An experimental investigation on the combustion and emissions performance of a natural gas-diesel dual fuel engine at low and medium loads

Guo, Hongsheng; Neill, W. Stuart; Liko, Brian

Publisher's version / Version de l'éditeur:
https://doi.org/10.1115/ICEF2015-1041
Proceedings of the ASME 2015 Internal Combustion Engine Division Fall Technical Conference, 2015

Proceedings of the ASME 2015 Internal Combustion Engine Division Fall Technical Conference
ICEF2015
November 8-11, 2015, Houston, TX, USA

ICEF2015-1041

AN EXPERIMENTAL INVESTIGATION ON THE COMBUSTION AND EMISSIONS PERFORMANCE OF A NATURAL GAS – DIESEL DUAL FUEL ENGINE AT LOW AND MEDIUM LOADS

Hongsheng Guo, W. Stuart Neill, Brian Liko
Energy, Mining and Environment Portfolio, National Research Council Canada
1200 Montreal Road, Ottawa, Ontario, Canada K1A 0R6

ABSTRACT

Natural gas is an abundant and inexpensive fuel in North America. It produces lower greenhouse gas emissions than diesel fuel when burned in an internal combustion engine. It is also considered to be a clean fuel because it generates lower particulate matter emissions than diesel fuel during combustion.

In this study, an experimental study was conducted to investigate the combustion and emissions performance of a natural gas – diesel dual fuel engine at low and medium loads. A single cylinder direct injection diesel engine was modified to operate as the dual fuel engine. The diesel fuel was directly injected into the cylinder, while natural gas was injected into the intake port. The operating conditions, such as engine speed, load, intake temperature and pressure, were well controlled during the experiment. The effect of natural gas fraction on energy efficiency, cylinder pressure, exhaust temperature, and combustion stability were recorded and analyzed. The emissions data, including particulate matter, nitric oxides, carbon monoxide, and methane at various natural gas fractions and operating conditions were also analyzed. The results showed that natural gas – diesel dual fuel combustion slightly decreased brake thermal efficiency at low and medium load conditions and significantly reduced carbon dioxide and particulate matter emissions. Methane and NO_x emissions increased in dual fuel combustion mode compared to diesel operation. The variation of carbon monoxide emissions in dual fuel mode depended on load and speed conditions.

INTRODUCTION

Being a low carbon fuel, natural gas produces lower carbon dioxide emissions and generates lower particulate matter (PM) emissions than diesel fuel when burned in an internal combustion engine [1-3]. Since heavy duty engine industry predominately uses diesel fuel, replacing diesel fuel by natural gas in internal combustion engines will reduce emissions of carbon dioxide and particulate matter from heavy duty engine industry. Most heavy duty engines are compression ignition engines that usually offer higher efficiency than spark ignition engines. Although it is theoretically possible to burn natural gas using compression ignition engines, pure natural gas is generally not used in these engines due to its high auto-ignition temperature. Alternatively, natural gas – diesel dual fuel combustion is attracting more and more attention in the application of natural gas in heavy duty engine industry.

In natural gas – diesel dual fuel engines, while diesel fuel is always directly injected into the cylinder, natural gas can be either directly injected into cylinder or injected into the intake manifold. Although direct injection of natural gas maximizes natural gas use and maintains higher horsepower and efficiency, it still generates significant soot emissions due to the diffusion combustion process [4]. Port injection of natural gas generates relatively lower soot emissions due to the premixed natural gas combustion process. In addition, port injection technology requires less engine modifications to convert a conventional diesel engine. This paper discusses port injection of natural gas.

The fundamental combustion process in natural gas – diesel dual fuel engines was reviewed by Karim [5]. Hountalas et al. [6] and Papagiannakis et al. [7] investigated the combustion and emissions characteristics of natural gas – diesel dual fuel combustion in a conventional diesel engine using a single cylinder, four-stroke, port injection of natural gas and naturally aspirated engine. They found that energy efficiency decreased for dual fuel combustion, especially at low load. The diesel injection timing in [6,7] was fixed. During the studies of pilot-ignited dual fuel combustion, Alla et al [8], Singh et al. [2], Krishnan et al. [9] and Srinivasan et al. [10] found that a change in diesel injection timing had a significant effect on the combustion performance of dual fuel combustion. The combustion performance of dual fuel combustion could also be affected by intake conditions [2,9], exhaust gas recirculation [11], and cetane number of diesel fuel [12]. Tablan [13] recently investigated the effect of diesel substitution by natural gas on the performance of a commercially available heavy duty diesel engine, and showed that natural gas could substitute diesel fuel in a dual-fuel heavy-duty diesel engine across multiple speed and load operating points while maintaining thermal efficiency. More recently, advanced low temperature combustion concept has been applied to natural gas – diesel dual fuel combustion. Königsson [14] showed that diesel fuel injection split could help to improve the combustion stability and obtain relatively lower emissions. Walker et al. [15] investigated natural gas – diesel dual fuel combustion from the view point of reactivity controlled compression ignition (RCCI) combustion, and found that natural gas – diesel RCCI combustion could extend the load limit of RCCI combustion compared to gasoline - diesel dual fuel combustion.

The objective of this paper is to investigate the combustion and emissions performance of a natural gas – diesel dual fuel engine modified from a conventional diesel engine. The investigation focuses on low and medium loads with the emphasis on energy efficiency and emissions characteristics when natural gas substitution rate gradually increases. The engine setup and experimental procedure are described first, followed by results and discussion. Finally, concluding remarks are provided.

ENGINE SETUP AND EXPERIMENTAL PROCEDURE

Engine Setup

The engine used in the investigation is a modified single-cylinder version of Caterpillar's 3400-series heavy-duty engine. The schematic of the experimental setup is shown in Fig. 1. The basic engine configuration is given in Table 1.

The original engine was modified to suit for natural gas – diesel dual fuel combustion. Natural gas was injected to the intake port by a fuel injection manifold that includes eight gas fuel injectors manufactured by Alternative Fuel Systems Inc. The number of injector needed, start of natural gas injection, and injection pulse width were controlled by a driven system provide by National Instruments (model PXI-1031chassis, 8184 embedded controller, and 7813 R RIO card connected to cRIO-9151 expansion chassis) and LabVIEW-based software (Drivven Inc., Stand-Alone Direct Injector Drive System). A natural gas chamber was installed before the natural gas injection manifold to reduce the pressure pulsation of natural gas. The original mechanically-actuated diesel injector was replaced by a prototype common rail fuel injector system to deliver diesel fuel directly into cylinder. The fuel rail pressure, start of diesel fuel injection and diesel injection pulse width were also controlled by the Drivven system and software that were used for gas injectors.

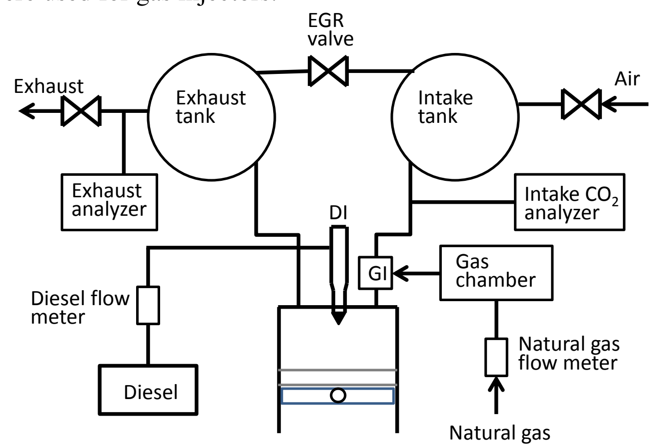

Fig. 1 Schematic of experimental setup.

Table 1 Basic engine parameters.

Parameter	Value
Basic engine model	Caterpillar 3401
Parent engine	Caterpillar 3400 series
Number of cylinder	1
Bore x stroke	137.2 mm x 165.1 mm
Compression ratio	16.25:1
Displacement	2.44 liter
Number of valve	4
Combustion chamber type	Quiescent
Diesel fuel injection type	Direct injection
Diesel fuel injector	Ganser CRS AG
National gas injection type	Port injection
Natural gas injector	AFS Gs60 injectors
Maximum power output	74.6 kW (@2100 rpm)

An intake surge tank and an exhaust surge tank were used to help reduce pressure pulsation and simulate the turbocharging system. Compressed and temperature-controlled air was supplied to the intake surge tank to provide required intake pressure and temperature. The intake air flow rate was measured by a turbine mass flowmeter (EG&G Flow Technology, model FT-20C1NA-GEA-1). The natural gas flow rate was measured by a Bronkhorst mass flowmeter (F-113AC-M50-AAD-55-V). The diesel flow rate was measured by a TRICOR mass flowmeter (TCM-325-FK-SGSS-AZZS-EX1).

The engine was connected to the dynamometer by a flexible drive coupling (KopFlex Inc., model Holset 3.0 Max-C "CB"). Engine loading was accomplished by an eddy-current dynamometer (Mid-West, model 1014) rated to absorb 131 kW at 2500 rpm. A load cell (Lebow, model 3169) measured the dynamometer load. Engine speed was sensed by a Hall-effect transducer. A DC electric motor was used to start and motor the engine. The engine speed and load were controlled by engine's electronic control module and an AVL Digalog Testmate.

A water-cooled pressure transducer (Krister Corp., model 6041A) flush-mounted in the cylinder head was used to measure cylinder pressure. The cylinder pressure data was measured for 100 consecutive engine cycles with 0.2 crank angle (CA) degree resolution using an AVL real-time combustion analysis system (AVL LIST GmbH, IndiModule).

A heated probe was mounted after the exhaust tank to sample the gaseous emissions. California Analytical Instruments' series 600 gas analyzers were used to measure carbon dioxide (CO_2), carbon monoxide (CO), oxides of nitrogen (NO_x), total unburned hydrocarbons (THC), and methane (CH_4) emissions. Non-CH_4 hydrocarbon emissions were obtained by subtracting CH_4 emissions from THC emissions. Soot emissions were measured by a commercialized laser-induced incandescence (LII) system (Artium LII300).

The diesel fuel used in the study was a Canadian ultra-low-sulfur diesel (ULSD) fuel derived from oil sands sources. The properties of the diesel fuel are listed in Table 2. Natural gas used in the research was supplied by Enbridge Inc. The six-month average (June ~ December, 2014) natural gas composition and lower heating value are listed in Table 3. The standard deviations of methane volume fraction and lower heating value (LHV) of natural gas are 0.65% and 0.70%, respectively, suggesting that the variation in natural gas properties is small.

Table 2 Properties of the diesel fuel.

Density, kg/m^3	Cetane number	LHV, MJ/kg	H/C ratio
841.0	42	42.76	1.847

Table 3 Six-month average natural gas composition and lower heating value.

Component	Volume composition (%)/LHV (MJ/m^3 @15^{o}C and 1 atm)
Methane	95.791
Ethane	2.343
Propane	0.189
n-Butane	0.023
iso-Butane	0.019
n-Pentane	0.003
iso-Pentane	0.005
Hexanes plus	0.003
Nitrogen	0.983
Carbon dioxide	0.641
Lower heating value	37.99

Experimental Conditions

In this investigation, the experiments were conducted at 20 and 50% engine loads, with brake mean specific pressure (BMEP) being 4.05 and 8.10 bar, respectively. At each engine load, experiment was conducted at engine speeds (N) of 910 and 1400 rpm. The intake manifold pressures at the two load conditions were 105 and 150 kPa, respectively. The intake temperature was kept constant as 40^{o}C during the experiment. The diesel fuel rail pressure was 525 bar. EGR was not used in the study of this paper.

Experimental Procedure

All tests were conducted at steady state conditions. At each engine load and speed condition, the experiment started from the pure diesel combustion condition, and then the fraction of natural gas was gradually incrased. At each natural gas fraction condition, a start of diesel injection (SODI) timing sweep was conducted so that CA50 was advanced from 8.0 CA degrees (after top dead center - ATDC) to -4.0 CA degrees (ATDC) or a crank angle position at which the peak pressure rise rate reached 15 bar/CA drgree. Here CA50 is the combustion phasing, defined as the crank angle position at which 50% cumulative heat release is reached. The intake manifold pressure and temperature were kept constant during the SODI sweep.

The natural gas fraction is defined as the energy fraction provided by natural gas, i.e.

$$\alpha_{NG} = \frac{m_{NG} LHV_{NG}}{m_{NG} LHV_{NG} + m_D LHV_D} \qquad (1)$$

where m is mass flow rate, LHV is lower heating value. The subscripts NG and D represent natural gas and diesel, respectively. The natural gas fractions investigated were 0, 25 and 50% at BMEP = 4.05 bar, and 0, 25, 50 and 70% at BMEP = 8.10 bar. Higher natural gas fractions were not tested in order to avoid an excessively narrow diesel injection pulse width, which might result in combustion instability and/or diesel injector tip overheat.

RESULTS AND DISCUSSION

Combustion Performance

Figure 2 shows the variation of combustion phasing as a function of SODI at BMEP = 4.05 and 8.10 bar when engine speed was 910 rpm. The results at engine speed of 1400 rpm were qualitatively similar and thus not shown. It is noted that natural gas subsititution retarded the combustion phasing at a given SODI. This is consistent with the general understanding that natural gas has higher auto-ignition temperature than diesel fuel, which leads to longer ignition delay.

Since CA50 is an important parameter and its variation was due to the SODI sweep at a given condition, the variations of other parameters during the SODI sweep will be presented as a function of CA50 below.

Fig. 3 displays the variation of peak pressure rise rate as a function of CA50 at BMEP = 4.05 bar and engine speed of 910 rpm. Fig. 4 shows the variation of peak pressure rise rate as a function of natural gas fraction at different load and speed conditions when CA50 was 4.0 CA degrees (ATDC). It is noted that the peak pressure rise rate increased with increasing the natural gas fraction, except when BMEP = 8.0 bar and N = 1400 rpm. The increase in peak pressure rise rate with increasing natural gas fraction at most conditions was due to the increase in the fraction of premixed combustion inside cylinder. This can be shown by the heat release rate profiles at BMEP = 4.05 bar and N = 910 rpm in Fig. 5. Heat release rate profiles at other load and speed conditions were qualitatively similar. The increase in peak pressure rise rate caused that CA50 didn't reach -4.0 CA degrees (ATDC) during the SODI sweep for higher natural gas fractions, as shown in Fig. 2. However, the peak pressure rise rate slightly decreased when

25% diesel was subsitituted by natural gas at BMEP = 8.10 bar and N = 1400 rpm. It is not clear what caused the difference at this condition, but repeated test confirmed the result.

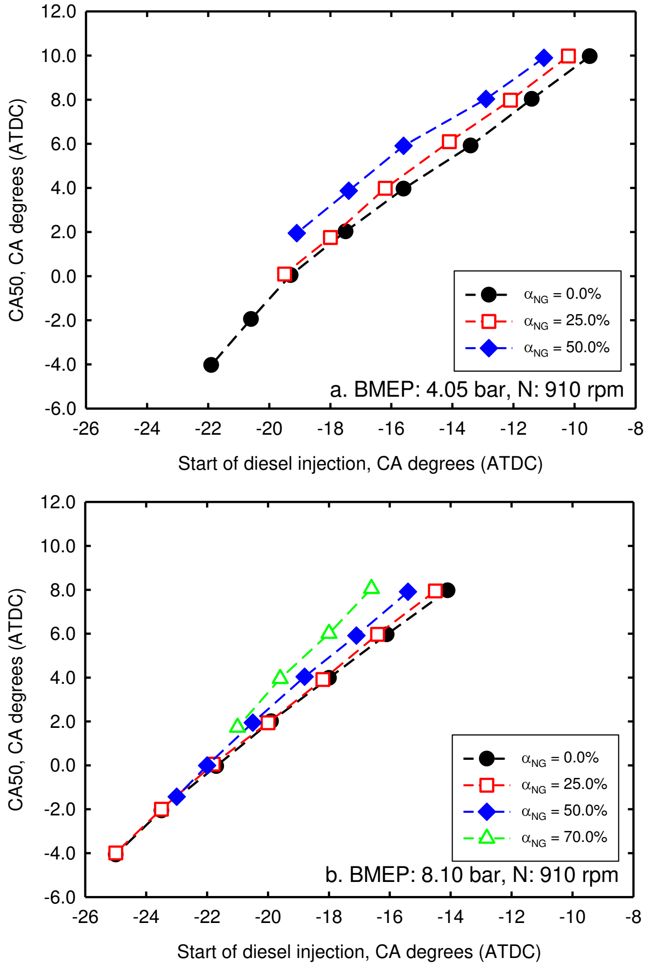

Fig. 2 Variation of CA50 as a function of SODI. a. BMEP = 4.05 bar and N = 910 rpm; b. BMEP = 8.10 bar and N = 910 rpm.

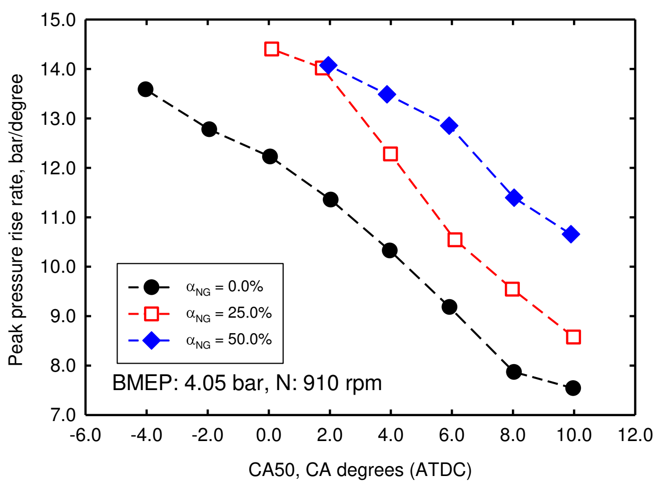

Fig. 3 Variation of peak pressure rise rate as a function of CA50.

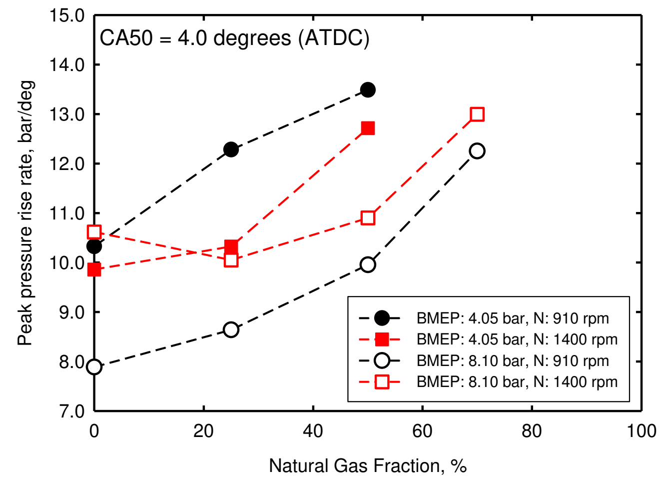

Fig. 4 Variation of peak pressure rise rate at CA50 = 4.0 CA degrees (ATDC).

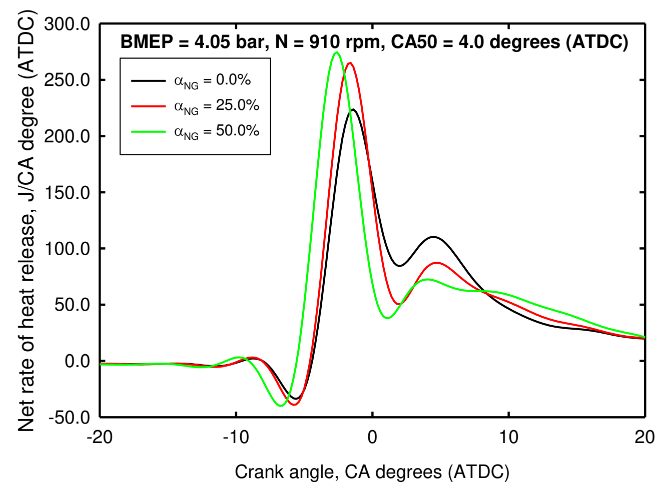

Fig. 5 Heat release rates at BMEP = 4.05 bar and N = 910 rpm.

Fig. 6 shows the variation of brake thermal efficiency as a function of CA50 at BMEP = 4.05 bar and N = 910 rpm. The results at other load and speed conditions are qualitatively similar. For pure diesel operation (α_{NG} = 0.0%), when CA50 was gradually advanced from 10.0 to -4.0 CA degrees (ATDC) due to the SODI sweep, brake thermal efficiency first reached a maximum value and then started to slightly decrease. However, for dual fuel operation, brake thermal efficiency kept increasing when CA50 was gradually advanced until a CA50 at which the critical peak pressure rise rate was reached. At a given CA50, brake thermal efficiency decreased with increasing natural gas fraction.

Fig. 7 shows the variation of brake thermal efficiency as a function of natural gas fraction for all tested load and speed conditions at CA50 = 4.0 CA degrees (ATDC). It reveals that natural gas subsititution resulted in a decrease in brake thermal efficiency at low and medium loads. The lower the load, the more significant decrease in brake thermal efficiency. This is consistent with the results obtained by other researchers [6,7].

The decrease in brake thermal efficiency might be due to the increase in CH_4 emissions, as will be discussed later.

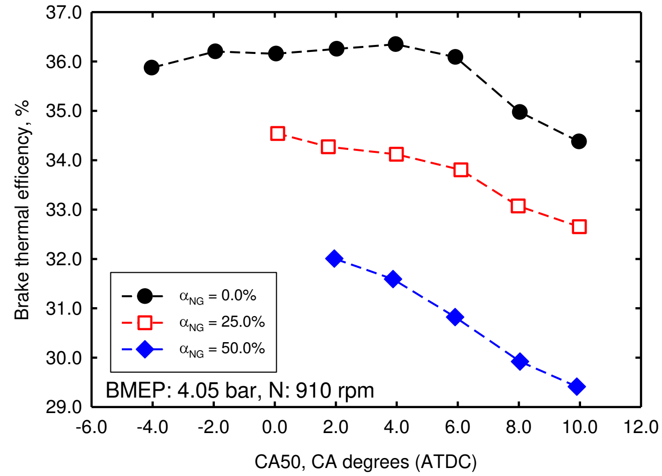

Fig. 6 Variation of brake thermal efficiency as a function of CA50.

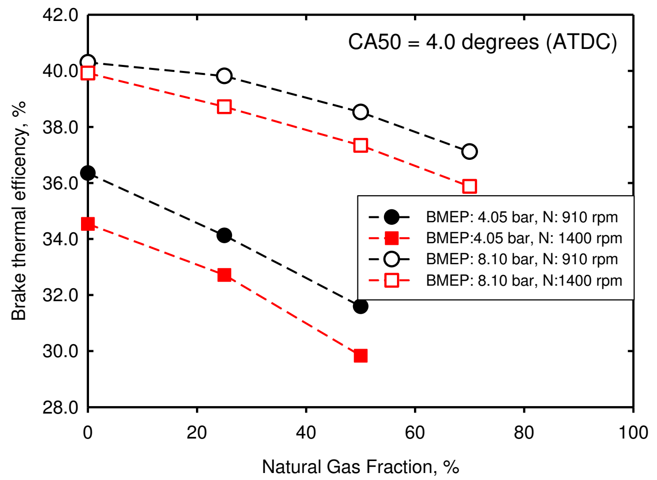

Fig. 7 Variation of brake thermal efficiency at CA50 = 4.0 CA degrees (ATDC).

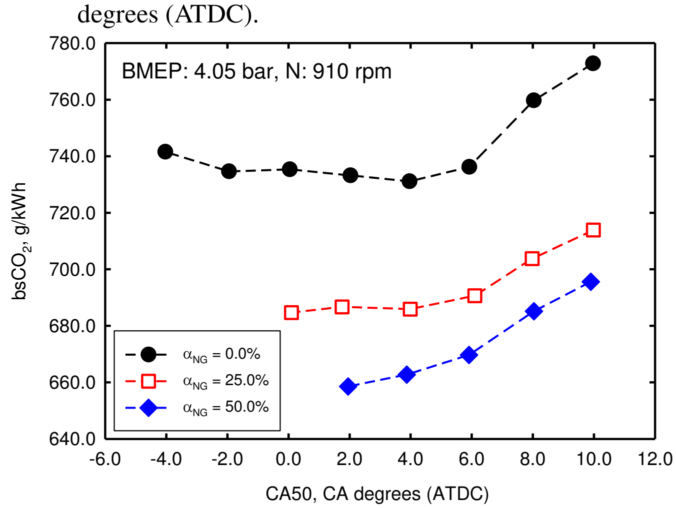

Fig. 8 Variation of CO_2 emissions as a function of CA50 at BMEP = 4.05 bar and engine speed of 910 rpm.

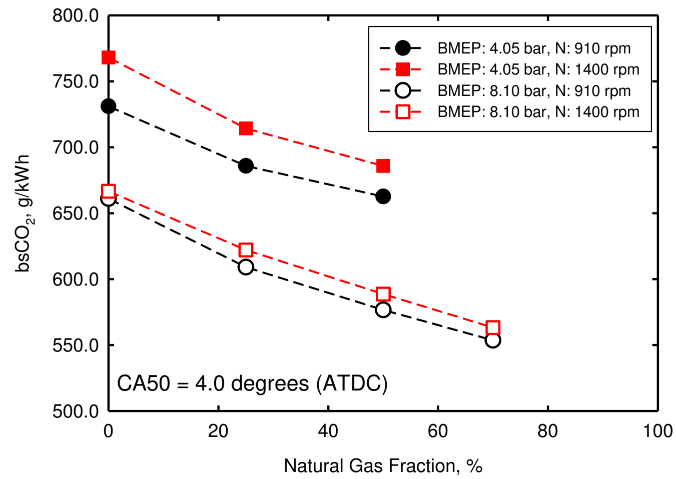

Fig. 9 Variation of CO_2 emissions as a function of natural gas fraction.

It should be mentioned that the comparison in Fig. 7 was at CA50 = 4.0 CA degrees (ATDC). The decrease in brake thermal efficiency with increasing natural gas fraction should have been lessened if the comparison was based on the maximum brake thermal efficiency of each natural gas fraction, as shown in Fig. 6, although the optimal brake thermal efficiency was not reached for dual fuel operation. Therefore, we can expect that if peak pressure rise rate can be reduced and the optimal brake thermal efficiency can be reached by further advancing CA50 for dual fuel operation, the decrease in brake thermal efficiency at low and medium loads should be further lessened.

Greenhouse Gas Emissions

The emissions of two greenhouse gases, CO_2 and CH_4, were measured in this study. Fig. 8 shows the variation of brake specific CO_2 ($bsCO_2$) emissions as a function of CA50 at BMEP = 4.05 bar and N = 910 rpm. The results at other load and speed conditions were qualitatively similar. The variation of brake specific CO_2 emissions at CA50 = 4.0 CA degrees (ATDC) as a function of natural gas fraction for different engine load and speed conditions is shown in Fig. 9. They clearly demonstrate that natural gas substitution reduced CO_2 emissions, even though there was a decrease in brake thermal efficiency. This has been well understood, because natural gas is a lower carbon fuel compared to diesel.

Methane is the main component of natural gas. The introduction of natural gas usually results in a variation in methane emissions [16]. Figs. 10 and 11 show the variation of brake specific CH_4 ($bsCH_4$) emissions as a function of CA50 at BMEP = 4.05 bar and N = 910 rpm and the variation of $bsCH_4$ as a function of natural gas fraction at CA50 = 4.0 CA degrees (ATDC) for different loads and speed conditions, respectively. As expected, natural gas subsititution caused a significant increase in CH_4 emissions. The reason for the significant increase in CH_4 emisions was that more CH_4 appeared in cylinder wall and crevice regions, because it was premixed with air. The lower the load, the more significant increase in methane emissions, because of the lower pressure and temperature inside cylinder. At a given load, an increase in engine speed resulted in a more significant increase in CH_4 emisisons, which was due to the shorter residence time at higher engine speed conditions.

The contributions of CO_2 and CH_4 to global warming are different. It is interesting to examine the overall greenhouse gas emissions, CO_2 equivalent, for natural gas - diesel dual fuel combustion. The CO_2 equivalent emissions is defined as the summation of CO_2 emissions and methane emissions multiplied by its global warming potential that is 25 [17].

Fig. 12 shows the variation of brake specific CO_2 equivalent (bsCO_2-equivalent) as a function of CA50 at three engine load-speed conditions. The result at BMEP = 4.05 bar and N = 1400 rpm is qualitatively similar to that at BMEP = 4.05 bar and N = 910 rpm, and thus not shown. We observe that at BMEP = 4.05 bar, natural gas substitution caused an increase in CO_2 equivalent emissions. This clearly was due to the significant increase in CH_4 emissions at low load conditions, as shown in Figs. 10 and 11.

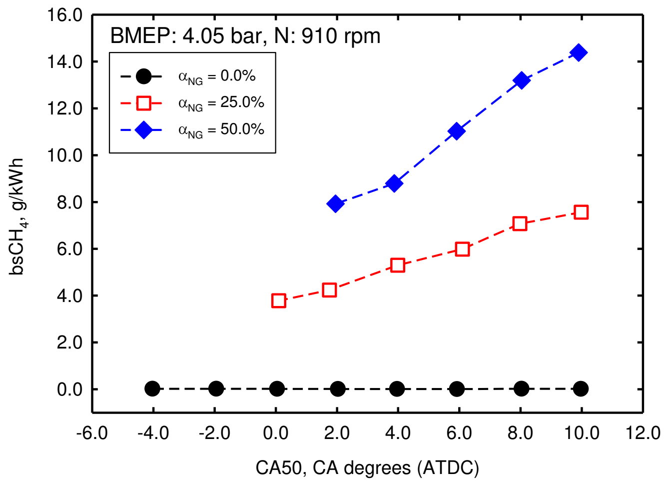

Fig. 10 Variation of bsCH_4 as a function of CA50 at BMEP = 4.05 bar and N = 910 rpm.

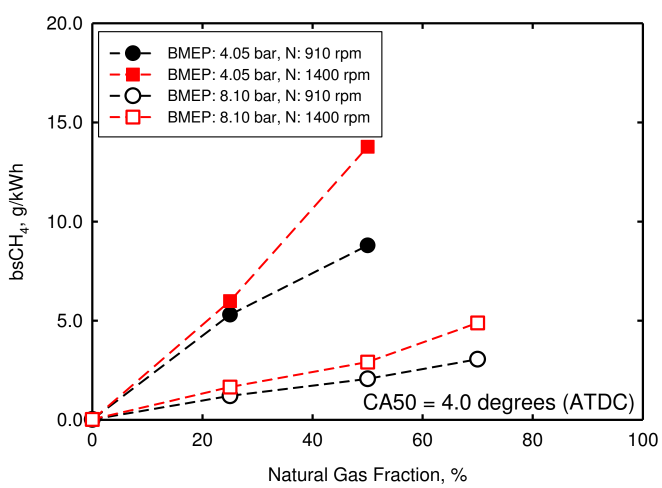

Fig. 11 Variation of bsCH_4 as a function of natural gas fraction at CA50 = 4.0 CA degrees (ATDC).

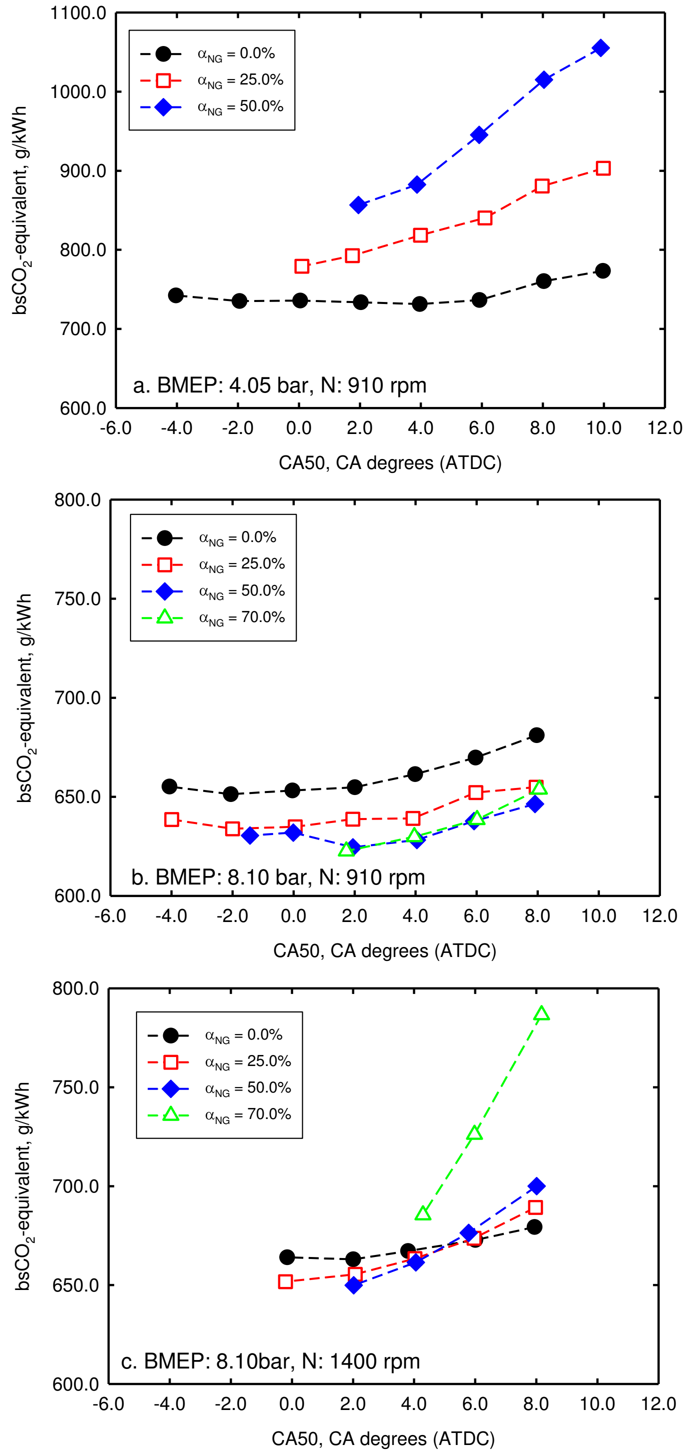

Fig. 12 Variation of bsCO_2-equivalent as a function of CA50. a. BMEP=4.05 bar and N = 910 rpm; b. BMEP = 8.10 bar and N = 910 rpm; c. BMEP = 8.10 bar and N = 1400 rpm.

At medium load conditions, the variation of CO_2 equivalent emissions became a little more complex. When engine speed was 910 rpm, the CO_2 equivalent emissions first decreased with increasing natural gas fraction from zero to 50%, because of the relatively lower CH_4 emissions than at the low load condition. However, when natural gas fraction was further increased from 50 to 70%, there was no significant change in CO_2 equivalent emissions. This was because of the slow decrease in CO_2 emissions and quick increase in CH_4 emisions at this stage.

When engine speed was 1400 rpm at BMEP = 8.10 bar, the variation of CO_2 equivalent emissions changed with CA50. When CA50 was later than 4.0 ~ 5.0 CA degrees (ATDC), CO_2 equivalent emissions increased with increasing natural gas fraction. However, the situation reversed when CA50 was more advanced than 4.0 ~ 5.0 CA degrees (ATDC). This was due to the fact that CH_4 emissions increased when combustion phasing was retarded, as shown in Fig. 10.

In summary, natural gas substitution increased CO_2 equivalent emissions at low load conditions. At medium load conditions, the variation of CO_2 equivalent emissions depended on engine speed and combustion phasing.

Pollutant Emissions

Fig. 13 shows the variation of brake specific soot (bsSoot) emissions as a function of CA50 at BMEP = 4.05 bar and N = 910 rpm. The results at other conditions were qualitatively similar. The variation of soot emissions as a function of natural gas fraction at CA50 = 4.0 CA degrees (ATDC) is shown in Fig. 14. It is observed that natural gas substitution significantly reduced soot emissions. This is consistent with the general understanding and results from other researchers [7,18]. There are two reasons for the significant reduction in soot emissions when diesel was subsitituted by natural gas. The first one is that the portion of the premixed combustion increased when natural gas substitution rate increased. The second reason is that the main component of natural gas, CH_4, is a linear alkane with higher H/C ratio, while diesel contains aromatics and other large size hydrocarbon components that usually generate much more soot than CH_4 during combustion process.

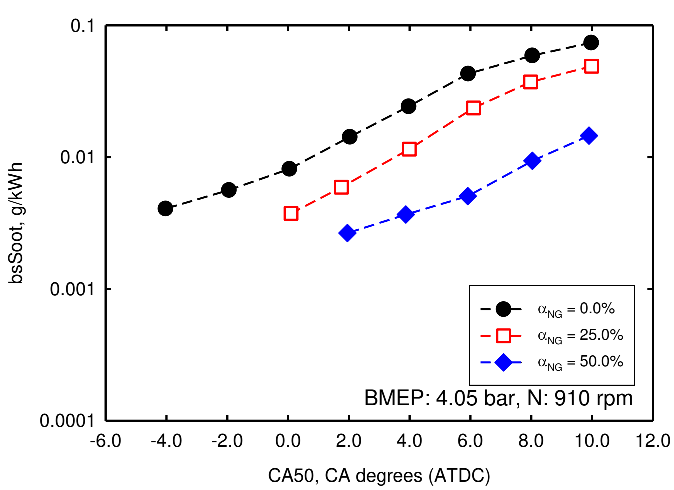

Fig. 13 Variation of soot emissions as a function of CA50.

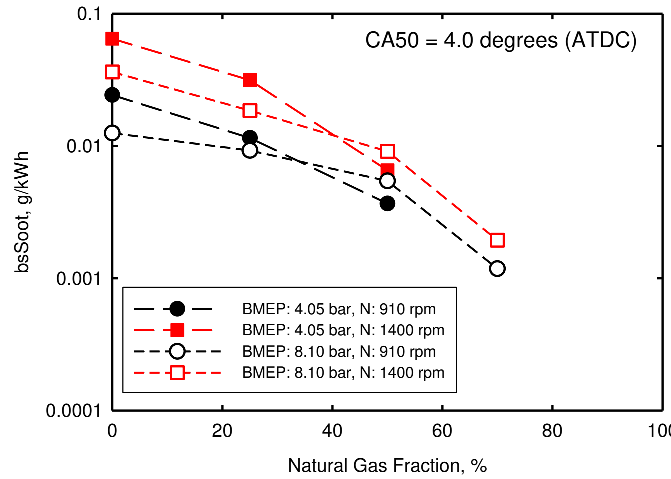

Fig. 14 Variation of soot emissions as a function of natural gas fraction.

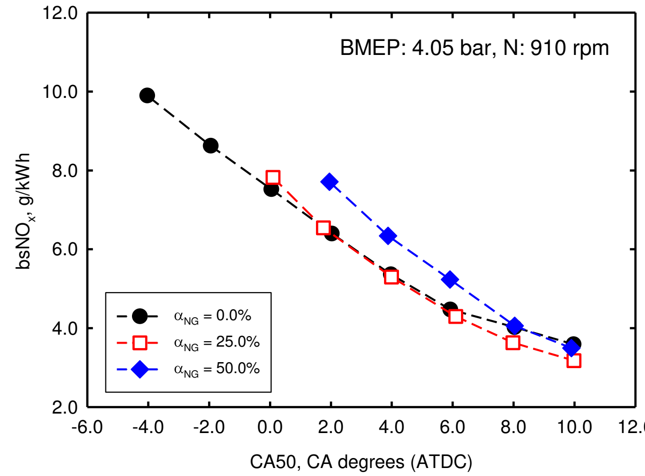

Fig. 15 Variation of bsNOx emissions as a function of CA50 at BMEP = 4.05 bar and N = 910 rpm.

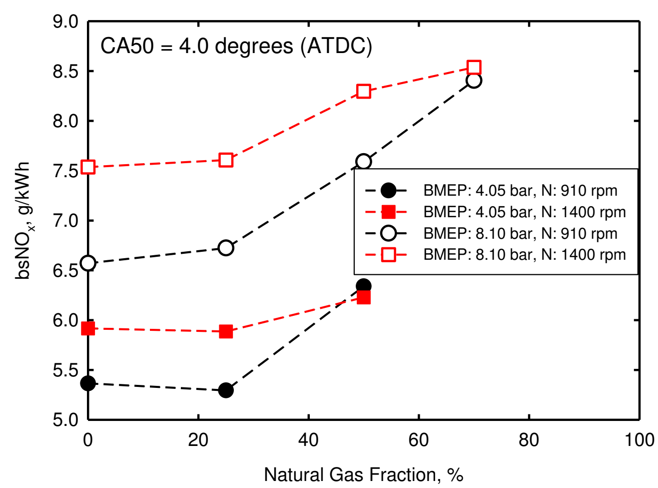

Fig. 16 Variation of $bsNO_x$ as a function of natural gas fraction.

Fig. 15 displays the variation of brake specific NO_x ($bsNO_x$) emissions as a function of CA50 at BMEP = 4.05 bar and N = 910 rpm. The results at other load and speed conditions were qualitatively similar. Fig. 16 shows the variation of $bsNO_x$ as a function of natural gas fraction for different load and speed conditions at CA50 = 4.0 CA degrees (ATDC). It is noted that when 25% diesel was subsitituted by natural gas, there was little change in NO_x emissions, but NO_x emissions slightly increased with higher natural gas fractions. This seems to be different from what Singh et al . [2] observed in a study of diesel pilot ignited natural gas combustion, but is consistent with the results of Tablan [13] at relatively lower natural gas substitution conditions.

NO, the primary component of NO_x, in diesel engines is primarily formed due to thermal and prompt mechanisms. While the NO formation due to thermal mechanism primarily depends on temperature, the NO formation due to prompt mechanism depends on both temperature and the concentration of CH radical. In the experiments, intake manifold pressure was kept constant at a given load condition, which reduced the relative air fuel ratio, λ, when natural gas was introduced. As a result, the temperature inside cylinder slightly increased with increasing natural gas fraction. This resulted in a slight increase in NO formation due to both thermal and prompt mechanisms. Therefore, NO_x emissions slightly increased with increasing natural gas fraction at most conditions, similar to the lower natural gas fraction condition results of Tablan's study [13].

When natural gas fraction increased from zero to 25% at low load conditions, the quick increase in CH_4 emissions (as shown in Fig. 11) led to little change in temperature and NO_x emissions, although λ decreased.

We should point out that the maximum natural gas fraction in this study was 70%. Further increasing natural gas fraction may result in the decrease in concentration of CH radical because of the significant increase in premixed combustion inside cylinder, which may cause a decrease in NO formation due to prompt mechanism. As a result, further increasing natural gas fraction may cause a decrease in NO_x emissions, as observed by Singh et al. for a pilot ignited natural gas engine [2] and by Tablan at higher natural gas fraction conditions [13]. Therefore, the effect of natural gas substitution on NO_x emissions may vary with the natural gas fraction and engine operating condition, although it was observed that NO_x emissions slightly increased in this study.

Fig. 17 displays the variation of brake specific carbon monoxide (bsCO) emissions as a function of CA50 at three load and speed conditions. The result at BMEP = 4.05 bar and N = 1400 rpm was qualitatively similar to that at BMEP = 4.05 bar and N = 910 rpm, and therefore not shown. At BMEP = 4.05 bar, natural gas substitution caused an monotonic increase in CO emissions. This could be due to the decrease in relative air fuel ratio when natural gas fraction increased at a giving load condition.

The variation of CO emissions at BMEP = 8.10 bar changed with engine speed. When engine speed was 910 rpm, CO emissions increased as natural gas fraction incrased from zero to about 25%, but decreased with further increasing natural gas fraction. This was due to the combined effects of

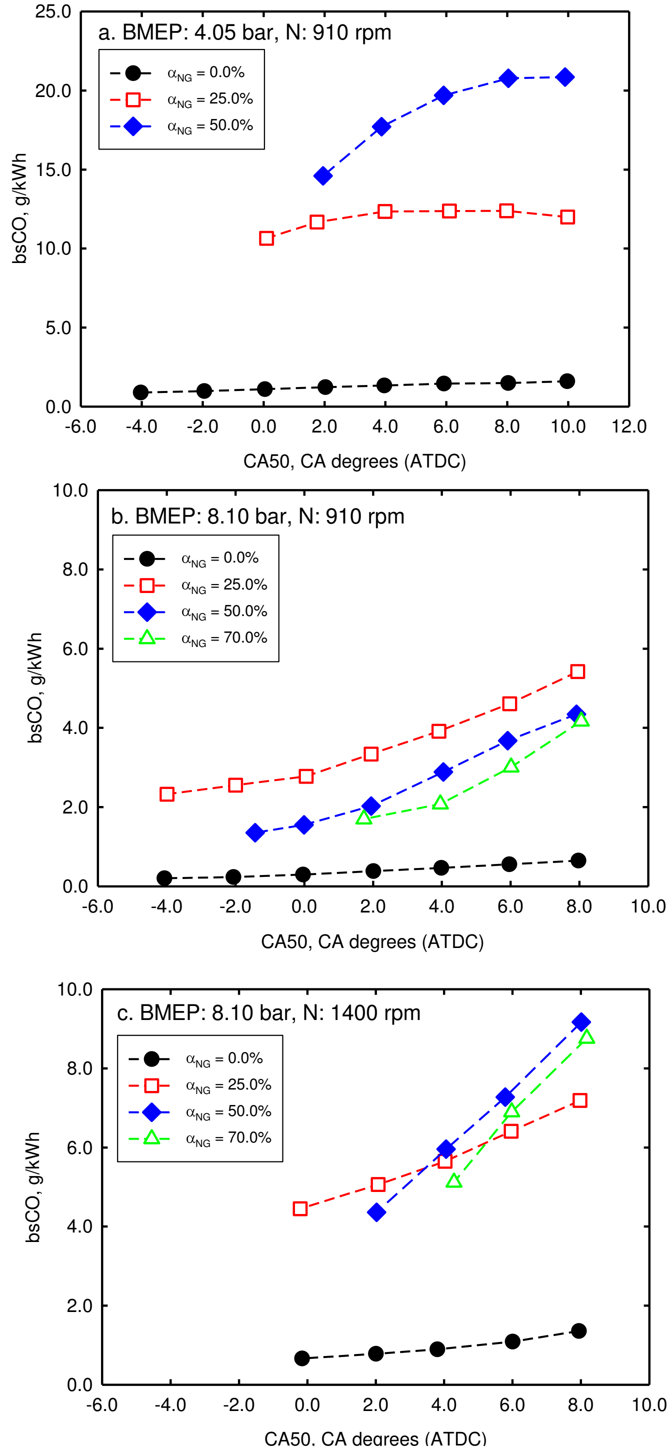

Fig. 17 Variation of bsCO as a function of CA50. a. BMEP = 4.05 bar and N = 910 rpm; b. BMEP = 8.10 bar and N = 910 rpm; c. BMEP = 8.10 bar and N = 1400 rpm.

relative air fuel ratio and temperature inside cylinder. As mentioned before, an increase in natural gas fraction caused a decrease in relative air fuel ratio and a slight increase in temperature. The decrease in relative air fuel ratio caused the increase in CO emissions when natural gas fraction increased from zero to 25%. With further increasing natural gas fraction from 25 to 70%, the increase in temperature caused CO emissions to decrease.

When engine speed was 1400 rpm at BMEP = 8.10 bar, the variation of CO emissions was qualitatively similar to that at engine speed of 910 rpm. However, the critical natural gas fraction above which CO emissions started to decrease changed with CA50. CO emissions started to decrease when natural gas fraction increased to above 50% at a later CA50, but did when natural gas fraction was above 25% at an earlier CA50. This change in the critical natural gas fraction was due to the change of residence time inside cylinder with changing CA50. The combustion of CO is a slower process. At a later CA50, the time from ignition to exhaust valve open (EVO) was shorter and there was less time for CO to be burned. Therefore, the critical natural gas fraction above which CO emission started to decrease was higher. At an earlier CA50, there was more time for CO and other components to be burned. Consequently, the critical natural gas fraction became smaller.

The difference between the CO emissions of N = 910 and 1400 rpm at BMEP = 8.10 bar might be due to the variation in turbulent intensity. At N = 910 rpm, the turbulent intensity inside cylinder was weaker. Therefore, the critical fraction was higher for all CA50s. When engine speed increased to 1400 rpm, the stronger turbulent intensity resulted in faster burning rate and thus reduced the critical natural gas fraction at an earlier CA50.

We note that advancing CA50 reduced CO emissions at all conditions for dual fuel combustion. However, the advancement of CA50 was limited by higher peak pressure rise rate, as mentioned before. Therefore, if peak pressure rise rate can be reduced and CA50 can be further appropriately advanced, CO emissions will be reduced, which will also improve engine thermal efficiency.

CONCLUSIONS

The combustion and emissions performance of a natural gas – diesel dual fuel engine has been investigated at low and medium load conditions by a modified single-cylinder version of Caterpillar's 3400-series heavy-duty engine. At each load and speed condition, the experiment started from pure diesel operation, and then gradually increased natural gas substitution ratio. At each natural gas fraction, a start of diesel injection timing sweep was conducted. Following results have been obtained:

(1) The maximum natural gas fractions reached at low and medium loads were 50 and 70%, respectively. Further increasing natural gas fraction caused excessively narrow diesel injection pulse width;
(2) Natural gas substitution retarded combustion phasing at a given diesel injection timing because of the higher ignition temperature of natural gas;
(3) Natural gas substitution decreased brake thermal efficiency at low and medium load conditions. The lower the load, the more significant decrease in brake thermal efficiency. The brake thermal efficiency of dual fuel combustion may be improved by further appropriately advancing CA50;
(4) Natural gas substitution decreased CO_2 emissions, but increased CH_4 emissions. The effect of natural gas substitution on CO_2 equivalent emissions changed with load. CO_2 equivalent emissions increased at the low load conditions, but decreased or increased depending on engine speed and combustion phasing at the medium load conditions;
(5) Natural gas – diesel dual fuel combustion significantly reduced soot emissions;
(6) Natural gas substitution caused NO_x emissions to slightly increase in the experiments of this study;
(7) The effect of natutal gas substitution on CO emissions varied with load. At the low load conditions, natural gas – diesel dual fuel combustion increased CO emissions. However, at the medium load conditions, CO emissions increased with increasing natural gas fraction first, but started to decrease with further increasing natural gas fraction.

ACKNOWLEDGMENTS

Funding for this work was provided by Natural Resources Canada through the PERD AFTER (Project AFTER16) and BEST (Project TBC012) programs, and National Research Council Canada through the internal Bioenergy Program.

REFERENCES

[1] Wagemakers, A., Leermakers, C, 2012, "Review on the Effects of Dual-Fuel Operation, Using Diesel and Gaseous Fuels, on Emissions and Performance", SAE Technical Paper Series, SAE2012-01-0869.
[2] Singh, S., Krishnan, S.R., Srinivasan, K.K., Midkiff, K.C., Bell, S.R., 2004, "Effect of pilot injection timing, pilot quantity and intake charge conditions on performance and emissions for an advanced low-pilot-ignited natural gas engine", Int. J. Engine Research, Vol. 5, pp. 329-348.
[3] Maxey, C., Kalaskar, V., Kang, D., Boehman, A., 2013, "Impact of supplemental natural gas on engine efficiency, performance, and Emissions", SAE Technical Paper Series, SAE 2013-01-0847.
[4] Jones, H.L., Taggart-Cowan, G.P., Rogak, S.N., Bushe, W.K., Munshi, S.R., Buchholz, B.A., 2005, "Source apportionment of particulate matter from a diesel pilot-ignited natural gas fuelled heavy duty DI engine", SAE Technical Paper Series, SAE 2005-01-2149.
[5] Karim, G.A., 1980, "A Review of the Combustion Process in the Dual Fuel Engine - the Gas Diesel Engine", Prog. Energy Combust. Sci., Vol. 6, pp. 277-285.
[6] Hountalas, D.T., Rakopoulos, C.D., Rakopoilos, D.C., 2008, "Combustion and Performance Characteristics

of a DI Diesel Engine Operating from Low to High Natural Gas Supplement Ratios at Various Operating Conditions", SAE Technical Paper Series, SAE 2008-01-1392.
[7] Papagiannakis, R.G., Rakopoulos, C.D., Hountalas, D.T., Rakopoulos, D.C., 2010, "Emission characteristics of high speed, dual fuel, compression ignition engine operating in a wide range of natural gas/diesel fuel proportions", Fuel, Vol. 89, pp. 1397-1406.
[8] Alla, G.H.A., Soliman, H.A., Badr, O.A., Rabbo, M.F.A. 2002, "Effect of injection timing on the performance of a dual fuel engine", Energy Conversion and Management, Vol. 43, pp. 269-277.
[9] Krishnan, S.R., Srinivasan, K.K., Singh, S., Bell, S.R., Midfiff, K.C., Gong, W., Fiveland, S.B., Willi, M., 2004, "strategies for reduced NOx emissions in pilot-ignited natural gas engines", Journal of Engineering for Gas Turbine and Power, Vol. 126, pp. 665-671.
[10] Srinivasan, K.K., Krishnan, S.R., Singh, S., Midfiff , K.C., Bell, S.R., Gong, W., Fiveland, S.B., Willi, M., 2006, "The advanced injection low pilot ignited natural gas engine: a combustion analysis", Journal of Engineering for Gas Turbine and Power, Vol. 128, pp. 213-218.
[11] Qi, Y., Srinivasan, K.K., Krishnan, S.R., Yang, H., Midkiff, K.C., 2007, "Effect of hot exhaust gas recirculation on the performance and emissions of an advanced injection low pilot-ignited natural gas engine", Int. J. Engine Research, Vol. 8, pp. 289-305.
[12] Yoshimoto, Y., Kinoshita, E., Luge, S., Ohmura, T., 2012, "Combustion characteristics of a dual fuel diesel engine with natural gas (lower limit of cetane number for ignition of the fuel)", SAE Technical Paper Series, SAE 2012-01-1690.
[13] Tablan, A.D., 2014, "Diesel and Compressed Natural Gas Dual Fuel Engine Operating Envelope for Heavy Duty Application", ASME 2014 Internal Combustion Engine Division Fall Technical Conference, ICEF 2014-5476.
[14] Königsson, F., 2012, "Advancing the Limits of Dual Fuel Combustion", Licentiate thesis, Royal Institute of Technology, SE-100 44, Stockholm.
[15] Walker, N.R., Wissink, M.L., DelVescovo, D.A., Reitz, R.D., "Natural gas for high load dual-fuel reactivity controlled compression ignition (RCCI) in heavy-duty engines", ASME 2014 Internal Combustion Engine Division Fall Technical Conference, ICEF2014-5620.
[16] Gatts, T., Liu, S., Liew, C., Ralston, B., Bell, C., Lin, H., " An Experimental Investigation of Incomplete Combustion of Gaseous Fuels of a Heavy-Duty Diesel Engine Supplemented with Hydrogen and Natural Gas", Int. J. Hydrogen Energy, Vol. 37, pp. 7848-7859.
[17] Forster, P., Ramaswamy, V., Artaxo, P., Berntsen, T., Betts, R., Fahey, D.W., Haywood, J., Lean, J., Lowe, D.C., Myhre, G., Nganga, J., Prinn, R., Raga, G., Schulz, M., Van Dorland, R., 2007, "Changes in Atmospheric Constituents and in Radiative Forcing", In: Climate Change 2007: The Physical Science Basis, Cambridge University Press, Cambridge, United Kingdom and New York.
[18] Shen, J., Qin, J., Yao, M., 2003, " Turbocharged Diesel/CNG Dual-Fuel Engines with Intercooler: Combustion, Emissions and Performance", SAE Technical Paper Series, SAE 2003-01-3082.