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INFLUENCE OF INJECTION TIMING ON THE PERFORMANCE AND EMISSIONS OF A DUAL FUEL ENGINE

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ABSTRACT
In this work, a 6.8L John Deere 6068H compression ignition engine is coupled with an aftermarket dual fuel kit and converted to dual fuel engine. This dual fuel kit enables the diesel engine to run on diesel and gaseous fuels. The engine is a six-cylinder turbocharged after cooled four-stroke cycle diesel engine. The engine has a maximum power output of 205kW at 2400rpm and the compression ratio is 17:1. The experiment is conducted to investigate the effect of injection timing variations on the combustion and emissions in dual fuel engine. The dual fuel engine is tested through five different load points while the speed is maintained constant at 1800rpm. The gaseous fuel used is natural gas and the performance characteristics are compared with baseline dual fuel engine with nominal injection timing. The dual fuel engine has a nominal injection timing of 6.5° before top dead center (BTDC) at 12, 25, 50 and 75% loads and 4.8° BTDC at 100% load. The injection timings are then varied between 0° to 16° BTDC. Data for combustion duration, ignition delay, peak pressure, combustion stability, nitrogen oxides (NOx), carbon monoxides (CO) and total hydrocarbon emissions are presented. Results showed that by advancing the timing away from the nominal start of injection (SOI) timing, useful power is increased over 70%. The combustion stability is also increased with these advancements. Whereas, the ignition delay and flame propagation does not show any improvement as the ignition starts or earlier than the nominal timing. Hydrocarbons and CO emissions are reduced over 30% at advanced SOI at intermediate BMEP, however NOx emission increases at advanced injection timings with over 30% increment at all BMEPs.

Keywords: diesel engine, dual fuel engine, emissions, effect of injection timing.

INTRODUCTION
Diesel engines offer a variety of advantages. It is much more efficient and preferable as compared with spark ignition engine due to lower maintenance cost, much more rugged, reliable and efficient. Therefore, they are frequently used in heavy duty equipment, transportations and various industrial applications. Despite these advantages, diesel generated more nitrogen oxides (NOx) and particulate matters (PM) [1-4]. NOx is hazardous to health because it is one of the main contributors to the formation of photochemical smog which is created by the reaction of emissions from automobiles and atmospheric air with the presence of light. On the other hand, PM is generated within the cylinder in the fuel-rich zones combustion. These two pollutants are traded against each other in many aspects of engine design and control of operating conditions. Extending the combustion time can reduce the PM since soot particles will have enough time to be mixed with oxygen and become carbon dioxide (CO_2); however it will create more NOx because the cylinder will have a high temperature. Therefore, reducing PM to minimal level solely by engine design and operating conditions control cannot be done [5].

To overcome issues with NOx, many have proposed exhaust gas recirculating (EGR) system [6-10]. EGR system recirculates the residual gases from the exhaust back to the intake manifold. These recycled gases act as an absorbent of combustion heat which helps to reduce high cylinder temperature. This system works solely on diesel engine with diesel fuel or combined with various biodiesel available on the market. This technology however increases PM and unburned hydrocarbon by at least 40% [11].

In order to minimize the formation of NOx and PM concurrently in diesel engines, a homogenous charge compression ignition (HCCI) engine is proposed. HCCI engine works by compressing a homogeneous mixture of fuel and oxidizer with the absence of fuel injector as in diesel engine or spark plug as in spark ignition engine. The combustion of HCCI appears in multiple locations instantaneously. This technique is reported to produce extremely low levels of NOx because it reduces the cylinder charge temperature. This technique also reduces the locally fuel rich regions in the cylinder [12, 13]. Since HCCI engines are fuel-lean, brake thermal efficiency also increases. HCCI is able to run solely on diesel or gasoline or combined with various fuels such as biogas, biodiesel dimethyl ester, natural gas and other gaseous fuels [14-16]. Apart from that, knocking, high unburned hydrocarbons and carbon monoxides (CO) emissions are also reported when running with HCCI.

There have been increasing demands for a diesel engine with no major modifications to run with gaseous fuels in order to reduce down the NOx and PM. This engine is called dual fuel engine. This dual fuel engine requires a source of ignition unlike HCCI to start combustion within the mixture of air and fuel inside the chamber. The engine can be run fully on fuel oil or oil-gas operation. Extended diesel substitutions with natural gas have been reported at 78% substitution combined with lowering the manifold temperature [17]. This engine however is observed to have elevated CO and hydrocarbons.

One way to overcome the issues with high CO and hydrocarbons in dual fuel engine is to vary the injection timing. Previous studies with algae oil blend showed that by injecting the fuel as earlier as 27° BTDC can lead to almost 6% improvement in brake thermal efficiency [18]. Extensive reduction of CO by 80% is also presented. Similarly, hydrocarbon emission is reduced by 30% at high load in dual fuel diesel engine with the algae oil blend. However, investigation of such engine with biodiesel shows that no significant reductions in CO and hydrocarbons with retarding or advancing the injection timing. The lowest point is observed at the original start of injection (SOI) timing [19].

The objective of this study is to determine the effect of various fuel injection timings on combustion and emissions of a dual fuel engine with natural gas as a supplement fuel. At each load, timings were both advanced and retarded around the original injection timing until engine instability which demonstrated by the coefficient of variance (COV) values greater than 8 or extremely high emissions levels were noted.

EXPERIMENTAL SETUP AND TEST PLAN

Experimental setup

The experiment is performed on a turbo-charged 6-cylinder, 6.8L direct injection compression ignition engine by John Deere. The model of the John Deere is 6068H industrial diesel engine which utilizes air-to-air after cooling system. This method allows an engine to comply with emission regulations with improved performance and engine efficiency. The engine specifications are listed in Table-1. The engine is equipped with a high pressure common rail fuel system. It provides variable common-rail pressure at higher injection pressure at multiple point injections. The engine control unit (ECU) system of the engine provides data of the fuel injection timing which controls the start and end of the engine's fuel supply. The engine schematic diagram and the photograph of the engine setup are presented in Figure-1. The engine is loaded with an AC motoring dynamometer with a maximum power of 300hp at a maximum speed of 2800rpm. The dynamometer is manufactured by Marathon Electric. It is controlled by a variable frequency drive by Eaton Corporation that has a rating of 350hp at 480 VAC. The real-time engine speed, torque, power and emissions were monitored using a LabView Virtual Instrument VI which is programmed to remotely control the system. The diesel fuel consumption rates are measured by AVL flow meter with an accuracy of 0.12% and also calculated using DevX software provided by John Deere.

Emissions measurements from exhaust were taken straight from exhaust tailpipe. The 5-gas analyzers which comprised of Rosemount instruments are used to measure CO, CO2 and NOx concentration while THC and O2 are measured with Siemens instruments. PM emissions were measured using a dilution tunnel. The dilution ratio used is 15:1 and remained constant throughout testing. This ratio is used to convert from diluted measurement to actual emissions output. A Kistler 6056A piezoelectric pressure transducer is employed to investigate the in-cylinder pressure. The measurements were recorded with a resolution of 0.25° crank angle (CA) at every 1000 cycles for engine performance investigation.

For natural gas flow rate measurement, a flowing fit was created from Omega FMA-1700 series thermal mass flow meter data to correlate throttle valve position to natural gas flow rate. From this fit, the natural gas flow rate was computed from the gas throttle valve position and temperatures of the gas supply. The flow rate of natural gas is calculated using Equation 1.

$$\frac{d_m}{d_t} = K\ x\ C_v x \sqrt{SG}$$

Equation 1

where dm/dt is the natural gas mass flow rate, K is the constant as a function of load, Cv is the coefficient as a function of plate angle and SG is the specific gravity of gas.

The data was taken at steady state condition after the engine was warmed up between one hour and half to two hours until the intake manifold temperature and block coolant reached 43°C and 88°C under conventional diesel operation mode.

The gaseous fuel used is natural gas supplied by the Fort Collins, CO city distribution system. The composition of the distributed natural gas for this experiment varies but the main component is methane and relatively low of ethane and other hydrocarbons. It also contains much smaller traces of nitrogen, carbon dioxide and water vapor. The natural gas composition as measured during the day is shown in Table-2.

Experiment test plan

Tests were conducted for the diesel with the dual fuel system for normal dual fuel operation at nominal injection timing. Second sets of data were taken at various injection timing as shown in Table 3. The injection timing is the time at which fuel is sprayed into the cylinder and is expressed in CA degree. The ECU sends an electronic trigger to diesel fuel injectors to provide the right amount of fuel. For this study, the injection timing was varied between 1° to 16° BTDC, while other parameters remain unchanged. In this work, advanced injection timing is characterized as the number of degrees before nominal timing, while retarded timing is defined as the timing that fuel ignition takes place later than nominal timing. In dual fuel baseline, the nominal injection timing is set by the engine control module (ECM) based on the diesel fuel amount injected. The timing map used in this study is shown in Table-3. These nominal timings are marked as Δ in all results. Data at five different engine loads were taken corresponding to 12%, 25%, 50%, 75% and 100% of the

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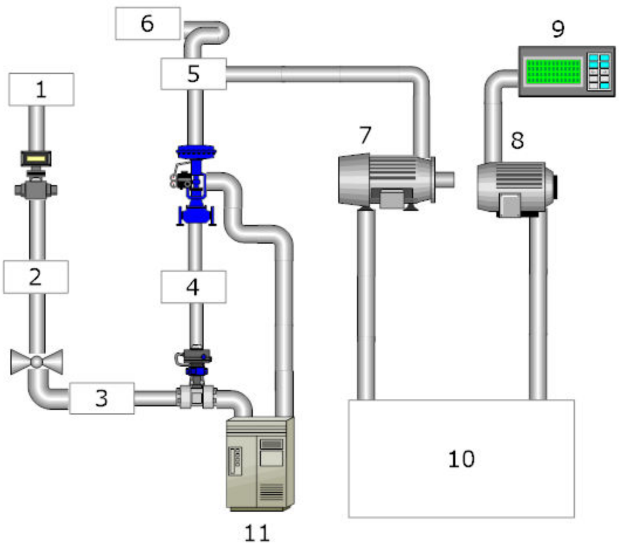
165 kW maximum load achievable on site at 1800 rpm. These loads correspond to BMEP = 0.182, 0.346, 0.669, 1.012 and 1.342MPa, respectively. The engine was maintained constant at the speed of 1800 rpm for all cases. These testing parameters were determined after consulting ISO Standard 8178:4 Cycle D2.

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
Nominal start of injection timing	6.5° BTDC
Rated power	205 kW (275 hp)
Rated speed	2400 rpm
Inlet valve closure	-156.75°
Bore	106 mm
Stroke	127 mm
Connecting Rod	203 mm
Compression Ratio	17:1

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

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



Item	Description
1	Natural gas supply
2	Zero pressure regulator
3	Filter
4	Zero pressure regulator
5	Mixer
6	Intake air
7	Compressor
8	Turbine
9	Gas analyzer/dilution tunnel
10	John Deere diesel engine
11	Dual fuel controller

Figure-1. Schematic diagram of experimental setup.

Table-3. Timing map used in this study.

BMEP (MPA)	ADVANCE (° BTDC)	NOMINAL (° BTDC)	RETARD (° BTDC)
0.182	16, 15, 14, 12, 10, 8	6.5	6, 4, 2
0.346	15, 13, 11, 9, 7	6.5	5, 3, 1
0.669	11, 10, 9, 8, 7	6.5	5, 3, 1
1.012	11, 10, 9, 8, 7	6.5	5, 3, 1
1.342	9, 8, 7, 6, 5	4.8	4, 3

RESULTS AND DISCUSSIONS

Flame development

Quality of the in-cylinder combustion represents the engine performance; therefore it is very important to improving in-cylinder combustion. To obtain information about the combustion development inside the cylinder, flame development period 0-10% and 10-90% flame propagation period in CA degree are presented using Equation 2 & 3. The speed of the initial phase of combustion or the flame development is represented by the 0-10% burn duration, often referred to as the ignition delay period. The 0-10% mass burn duration is characterized as the crank angle intervals between SOI timing and 10% of mass fraction burned. On the other hand, the 10-90% burn duration or rapid burn angle is defined as the major heat release process. It is the period between the end of flame development and flame propagation [20].

0-10% Burn Duration=SOI (°BTDC) + mass fraction burned at 10% (CA)

Equation 2

10-90%= 90% mass fraction burned location (CA)–10% mass fraction burned location (CA))

Equation 3

The 0-10% and 10-90% burn durations for different BMEPs are shown in Figures 2-6. The general trends for the 0-10% burn durations are decreasing at nominal SOI, while increasing as SOI are retarding and advancing at all BMEPs. There is no significant ignition delay is observed for dual fuel at advanced/retarded SOI as shown by the 0-10% burn duration. The 10-90% burn duration trends peak at nominal SOI, except for BMEP=0.182MPa. The general trend for all loads is decreasing as the SOI is advanced indicating faster combustion rates at advanced timing. At retarded SOI, less time is acquired per CA degree for combustion. The data show that at all loads except lowest load, dual fuel engine at nominal SOI experiences longer flame propagation which indicates slower combustion rate as compared to advanced/retarded SOI. Advanced fuel injection timings have a significant influence on the distributions of liquid fuel in the cylinder. It also greatly enhances the turbulent intensity. Consequently, early flame development and flame propagation period are affected at which indicated by a shorter combustion process and better HC and CO emission (Figures 9-12). However, there is an increment in NOx level with better combustion process due to increases the cylinder temperature.

Peak pressure

Another important parameter that indicates the in-cylinder combustion performance is peak pressure. The variations of peak pressure with injection timings under dual fuel operation are shown in Figure-7. The data show that no significant improvement as SOI is advanced, except at higher BMEPs show at advanced timing, maximum pressure is achieved. This indicates that as the fuel is injected earlier a number of combustion of the premixed mixture is involved in the initial combustion causing the peak pressure to increase. Results show over 70% increment of peak pressure when the timing is advanced at higher BMEPs.

COV of IMEP

Variation in combustion from cycle-to-cycle and cylinder-to-cylinder is determined using the coefficient of variance in indicated mean effective pressure (COV of IMEP). The COV is the standard deviation of the data of its mean value as shown in Equation 4. It is expressed as a percentage. If the value of the COV is below than 10% the normal automotive maneuverability is guaranteed [21]. The limit value of COV of IMEP determines the limit of mixture leaning. This cyclic variation also indicates the variations of flame development during the initial phase of combustion.

$$COV_{IMEP} = 100\frac{\sigma_{IMEP}}{IMEP}$$

Equation 4

where σ is the standard deviation. Figures 2-6 show the COV of IMEP at various SOI and BMEPs. At lower BMEPs (BMEP=0.182 and 0.346MPa), the trend is increasing beyond 10% as the SOI is advanced. This data show combustion instability when the timing is advanced further from nominal. As the BMEP increases, combustion benefits from the advancement of the timings. When SOI is advanced further from TDC, the combustion becomes more stable with COV of IMEP below 4% for BMEP=0.669MPa, 3% for BMEP=1.012MPa and 1% for BMEP=1.342MPa compared to nominal SOI timing.

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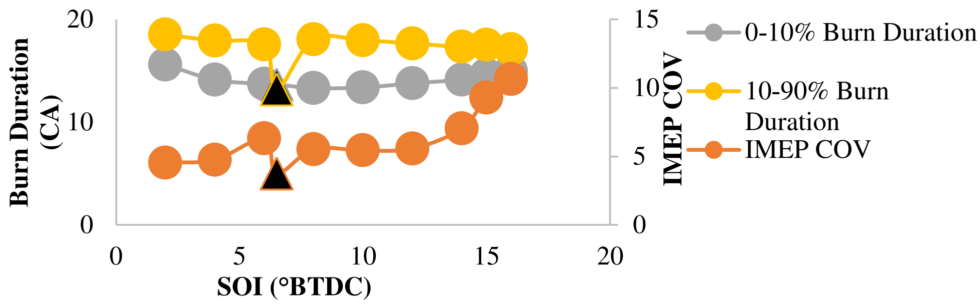

Figure-2. Burn durations and IMEP COV at various SOI at BMEP = 0.182MPa.

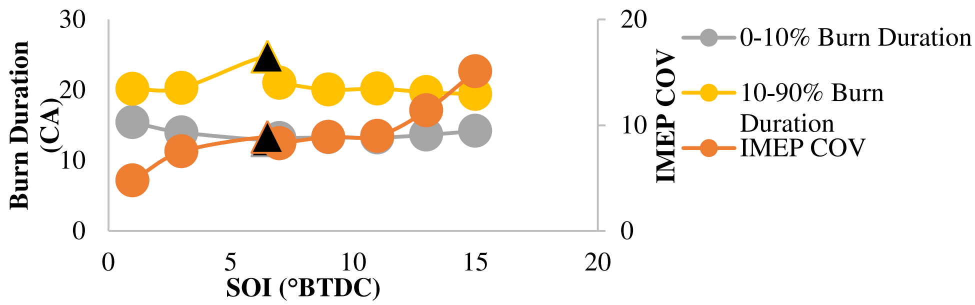

Figure-3. Burn durations and IMEP COV at various SOI at BMEP = 0.346MPa.

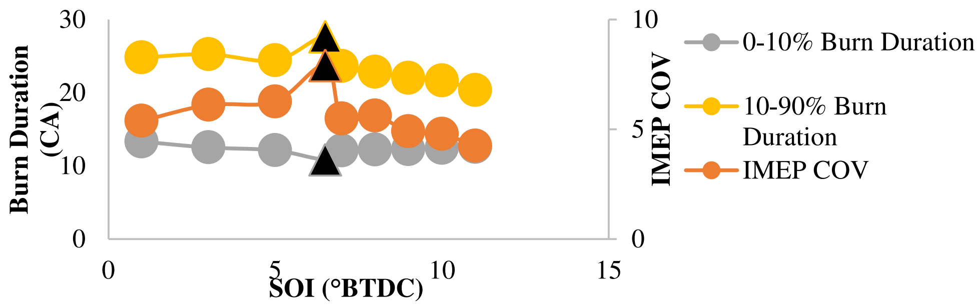

Figure-4. Burn durations and IMEP COV at various SOI at BMEP = 0.669MPa.

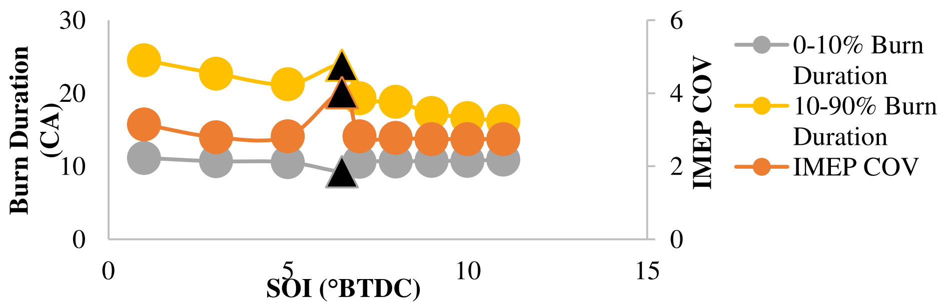

Figure-5. Burn durations and IMEP COV at various SOI at BMEP = 1.012MPa.

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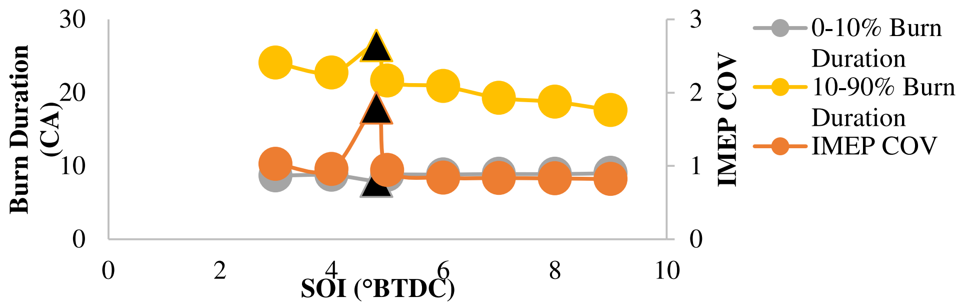

Figure-6. Burn durations and IMEP COV at various SOI at BMEP = 1.342MPa.

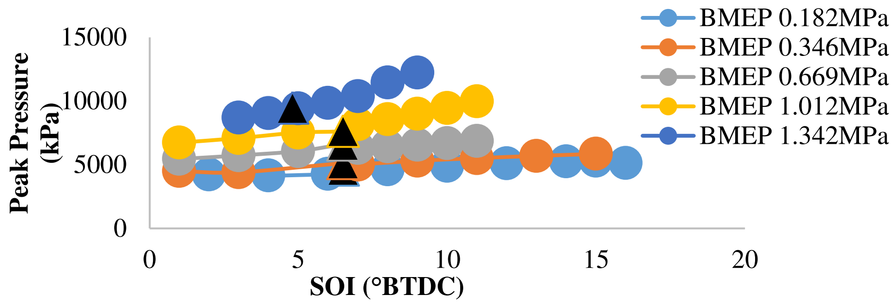

Figure-7. Peak pressure at various SOI.

Exhaust emissions

Figure-8 presents emission variations as injection timing is varied at BMEP = 0.182MPa. This shows no significant changes as the timing is advanced or retarded except at 15°BTDC. It should be noted that at this load, it was occasionally difficult to maintain the target engine load of ~15 kW, and even a 5 kW deviation from the target represents a 30% change in load. This uncertainty is a potential factor in the variation of the brake-specific emissions at these loads. It shows that NOx emissions trend is increasing with advances in timing. As the injection is advanced, more time as allotted for the production of NOx and peak pressure occurs earlier, producing higher peak pressure and temperature. HC decreases as due to higher combustion temperature and more time for combustion to occur before expansion. At nominal timing, all emissions are at the lowest point.

Figures 9-11 illustrates emission variations as a function of injection timing at BMEP=0.346, 0.669 and 1.012MPa. At these loads, HC and CO trend similarly, while NOx trends in the opposite direction. For BMEP=0.346, 0.669 and 1.012MPa, the ideal trade-off between minimizing both HC and CO occurred at SOI=11, 9 and 7 °BTDC, respectively. At this timing, HC is reduced over 20%, 50% and 30% if were to be advanced earlier than nominal timing. As the combustion time increases with advanced timing, more time is provided for the mixture to be burned. Figure-12 shows brake specific emission variations when injection timing is varied at 100% load (BMEP=1.342MPa). It shows at high load all three emissions are on the same order of magnitude. CO emissions overtake HC emissions at high load, whereas previously HC emissions were larger in magnitude. At this load, it shows a slight improvement in the emissions at SOI=5°BTDC. It is expected that at more advanced injection timing, the dual fuel engine will suffer from NOx emissions since higher cylinder temperature is acquired. For all loads, it was observed that the ideal trade-off between minimizing both HC and NOx emissions occurred at or very near the stock SOI.

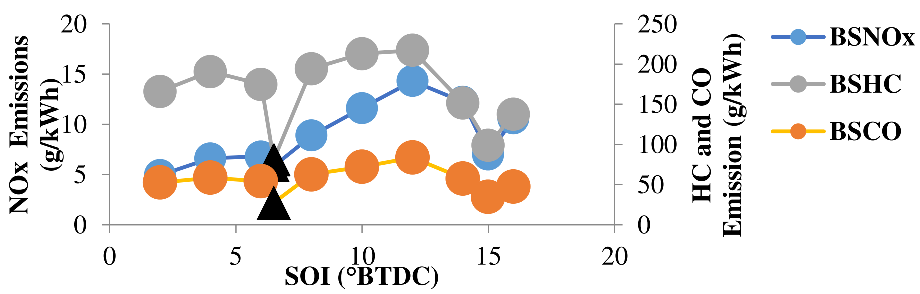

Figure-8. Emissions as a function of injection timings at BMEP = 0.182MPa.

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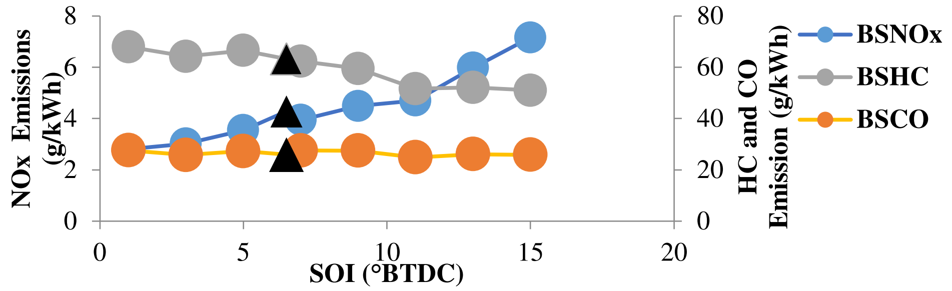

Figure-9. Emissions as a function of injection timings at BMEP = 0.346MPa.

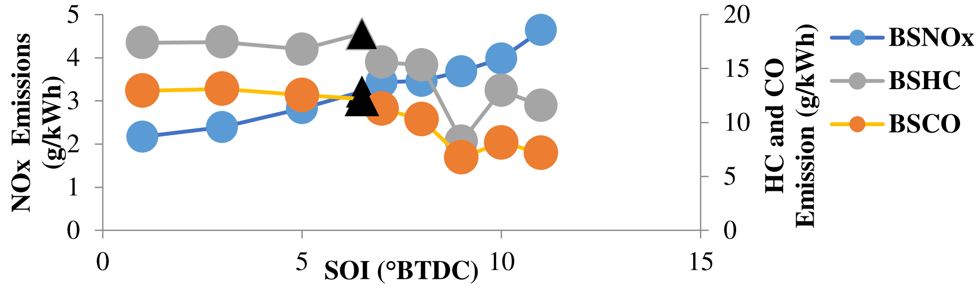

Figure-10. Emissions as a function of injection timings at BMEP = 0.669MPa.

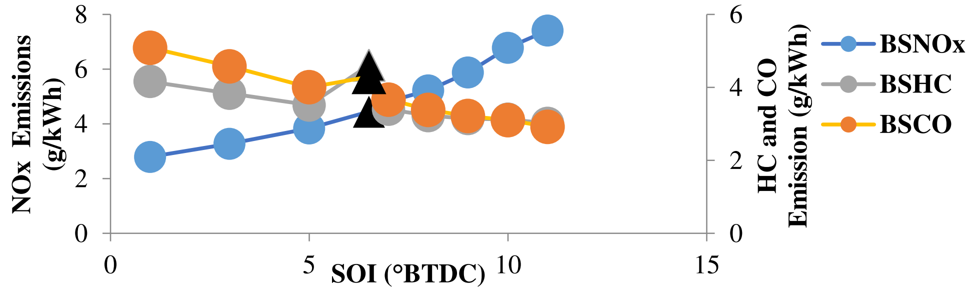

Figure-11. Emissions as a function of injection timings at BMEP = 1.012MPa.

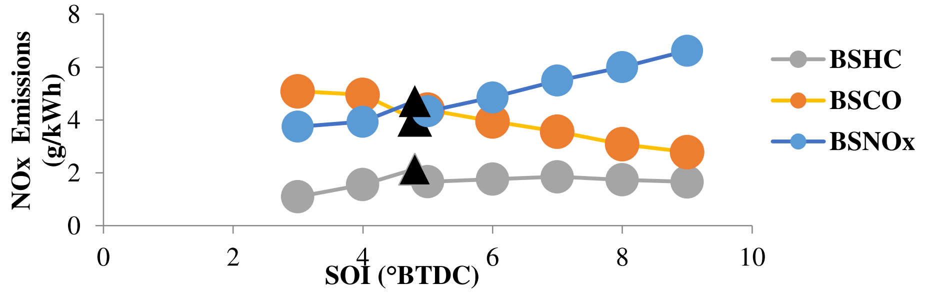

Figure-12. Emissions as a function of injection timings at BMEP = 1.342MPa.

CONCLUSIONS

Effects of different SOI timings on John Deere compression ignition engine with natural gas have been investigated at an engine speed of 1800 rpm. The combustion performance as indicated by the cylinder peak pressure, ignition delay, flame development interval and combustion duration, as well as the exhaust emissions have been presented. The ignition delay, flame development duration, as well as combustion duration are also calculated. Conclusions from the experimental work comparing dual fuel engine performance at nominal and various injection timings are given below:

a) No significant changes in ignition delays, represented by 0-10% burn duration angle. However, the delays are decreasing by about 30% at higher BMEPs.
b) The flame development represented by 10-90% burn duration trends peaks at nominal SOI, except for BMEP=0.182MPa. Faster combustion with an average of 20% is observed at higher BMEPs as the SOI is advanced further from the TDC.
c) Peak pressure increases almost 20% when the SOI is advanced from nominal SOI.
d) HC and CO emissions are reduced over 30% at advanced SOI at intermediate BMEP
e) NOx emission increases at advanced injection timings with over 30% increment at all BMEPs.

Therefore, it is concluded that SOI timing affects emissions production in dual fuel engine. Advanced injection away from TDC can reduce CO and HC emissions but increases NOx emissions. NOx emissions can be reduced only at retarded fuel injection timing conditions in some cases. To reduce and control further the CO and HC emissions in dual fuel engine, the use of oxidation catalyst is recommended.

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