diesel should a gas supply not be available.

Interest in natural gas replacement of diesel fuel in compression ignition engines have substantially increased in recent years. Natural gas, which is primarily composed of methane, is considered to be a viable alternative to liquid petroleum-based fuels. As a global resource natural gas is abundant. It is particularly well suited for dual fuel applications where it is introduced to high compression ratio engines due to high octane numbers and auto-ignition temperatures. The dual fuel engine can be run with 100% diesel fuel or operate as a dual fuel engine with the availability of natural gas. The engine will not operate as a dedicated natural gas unit because an ignition source is required.

Dual fuel is proven to reduce NOx and soot emissions compared to 100% diesel. However, CO and THC emissions are considerably higher especially at low and intermediate loads [1-4]. Many researchers have reported these challenges due to the poor utilization of gaseous fuel at low load in dual fuel engines. A number of research investigations have been published analyzing the variation of diesel fuel quantity, the effect of injection timing, and the introduction of exhaust gas recirculation to control emissions and performance [5-11].
This project’s goal is to evaluate the performance and combustion characteristics from diesel derivative dual fuel engines on a 6-cylinder John Deere 6068H. Additionally, exhaust emissions measurement is performed in order to maintain compliance with Tier 2 emissions levels. The engine does not have any after treatment system. It is theorized that at part load conditions, the natural gas - air mixture is below the flammability limit in the cylinder, while penetration of the diesel fuel jet does not extend across the cylinder. Consequently, regions near the edge of the combustion chamber with natural gas and air mixtures below the flammability limit are either unburned or partially combusted. For the baseline conditions, the primary parameters presented here are in-cylinder pressure trace, heat release rate (HRR), mass fraction burned profile, brake specific fuel consumption (BSFC), efficiency, NOx, PM, CO and THC. An analysis is also presented showing the dual fuel performance in terms of ISO weighted average emissions based on the appropriate ISO test cycle.

**EXPERIMENTAL APPARATUS AND PROCEDURE**

Experiments were carried out on a four-stroke, turbocharged and after cooled, 6 - cylinder direct injection diesel engine. The engine has a 6.8 liter cylinder volume and compression ratio of 17:1. The engine was loaded with an eddy-current dynamometer. The dynamometer is manufactured by Marathon Electric, model number JVN. It is controlled by a variable frequency drive by Eaton Corporation that has a rating of 350hp at 480 VAC. A LabView Virtual Instrument VI is programmed to remotely control the system. The default manufacture injection timing was set at 6.5° before top dead center (bTDC) and speed at 1800 rpm were used throughout the diesel experiments. The instrument panel on the test bed indicated engine speed, brake power and various temperatures. A sketch of the instrumented engine setup and its specification is shown in Figure-1 and Table-1. The photo of the experimental apparatus is shown in Figure-2. The gaseous fuel used in this experiment is natural gas supplied by the Fort Collins, CO city distribution system. The original equipment manufacturer (OEM) engine control module (ECM) will detect this discrepancy in power production and automatically reduce the injection timing. A sketch of the instrument panel on the test bed indicated engine speed, brake power and various temperatures. A sketch of the instrumented engine setup and its specification is shown in Figure-1 and Table-1. The photo of the experimental apparatus is shown in Figure-2.

The Rosemount 5-gas analyzer was used to collect NOx, CO and THC emissions data. The exhaust samples were passed through a Pelilt-type condenser before entering the gas analyzer, to obtain complete drying of the gases prior to entering the analyzer. The analyzer uses infrared radiation (IR) to sample and measure the CO concentrations in the exhaust gases, while the THC is detected using a flame ionization detection (FID) method. A regulated flow of sample gas passes through a flame sustained by regulated flows of fuel gas and air. Within the flame, the hydrocarbon sample stream undergoes a complex ionization that produces electrons and positive ions, which are collected by an electrode, causing a measurable current flow. The ionization current is proportional to the rate at which carbon atoms enter the burner and is therefore a measure of the concentration of hydrocarbons in the sample. A chemiluminescence technique is used to detect NOx formation. The concentration of NOx is directly proportional to the intensity of the chemiluminescence which relies on the measurement of light produced by the gas-phase reaction of NO and ozone (O3). NO is a relatively unstable molecule which will readily oxidize to NO2 especially in the presence of O3. This reaction produces a quantity of light for each NO molecule which is reacted and measured by a photodiode.

PM in the exhaust was measured using a dilution tunnel. The dilution tunnel collects particulates suspended in the exhaust and mixes them with clean laboratory air. Laboratory air is inducted by a pump and drawn through a High Efficiency Particulate Air (HEPA) filter. The exhaust sample flow is measured using a venturi/differential pressure measurement and the dilution air flowrate is measured with a turbine meter. The purified air and exhaust sample flow are mixed in the dilution tunnel, then cooled down and equilibrated in a large residence chamber. The mixture is pulled from the base of the residence chamber by a pump and passed through the filter assembly. The particulate sample is collected onto a pre-weighed Teflon or quartz filter which is then weighed again to give the mass of sample collected. Ball valves located upstream and downstream of the filter holder are used to stop the flow when a data point is not being taken and to isolate the filter housing when filters are being changed.

In all tests, the engine was allowed to warm up to reach steady state condition, which was determined by monitoring the intake manifold and block coolant temperature at 43°C and 88°C. Once steady state operation was reached, a complete set of data would be collected. Test runs were started by operating the engine at 1800 rpm while being fueled with baseline diesel fuel at different load settings to obtain different rates of energy input. Similar experiments then proceeded with diesel and natural gas mixture over a similar range of amounts of energy input. During dual fuel operation, the power output of the engine is controlled by varying the amount of natural gas inducted into the system. This is achieved by adjusting the gas control valve. The injection timing was left at stock configuration and is varied by the ECM according to the amount of diesel fuel injected. In all experimental runs, the pressure traces developed inside the cylinder were monitored using Kistler PiezoStar Type 6056A in-cylinder pressure sensors. The sensors measure combustion pressure. These sensors rely on the measurement of light produced by the gas-phase reaction of NO and ozone (O3). NO is a relatively unstable molecule which will readily oxidize to NO2 especially in the presence of O3. This reaction produces a quantity of light for each NO molecule which is reacted and measured by a photodiode.

The dual fuel kit system used in this experiment consists the following major components: Programmable Logic Controller (PLC)-based control panel, supply gas filter, pressure regulator, safety solenoid shut-off valve,
and mixer. Two pressure regulators control the natural gas pressure supply. From the natural gas supply, the incoming gas first encounters two pressure regulators. The first regulator regulates the pressure to 5 psi and the second regulator reduces the pressure to approximately 2 psi. A gas filter is located in the gas train to remove any particulates from the gas stream, thus protecting the control valves and engine componentry. A gas shut-off solenoid valve is also included and is controlled by the PLC.

Table-1. Specifications of the John Deere 6068H.

<table>
<thead>
<tr>
<th>Specification</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>106 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>127 mm</td>
</tr>
<tr>
<td>Connecting Rod</td>
<td>203 mm</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>17:1</td>
</tr>
<tr>
<td>Normal operation speed</td>
<td>1800 rpm</td>
</tr>
<tr>
<td>Number of nozzle holes</td>
<td>6</td>
</tr>
<tr>
<td>Nozzle hole diameter</td>
<td>1.75E-04 cm</td>
</tr>
<tr>
<td>Nominal start of injection timing</td>
<td>6.5 bTDC</td>
</tr>
<tr>
<td>Rated power</td>
<td>205 kW (275 hp)</td>
</tr>
<tr>
<td>Rated speed</td>
<td>2400 rpm</td>
</tr>
<tr>
<td>Inlet valve closure</td>
<td>-156.75°</td>
</tr>
</tbody>
</table>

Table-2. Typical natural gas composition seen during testing. Percentages by volume.

<table>
<thead>
<tr>
<th>Component</th>
<th>Percentage by Volume</th>
</tr>
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<tbody>
<tr>
<td>Methane</td>
<td>94%</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>1%</td>
</tr>
<tr>
<td>Carbon dioxide</td>
<td>1.3%</td>
</tr>
<tr>
<td>Ethane</td>
<td>3.14%</td>
</tr>
<tr>
<td>Propane</td>
<td>0.45%</td>
</tr>
<tr>
<td>Butane</td>
<td>0.11%</td>
</tr>
</tbody>
</table>

Experiment Test Plan
Two sets of testing were performed on the Tier-II JD6068 diesel engine. Tests were conducted for the diesel
engine with dual fuel system turned off (diesel baseline) and for normal dual fuel operations (dual fuel baseline). Baseline dual fuel operation corresponds to the nominal natural gas substitution map recommended by the manufacturer. Points at five engine loads were taken corresponding to 12%, 25%, 50%, 75% and 100% of the 139kW maximum load attainable from the test setup at altitude. The engine speed was maintained at 1800 rpm for all cases. These testing parameters were determined after consulting ISO Standard 8178:4 Cycle D2. Table 3 shows the tests performed and the analyzers used on each test. Table 4 & 5 shows the operating conditions of the dual fuel engine operation runs on diesel and natural gas. At 100% load for the dual fuel baseline, the engine automatically retards the timing to be nearer to TDC. Other than that, the injection timing was 6.5° bTDC. The diesel displacement represents the percentage of natural gas substitution in the diesel engine.

The substitution level corresponds to the amount of diesel reduction realized as natural gas is added via the dual fuel kit. It is computed by measuring the diesel fuel mass flow rate for diesel baseline and the diesel fuel mass flow rate in the dual fuel engine. The difference of these mass flow rates represents how much natural gas is substituted in the dual fuel engine. The equivalence ratio is defined as the stoichiometric air/fuel ratio divided by the actual air/fuel ratio. The actual natural gas air fuel ratio was evaluated in several steps. The actual air/fuel ratios were calculated for baseline diesel using exhaust gas analysis. Using measured fuel consumption and the air/fuel ratios the air flow through the engine was computed.

**RESULTS AND DISCUSSIONS**

Measurements were taken under two different operational modes, baseline diesel and dual fuel operation. To examine the phenomena inside the combustion chamber, the cylinder pressure, net heat release rate, and mass fraction burned data are presented. The raw net heat release data is smoothed a running average of ten data points. Emissions were collected according to the appropriate ASTM, EPA, and ISO standards and analyzed using the equipment described previously.

**Engine performance**

Engine performance is most significantly represented by engine brake thermal efficiency. Alternatively, the BSFC is a useful parameter describing the power output per unit of fuel consumed and is essentially the inverse of the engine efficiency. The brake thermal efficiency is calculated by using inverse proportion of BSFC and the weighted average lower heating value (LHV). Figures 3 & 4 show efficiency and BSFC for the diesel and dual fuel baseline cases. At low loads, the efficiency is lower for dual fuel operation compared to normal diesel operation while the BSFC is
higher for dual fuel operation. This may be due to slower combustion rate and poor utilization of the gaseous fuel in the combustion chamber. Additionally, on a mass basis the specific heat of natural gas is higher than pure air and higher than diesel vapor. This may also play a role by reducing combustion temperature and consequently slowing the combustion process. The trends are improved at intermediate and high loads.

Cylinder pressure and heat release rate

Figures 5 and 6 show combustion pressures and net heat release rates vs. crank angle at 12% and 25% loads. These loads are characterized as low loads. At low loads, the cylinder pressure is lower in dual fuel operation due primarily to later heat release. While the motoring pressure peaks of both traces are nearly identical, the dual fuel case fails to reach similar peak pressures as the diesel case due to a late pressure peak. Also at both loads, the diesel case reaches complete combustion more quickly. According to Heywood [12], the typical HRR diagram of a CI engine should have a total of four stages of combustion, starting with start of injection, premixed, mixing controlled and late combustion phase. For low loads configurations it is shown that only premixed combustion phase appears in both operation.

Figure-7 shows pressure and heat release rate trend at 50% load. Dual fuel motoring and peak combustion pressures are slightly lower than the diesel condition. The heat release rate is also lower in dual fuel engine. The diesel case shows two heat release peaks, the first premixed combustion and the second mixing controlled combustion. For dual fuel only one heat release peak is observed.

The pressure and heat release rate profile at higher loads (75% and 100%) are presented in Figures 8 and 9. The motoring pressures for dual fuel are lower at 75% and 100% loads. Peak combustion pressures double over the selected testing range. Although the compression ratio is fixed, the intake air pressure is much higher at 100% load compared to 12% load. Two heat release peaks (premixed and mixing controlled combustion phase) are present for diesel and dual fuel at the higher loads. At high loads, dual fuel and diesel combustion appears similar. Both regimes have completed combustion by approximately 45° after TDC.

Exhaust emissions

Figure-10 shows the variation of PM with load. Under dual fuel operation, PM emissions are lower for every load except high load, which exhibits 60% higher PM than normal diesel operation. In normal diesel operation, the PM emissions show a decreasing trend with increasing load. The general PM reduction in dual fuel operation is expected. Natural gas contains primarily methane, the simplest hydrocarbon, which tends to produce lower PM emission. PM formation is dependent on the fuel composition. The carbon content of diesel is higher than natural gas and tends to produce more PM. PM formation is also dependent on the combustion regime. CI diesel engines are characterized by mixing controlled combustion. In dual fuel operation, the fraction of premixed natural gas combustion increases while the fraction of mixing controlled diesel combustion decreases. Generally this tends to reduces PM emissions. The high load data was unexpected, and the explanation is unclear. The high load data points were subsequently repeated on a different test days, confirming the result.

Figure-11 shows measured NOx emissions at five loads for the two operating modes. The figure indicates that dual fuel operation emits less NOx throughout the load map. NOx formation is dependent on the oxidation of atmospheric nitrogen. The most significant component, NO is described by the extended Zeldovich mechanism, which is strongly affected by cylinder charge temperature. In dual fuel operation, part of the combustion process occurs in the lean, premixed regime. For normal diesel operation, most of the fuel is burned as a diffusion flame near stoichiometric equivalence ratio. Stoichiometric combustion produces higher NOx due to higher combustion temperatures. Both operating modes display minimum NOx production at about 75 kW and follow similar trends. Thus, it appears that the same combustion and engine operating phenomena are affecting both operating modes.

The relationship between CO emissions and load is shown in Figure-12 for both operating modes. Emission of CO is indicative of incomplete combustion. CO emissions are dependent on equivalence ratio, partially burned gaseous fuel, and cylinder charge temperature. According to the figure, it is revealed that dual fuel operation suffers from high CO emissions, particularly at low loads. Diesel operations, on the other hand, produce low CO emissions. At low loads, duel fuel combustion is characterized by slower combustion rates compared to diesel, which results in lower cylinder charge temperature and poor quality combustion. At intermediate loads, dual fuel combustion is improved with lower CO emissions compared to low loads. This declining trend continues at high loads. However, even with this reduction, CO emissions are still higher than normal diesel operation. The extent of diesel jet penetration is another factor that likely influences incomplete combustion and CO emissions at low loads. At low loads, the mass of diesel fuel injected is much smaller than high load for dual fuel operation. When less mass is injected the diesel jet penetration is not as great and less likely to extend throughout the combustion chamber. This reduces the probability of complete combustion of the entire air and natural gas mixture.

Figure-13 shows THC levels in the exhaust for dual fuel and diesel operating modes across the load map. THC emissions for both cases decrease with increasing load and are minimum at 100% load. THC emissions are controlled by the quality of the combustions process inside the cylinder. The trend in THC emissions is similar for dual fuel and diesel, but dual fuel THC emissions are nearly 100 times that of diesel THC emissions. An explanation can be made similar to that made for CO emissions. The air and natural gas is very lean, especially at low loads. Referring to Table 5, the natural gas
equivalence ratio varies from 0.15 to 0.27. At this ratio, flame propagation is slow or non-existent. The extent of penetration of the burning diesel jet may play a critical role in oxidizing the lean natural gas/air mixture. The high THC emissions for dual fuel are therefore a product of incomplete combustion, primarily, of the lean air and natural gas mixture.

Regulated emission

Figure-14 shows ISO 8178 weighted emissions for diesel and dual fuel operation. The John Deere Tier II 6068H compression ignition engine is required to meet the US EPA limits at sea level conditions. These limits are indicated on the figure. NOx is combined with Non-Methane Hydrocarbons (NMHCs). The NMHC emissions are evaluated by subtracting methane measured with the FTIR from THC. In normal diesel mode the engine meets the Tier II limits, although the test was carried out at an altitude of 5000 feet. In dual fuel mode, however, NOx + NMHC and CO limits are exceeded. PM emissions are reduced further below the Tier II limit. The limit for NOx + NMHC is exceeded because, although NOx is reduced, the increase in NMHC emissions is much larger. Possible approaches to meet Tier II limits in dual fuel operation are: (1) modify PLC control to operate with 100% diesel at low loads, and (2) employ an oxidation catalyst. These approaches could be employed independently or combined.

Figure-3. Engine performance in terms of efficiency for diesel and dual fuel baseline.

Figure-4. Engine performance in terms of BSFC for diesel and dual fuel baseline.

Figure-5. Pressure trace and heat release rate profile at 12% load.

Figure-6. Pressure trace and heat release rate profile at 25% load.

Figure-7. Pressure trace and heat release rate profile at 50% load.

Figure-8. Pressure trace and heat release rate profile at 75% load.
**CONCLUSIONS**

This paper presented a study about experimentally investigating the effect of natural gas addition on improving the emission formation of a diesel engine at various loads. The main consequences are listed as follows:

a) Significant diesel displacement is observed at all loads with a maximum of 70% at 50% load. This substitution rate is restricted by the knocking, engine stability and increasing of CO and THC emissions.

b) At 12% and 25% loads, the dual fuel engine in-cylinder pressure is lower than diesel baseline. At 12% load, the peak HRR reduction in dual fuel engine is 17% while at 25% load, the reduction of peak HRR is approximately 40%.

c) At 50%, 75% and 100% loads, the in-cylinder pressure for dual fuel operation is improved with a slightly higher than the diesel engine. The HRR profiles showed two peaks in both operations except at 50% load where one peak is observed in dual fuel engine. The dual fuel HRR rising earlier than diesel at these loads. This is due to advanced injection timing by the engine.

d) Diesel operation produces higher brake thermal efficiency at low load whereas dual fuel achieves higher brake thermal efficiency at intermediate and high loads.

e) Maximum PM emission reductions was achieved at 60% lower than the diesel baseline when natural gas...
displaced 35% of the diesel fuel in dual fuel operation at 12% load.

f) NOx emission was reduced by up to 47% in dual fuel operation. Maximum reduction occurs at low loads when the engine operates with very lean premixed mixture.

g) Dual fuel operation emits up to 98% of THC emissions and 70% of CO emissions. Even though the trend is decreasing with loads, the values are still considerably higher than diesel baseline.

h) EPA regulated NOx + NMHC and CO emissions are exceeded for dual fuel operation. To meet emissions limits with dual fuel it is suggested that disabling the dual fuel mode at low loads employing an oxidation catalyst be investigated.

It is clear from the above conclusions that the addition of natural gas can provide dramatic reductions in NOx and PM emissions of CI engines. However, this creates adverse results in CO and THC emissions. In order to reduce and control further the CO and THC emissions in dual fuel engine, the use of oxidation catalyst is recommended. Oxidation catalyst technology has been proven to be capable of reducing up to 93% of CO and 85% of non-methane hydrocarbons (NMHC) in dual fuel engine [13].

REFERENCES


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