VOL. 10, NO 20, NOVEMBER, 2015 ISSN 1819-6608
ARPN Journal of Engineering and Applied Sciences

www.arpnjournals.com

EFFECTS OF DIESEL DISPLACEMENT ON THE EMISSIONS CHARACTERISTICS OF A DIESEL DERIVATIVE DUAL FUEL ENGINE

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ABSTRACT

In this research a John Deere 6068H diesel engine is converted to dual fuel operation. The engine is a Tier II, 6 cylinders, 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. A natural gas fuel system is installed to deliver fuel upstream of the turbocharger compressor. The engine operates at 1800 rpm through five different load points in diesel and dual fuel operating modes. The natural gas substitution values tested are representative of standard dual fuel tuning, with a maximum diesel displacement of 70%. Emissions were collected according to the appropriate ASTM, EPA, and ISO standards. Through the experimental investigations, it is shown that dual fuel engines are capable of reducing nitrogen oxides (NO_x) and particulate matters (PM) emissions. However, dual fuel engines emitted excessive total hydrocarbons (THC) and carbon monoxides (CO) especially at low and intermediate loads. In order to formulate an optimized substitution scheme that would reduce these emissions in dual fuel engines, a natural gas substitution sweep as a result of diesel displacement was conducted at each load. Results showed a reduction in CO and THC emissions with optimized substitution scheme when compared with dual fuel baseline emission. Therefore, it is suggested that dual fuel would be turned off until intermediate loads were reached. Subsequent target diesel displacements were determined by selecting the highest diesel displacement observed during the natural gas substitution sweeps which maintained engine stability.

Keywords: dual fuel engine, diesel natural gas engine, emissions, diesel displacement.

INTRODUCTION

In the effort to reduce undesirable emissions and expensive diesel fuel cost, many have proposed alternative ways to combine cleaner and lower cost gaseous fuel with diesel fuel as a diesel fuel supplement. Gaseous fuels are considered to be good alternative fuels for passenger cars, truck transportation and stationary engines that can provide both reduced pollutant emissions and energy security (M.Y.E. Selim, 2004). Most combustion devices are easily adaptable to the use of gaseous fuels for power production. Gaseous fuels with high octane numbers such as natural gas and biogas have a knock resistance which makes them suitable for engines with relatively high compression ratios. Gaseous fuels produce less pollutant exhaust emissions if appropriate conditions are satisfied for its mixing and combustion (B.B. Sahoo *et al*., 2009).

There have been many published studies on the use of gaseous fuels in compression ignition (CI) engines such as biogas, producer gas, natural gas and liquefied petroleum gas (LPG). These fuels vary in chemical composition, which has a significant impact on engine emissions and performance. Therefore, the choice of fuel is important.

Interests in natural gas replacement of diesel fuel in CI engines have substantially increased in recent years. Natural gas in dual fuel engines produce high output torque and a better thermal efficiency compared to LPG due to a larger heating value (M.Y.E. Selim, 2004). It also helps to reduce the NO_x emissions by lowering the charge temperature, which reduces the NO_x formation (M. S.Lounici *et al*., 2014). Methane is the main component in natural gas and has a high octane number, greater than 120, which makes it knock resistant and suitable for high compression ratio engines. Natural gas is also favorable because it contains minimal impurities such as sulfur and when used with diesel as a dual fuel engine, tends to reduce PM compared to conventional diesel engines (R.G. Papagiannakis *et al.*, 2004).

The developments of horizontal drilling and hydraulic fracturing technologies have improved the exploration of natural gas from shale reservoirs. Shale gas, which is a natural gas that is trapped within the shale formations, has become the fastest growing source of natural gas in the USA. In the USA, at least 2 million wells of oil and shale gas have been hydraulically fractured since 1860. Today 95% of new wells are hydraulically fractured which accounts for more than 43% of total U.S. oil production and 67% of natural gas production (K. F. Perry, 2010).

Even though natural gas fumigated CI engines can reduce the NO_x and PM emissions, it is also observed that CO and HC emissions were higher especially at part load compared to conventional diesel engine (R.G. Papagiannakis *et al*., 2004). The combustion in natural gas

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fumigated diesel engines is slower at part load but improved at higher load. This way of reducing pollution from diesel exhaust emissions is becoming more common. A diesel engine fumigated with natural gas is called a diesel derivative dual fuel engine.

The dual fuel system is a retrofit to the CI engine. The dual fuel engine works using a CI engine but operates on a combustion process with characteristics from both spark ignition (SI) and CI engines. The dual cycle contains features of both Otto and Diesel cycles. In the dual fuel engine, natural gas is fumigated with air in the intake stream by a venturi installed before the turbocharger. Natural gas flow is controlled by a throttle. The amount of natural gas admitted into the intake air stream is dependent on the engine load and speed. The mixture is compressed and a small amount of diesel fuel is injected near the end of compression stroke to initiate combustion. No modifications are made to the internal workings of the engine or the diesel injection system. The natural gas will displace some of the diesel required to run the engine, decreasing diesel fuel consumption for the same power output (W.N. Wan Mansor *et al*., 2014). In the dual fuel natural gas-diesel engine, the transitions between diesel and dual fuel modes can be achieved while the engine is running without any interruption to the required engine load. If natural gas is not available, the control valve will be shut off and the engine continues on diesel fuel as a conventional diesel engine.

There are substantial numbers of studies investigating emissions from dual fuel engines. Research studies have reported that presence of gaseous fuels in dual fuel operation reduces the amount of diffusion combustion and replaces it with lean premixed combustion, which affects the rate of THC and CO formation especially at low and intermediate loads (Uma *et al*., 2004). To overcome elevated CO and THC emissions in dual fuel engines, quantity of pilot fuel is reported to be one of the most important parameter for controlling the emissions (Sombatwong *et al*., 2013). However, almost none have investigated limits of dual fuel operation in John Deere Tier 11 engine and formulated an optimized substitution scheme that will maximize diesel displacement, hence reducing emissions.

The objective of the current study is to study the effect of diesel displacement in dual fuel engines that will reduce THC and CO emissions. Since dual fuel engines produce higher THC and CO emissions than regulations allow, concepts will be developed to advantageously control combustion and emissions in this particular engine.

EXPERIMENTAL SETUP AND TEST PLAN

Design of Experiment

The engine used is a 6-cylinder Tier II, 6.8 liter John Deere 6068H diesel engine with bowl-shaped piston/combustion chamber. It is equipped with a turbocharging system and a high-pressure common rail fuel injection system. The basic specifications are presented in Table-1. The dual fuel system is a retrofit to the diesel engine. Instead of aspirating only air, natural gas is supplied to the intake upstream of the turbocharger. This allows the air and gas to be mixed by the turbocharger. This premixed charge then passes through the intercooler and into the intake manifold. No modifications are made to the internal workings of the engine or the diesel injection system. The natural gas will displace some of the diesel required to run the engine, decreasing diesel fuel consumption for the same power output. The dual fuel kit system 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. The test system is presented in Figure-1 and discussed in the experimental setup section. Incoming natural gas composition to the laboratory varied and is slightly different at each point. Table-2 shows a typical composition of natural gas during the testing.

Table-1. Engine specifications.

Bore	**106 mm**
Stroke	127 mm
Connecting Rod	203 mm
Compression Ratio	17:1
Normal operation speed	1800 rpm
Number of nozzle holes	6
Nozzle hole diameter	1.75E-04 cm
Nominarl start of injection timing	6.5 bTDC
Rated power	205 kW (275 hp)
Rated speed	2400 rpm
Inlet valve closure	-156.75°

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

Methane	**94%**
Nitrogen	1%
Carbon dioxide	1.3%
Ethane	3.14%
Propane	0.45%
Butane	0.11%

Experimental Apparatus

A schematic of the engine system is shown in Figure-1. From the natural gas supply, the incoming gas first encounters two pressure regulators. The first regulator

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regulates the pressure to 5 psi and the second regulator, which is included in the gas train, 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. It is energized before the gas control valve is allowed to open. This control valve is a Woodward L-Series Air/Fuel Ratio Control which provides precise air-fuel ratio control for engines. It is a microprocessor-based actuator with built-in speed control and internal fault detection. The valve is directly mounted to the mixer, which is located between the clean air intake and turbocharger. The air-gas mixture is then fed to the turbocharger, which facilitates complete mixing of the natural gas with air before it is fumigated into the intake stream.

The test engine is equipped with a fixed geometry turbocharging system and a high pressure common rail fuel injection system. The high injection pressures are maintained at all engine speeds by the common rail fuel injection system which used a high pressure pump and a pressure accumulator. The engine is loaded with an AC motoring dynamometer that has maximum rated power of 300 hp and maximum rated speed of 2800rpm.

The exhaust emissions captured in this test include CO, THC, NO_x, carbon dioxides (CO_2), oxygen (O_2), PM, volatile organic compounds (VOCs), methane, and ethane. However, only CO, NO_x, THC and PM results are presented in this paper. A Rosemount 5-gas emissions bench measures CO, CO_2, THC, NO_x and O_2 concentrations. Table-3 presents each analyzer and the measurement technology. PM is measured using a dilution tunnel. The dilution tunnel blends exhaust gas with clean purified laboratory air. The mixture is cooled down and equilibrates in a residence chamber. The mixture is pulled from the residence chamber by a pump through a 10 µm cyclone and a filter assembly. Filter assembly uses pre-weighted Teflon filters to collect particulate matter. The Teflon filters weighed post-test to give the mass of sample collected. Finally, VOCs, methane, ethane, and other compounds are measured using a Fourier Transform Infra-Red (FTIR) spectrometer.

Test Plan

Tests were conducted for the diesel baseline with the dual fuel system off and for normal dual fuel operation. Data at five different engine loads were taken corresponding to 12%, 25%, 50%, 75% and 100% of the 165 kW maximum load achievable on site at 1800 rpm. The engine speed was maintained at 1800 rpm for all cases. These testing parameters were determined after consulting ISO Standard 8178:4 Cycle D2. Experiments were carried out at steady state with an intake manifold temperature set point of 43°C and block coolant temperature maximum of 88°C.Under dual fuel operation, varying the amount of diesel fuel injected and natural gas inducted into the system controls the power output of the engine. The engine governor was configured in speed control mode to simulate a generator application. The PLC controlled substitution map for the dual fuel system is a result of the throttle valve position vs. intake manifold pressure map. In this case it is selected to maximize diesel displacement across the load map while maintaining stable operation. The resulting baseline substitution map observed in this installation is shown in Table-4.

Diesel displacement baseline is calculated using Equation-1 and is shown in Table-4. The diesel displacement represents the percentage of natural gas substitution in the diesel engine.It is computed by measuring the diesel fuel mass flow rate for diesel baseline, $\dot{m}_{\text{Baseline Diesel}}$, and the diesel fuel mass flow rate in the dual fuel engine, $\dot{m}_{\text{Diesel in Dual Fuel Mode}}$. The difference of these mass flow rates represents how much natural gas is substituted in the dual fuel engine. It is shown in Table-4 that the maximum displacement in the dual fuel engine is 70%, which is limited by engine stability and knocking. Further increasing the natural gas flow in the system caused engine instability, observed as significant vibrations in the test cell. In-cylinder pressures also dropped significantly. Considering these issues, maximum diesel displacement baseline for this setup is 70%.

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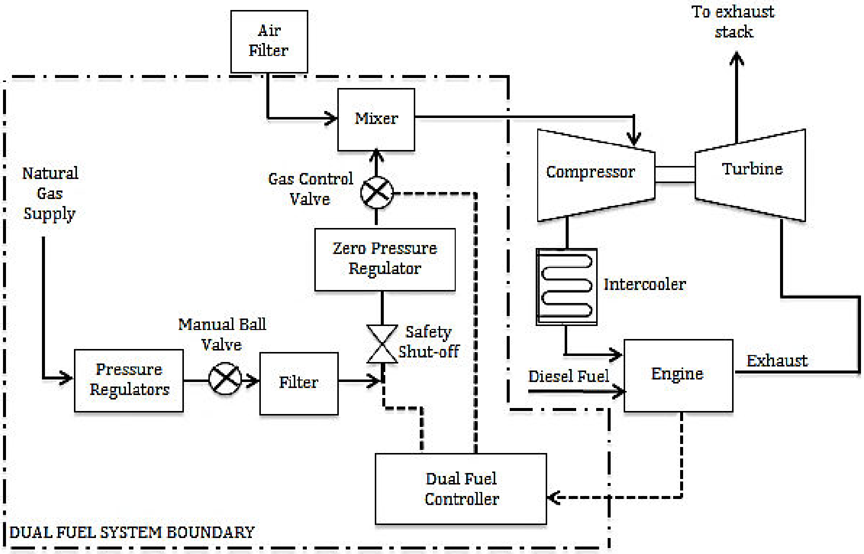

Figure-1. The schematic diagram of John Deere diesel engine with a dual fuel kit.

Table-3. Testing plan and analyzers used in each test.

Test No.	Test Description	5-gas	Combustion	FTIR	Fuel GC	Dilution Tunnel / PM
1	Baseline diesel, ISO 8178 Cycle D2 5-mode	√	√	√	•	√
2	Baseline dual fuel, standard tuning, ISO 8178 D2 5-mode	√	√	√	√	√
3	Diesel displacement sweep, ISO 8178 Cycle D2 5-mode	√	√	•	•	•

Table-4. Stock configuration diesel displacement map for the dual fuel kit.

Load	**12%**	**25%**	**50%**	**75%**	**100%**
Diesel displacement baseline (%)	**35.0**	**59.6**	**70.0**	**69.4**	**58.7**
Natural gas equivalence ratio	0.15	0.23	0.22	0.27	0.24

$$\text{Diesel Displacement (\%)} = 100 \times \frac{\dot{m}_{\text{Baseline Diesel}} - \dot{m}_{\text{Diesel in Dual Fuel Mode}}}{\dot{m}_{\text{Baseline Diesel}}}$$

Equation-1.

Table-4 also shows the equivalence ratio for the natural gas/air mixture entering the cylinder prior to diesel injection. 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

VOL. 10, NO 20, NOVEMBER, 2015 ISSN 1819-6608
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fuel consumption and the air/fuel ratios the air flow through the engine was computed. The volumetric efficiency was determined from the air flow and intake air properties. The volumetric efficiency values for diesel operation at each load were utilized to compute intake flow for dual fuel operation. Finally, the intake flow and measured natural gas flow rates were used to compute natural gas equivalence ratios for dual fuel operation.

Table-5 gives some of the emission standards for non-road and stationary CI engines that were established by EPA. Tier 4 requires more than 90% reduction in NO_x emission. Tier 4 also requires substantial reduction in HC emission as well as PM emission. In natural gas engine, non-methane hydrocarbon (NMHC) is considered as THC emission. CO emission limits are unchanged since Tier 2 and 3. Looking at the data below, the standards are significantly tightened over time. Therefore, it is crucial for the engine manufacturers to continuously improve the engine efficiency and technology in order to meet the regulations.

Table-5. Emission standard for non-road and stationary CI engines with power more than 900 kW.

	Tier 1	Tier 2	Tier 3	Tier 4
Year released	**2004**	**2006**	**2011**	**2015**
NO_xLimit (g/kWh)	9.2	Sum 6.4	0.67	0.67
HC (NMHC) Limit (g/kWh)	1.3		0.4	0.19
CO Limit (g/kWh)	11.4	3.5	3.5	3.5
PM Limit (g/kWh)	0.54	0.2	0.1	0.03

Effects of Varying Diesel Displacement

Effects of diesel displacement variation were considered in order to reduce the emissions. A diesel displacement sweep was conducted at each of the five loads to find limits of operation and formulate an optimized substitution scheme that will maximize diesel displacement. In this particular testing, natural gas flow rate was varied until engine instability or extremely high emissions levels were observed.

RESULTS AND DISCUSSIONS

Diesel and Dual Fuel Baseline Emission Analysis

NO_xEmissions

NO_x is the sum of nitrogen monoxides (NO) and nitrogen dioxides (NO_2). There are other oxides of nitrogen sometimes included, but they are insignificant in diesel and natural gas engines. Figure-2 shows measured NO_x emissions at five loads for the two operating modes. The figure indicates that dual fuel operation emits less NO_x throughout the load map. NO_x 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 NO_x due to higher combustion temperatures. Both operating modes display minimum NO_x 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. At high loads the overall air/fuel ratio decreases (equivalence ratio increases) due to turbocharger limitations. For example, in the dual fuel case the overall equivalence ratio, considering both fuels, is 0.55 at 68 kW and 0.71 at 135 kW. This effect impacts both operating modes by increasing average combustion temperatures at high loads, tending to increase NO_x.

PM Emissions

Figure-3 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.

CO Emissions

The relationship between CO emissions and load isshownin Figure-4 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

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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 (about 70% reduction) 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.

THC Emissions

Figure-5 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 100X 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-4, the natural gas equivalence ratio varies from 0.15 to 0.27. At equivalence ratios this low, 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. At low loads the natural gas equivalence ratio and diesel jet penetrations are the lowest, likely contributing to high THC emissions at low load. The high THC emissions for dual fuel are therefore a product of incomplete combustion, primarily, of the lean air and natural gas mixture.

Regulated Emissions for Baseline Testing

Figure-6 shows ISO 8178 weighted emissions for diesel and dual fuel baseline operation. The John Deere Tier II 6068H CI engine is required to meet the US EPA limits at sea level conditions. These limits are indicated in the Figure. NO_x is combined with NMHCs. The NMHC emissions are evaluated by subtracting methane measured with the FTIR from HC. In normal diesel mode the engine meets the Tier II limits with the exception of NO_x + NMHC, which is slightly above the limit. The test was carried out at an altitude of 5000 feet. The turbocharger system does not provide as much boost at high altitude, which decreases the air fuel ratio and increases NO_x. This is most likely why NO_x + NMHC is slightly above the limit. In dual fuel mode NO_x + NMHC and CO limits are significantly exceeded. PM emissions are reduced further below the Tier II limit. The limit for NO_x + NMHC is exceeded because, although NO_x is reduced, the increase in NMHC emissions is much larger.

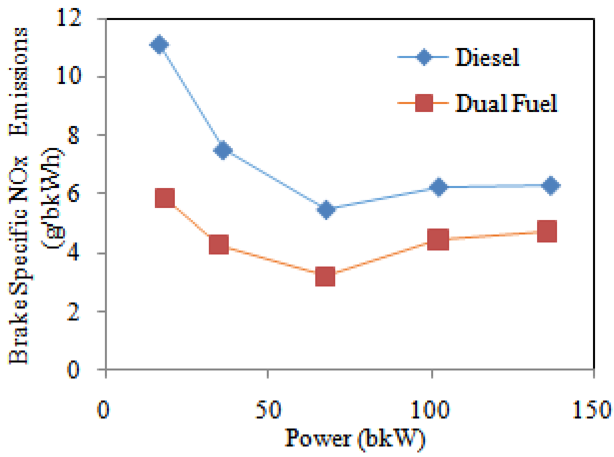

Figure-2. NO_x formation comparison for diesel anddualfuel baseline operation.

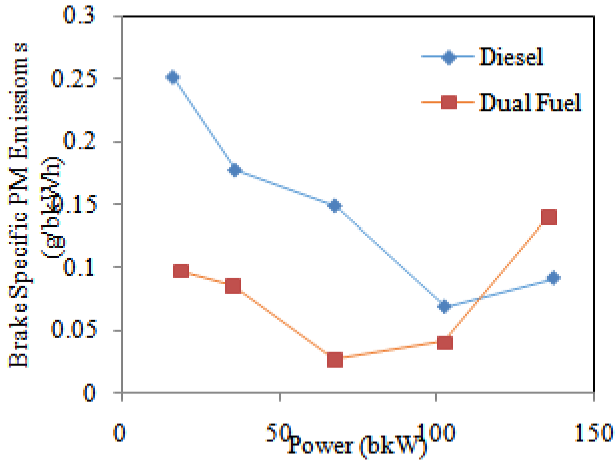

Figure-3. PM formation comparison for diesel anddualfuel baselinesoperation.

ISSN 1819-6608
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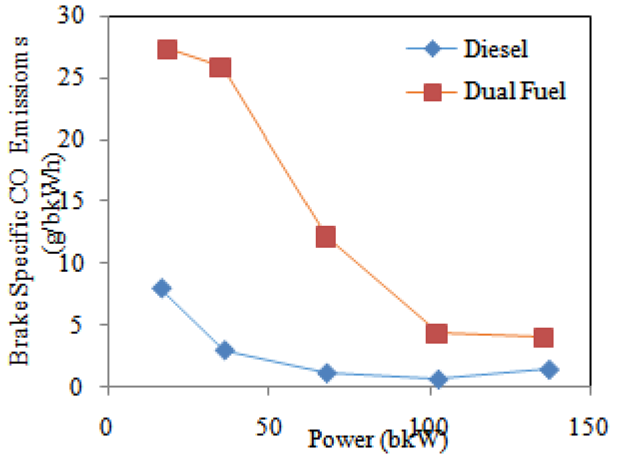

Figure-4. CO formation comparison for diesel anddualfuel baselinesoperation.

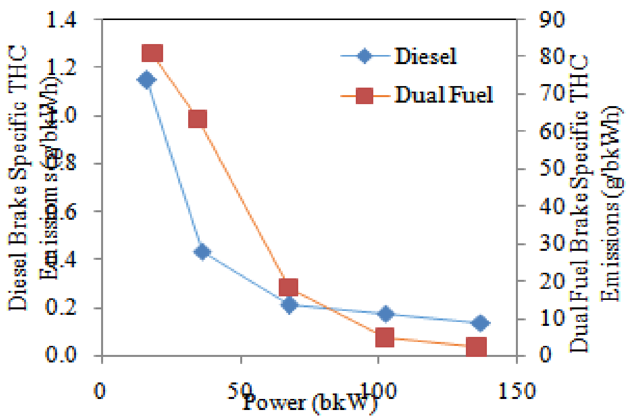

Figure-5. THC formation comparison for diesel anddualfuel baselines operation.

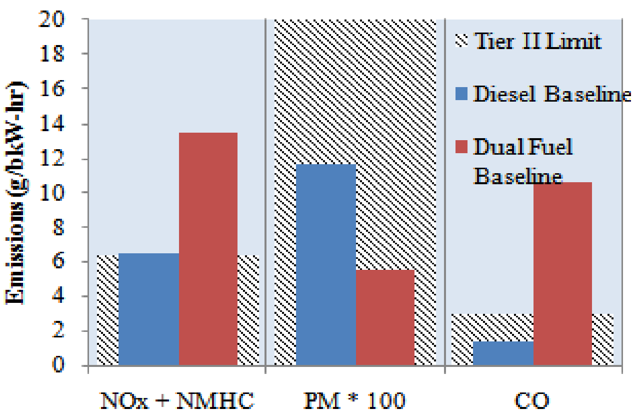

Figure-6. ISO 8178 weighted emissions compared to EPA Tier II limits.

The Effects of Diesel Displacement

Low and intermediate loads in dual fuel operation suffer from high CO and incredibly high THC emission levels. In order to optimize diesel displacement over the load map, a natural gas substitution sweep was conducted at each load in order to find the limits of operation and formulate an optimized substitution scheme that would maximize diesel displacement while reducing emissions. Figure-7 and Figure-8 shows brake specific variations at low loads. These figures show that THC and CO emissions increase with increasing diesel displacement at low loads. NO_x displays a slight decreasing trend with increasing diesel displacement. At 25% load the THC emissions are reduced for the first diesel displacement point (~17.5%), but then increases for further increases in diesel displacement. At 50% load each emission species displays a different trend.

Figure-9 shows the brake specific emissions trend with diesel displacement at intermediate load. THC emissions start below 1 g/bkW-hr at 0% diesel displacement and increases to about 25 g/bkW-hr at maximum diesel displacement. CO emissions gradually increase as diesel displacement increases, peaking near 60% diesel displacement, then significantly decrease as diesel displacement increases further. NO_x emissions display a gradual decreasing trend as diesel displacement increases. As diesel displacement increases from 0% to maximum diesel displacement NO_x emission are approximately cut in half.

Figure-10 shows the emissions trend when the amount of diesel replaced with natural gas is varied at 75% load. NO_x emissions gradually decrease as diesel is replaced with natural gas until about 40% replacement, at which point NO_x increases. In an opposite fashion, emissions of CO significantly increase with increasing diesel replacement up to about 40%, where the trend reverses. Natural gas equivalence ratio is shown earlier in Table-4. The equivalence ratio steadily increases as diesel replacement is increased. At an equivalence ratio between 0.16 and 0.21 the CO and NO_x trends are reversed. THC emissions increase with increasing diesel replacement throughout the entire sweep. As the mass of diesel injected decreases, it is likely that more regions in the cylinder are not ignited by the diesel jets. At high diesel replacement values above 60%, the CO trend reverses again and CO increases.

Similar trends are noted for the 100% load point, shown in Figure-11. However, there appears to be an outlier at 22% diesel displacement. Extremely high THC emissions are noted at this point. It is unclear why there would be such a large increase at this condition. However, a corresponding increase in CO indicates that the increase may be a real effect since CO and THC often trend in the same direction. NO_x emissions gradually decrease with increasing diesel displacement, reaching a minimum at 40% diesel displacement, followed by a significant increase in NO_x continuing to the maximum diesel displacement.

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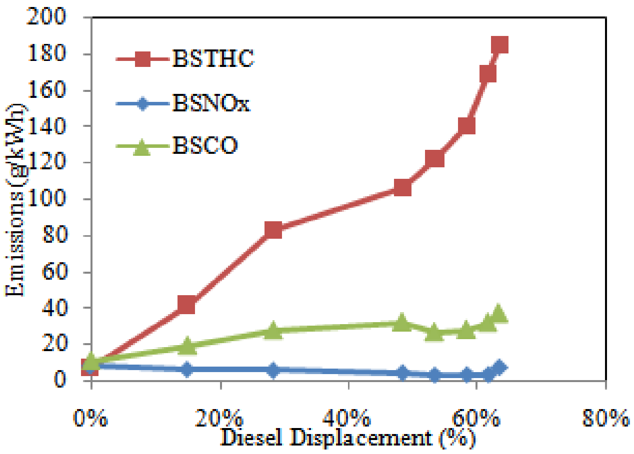

Figure-7. Emissions as a function of diesel displacement at 12% load.

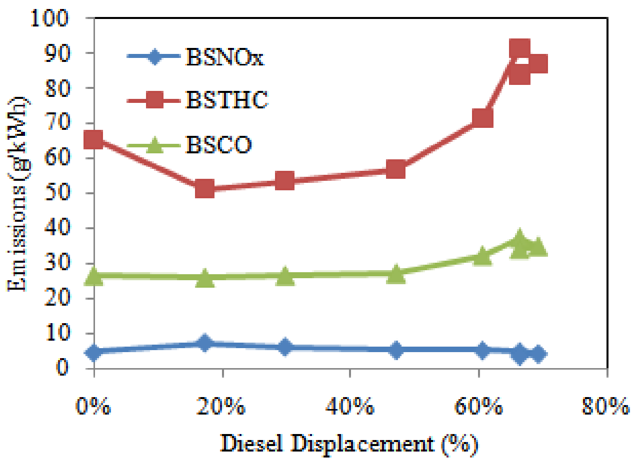

Figure-8. Emissions as a function of diesel displacement at 25% load.

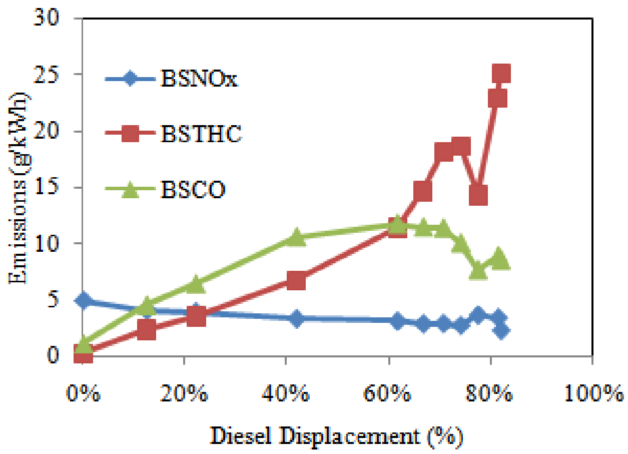

Figure-9. Emissions as a function of diesel displacement at 50% load.

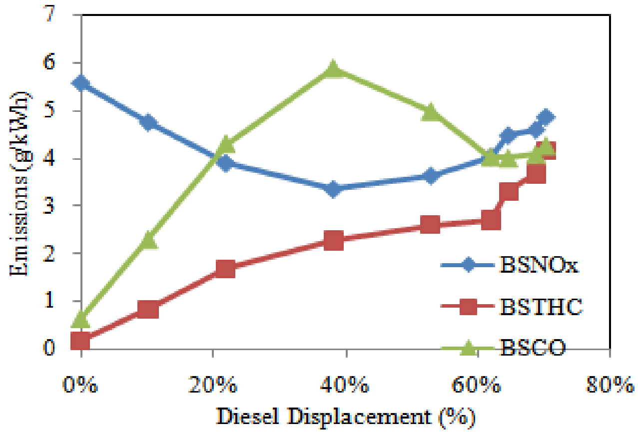

Figure-10. Emissions as a function of diesel displacement at 75% load.

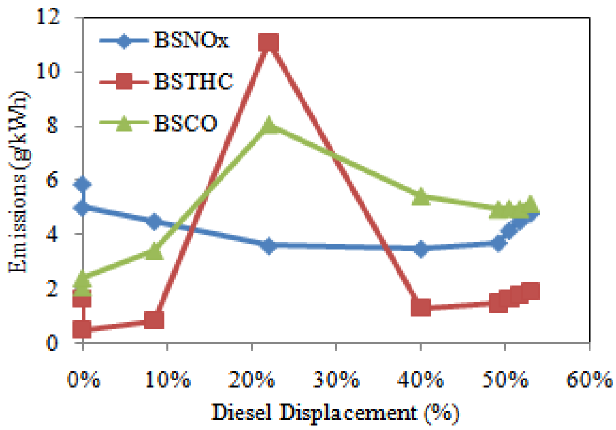

Figure-11. Emissions as a function of diesel displacement at 100% load.

Optimization Testing

The diesel displacement sweep performed indicated the major factor influencing the decision of the optimized testing parameters:

a) At low loads, the dual fuel system provides more gas than can be utilized, resulting in very high THC and CO emissions.

Taking this factor into account, it was decided that the following load map as shown in Table-6 would represent an optimized profile. Because of the high THC and CO emissions at low loads, it was decided that dual fuel would be turned off until intermediate loads were reached. Subsequent target diesel displacements were determined by selecting the highest diesel displacement observed during the natural gas substitution sweeps which maintained engine stability. Figure-12 shows emissions levels in diesel and dual fuel baselines, optimized dual fuel, and Tier II limit. Although emissions with optimized substitution map are still higher than limits, it shows an improvement when compared with dual fuel baseline emission.

VOL. 10, NO 20, NOVEMBER, 2015 ISSN 1819-6608
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Table-6. Dual fuel baseline and optimized loading map.

Load	Dual Fuel Baseline Diesel Displacement	Target (Optimized) Diesel Displacement
12%	35%	0%
25%	60%	0%
50%	70%	76%
75%	70%	66%
100%	59%	52%

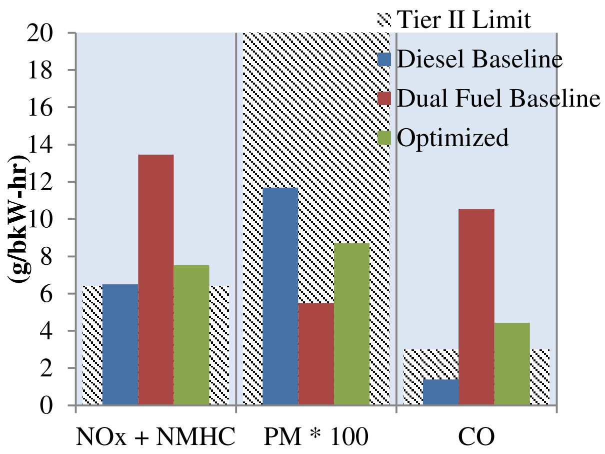

Figure-12. EPA regulated emissions for the diesel and dual fuel baselines and the optimized dual fuel map.

CONCLUSIONS ANDRECOMENDATION

An experimental evaluation of the emission characteristics, fuel cost savings through diesel displacement of a natural gas-diesel dual fuel engine have been performed. This research provides solution for a high NO_x and PM emission associated with diesel engine as well as addresses the key challenges related to natural gas-diesel dual fuel engines.

Dual fuel engines reduce NO_x and PM emission compared to diesel engines. Natural gas combustion with methane as a lowest member in paraffin family is expected to produce less PM. In addition, the premixed combustion in dual fuel engine reduces the formation of locally rich mixtures. Therefore, with lean conditions, the flame temperature is lower, thus reduces the NO_x formation in dual fuel engine. EPA regulated NO_x + NMHC and CO emissions are exceeded for dual fuel operation.

In order to optimize emission, a diesel displacement sweep was conducted. It is suggested to turn off dual fuel at low loads and increasing the diesel displacement at intermediate and high loads. To reduce and control further the CO and HC emissions in dual fuel engine, the use of oxidation catalyst is recommended. Engine testing incorporating the use of oxidation catalyst would allow further optimization of the dual fuel system.

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