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Full Length Article

Effect of diesel injection timing on the combustion of natural gas/diesel dual-fuel engine at low-high load and low-high speed conditions

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ABSTRACT

Past research has shown that advancing diesel injection timing is a promising approach to decrease the unburned methane and greenhouse gas (GHG) emissions of natural gas/diesel dual-fuel engines at lower engine loads. However, this benefit may not persist under medium to high load-low speed conditions. To explore this, the present paper uses experiments and detailed computational fluid dynamic (CFD) modeling to investigate the impacts of diesel injection timing on the combustion and emissions performance of a heavy-duty natural gas/diesel dual-fuel engine under four different engine load-speed conditions. The results showed that advancing diesel injection timing increases the peak pressure, thermal efficiency, and NO_x emissions for all examined engine load-speed conditions. Advancing diesel injection timing also significantly decreases the unburned methane and CO_2-equivalent (GHG) emissions of the dual-fuel engine under low load-low speed and medium load-high speed conditions. The concentration of OH and CH_4 revealed that the central part of the combustion chamber is the main source of the unburned methane emissions under low load-low speed and medium load-high speed conditions, and advancing diesel injection timing significantly improves the combustion of natural gas-air mixture in this region. However, advancing diesel injection timing slightly increases the unburned methane emissions trapped in the crevice volume. However, this slight increase in the unburned methane emissions in the crevice volume is much lower than its significant decrease in the central region of the combustion chamber. At medium to high load-low speed conditions, there is almost no unburned methane in the central part of the combustion chamber, and the crevice region is considered as the main source of unburned methane emissions. As a result, advancing diesel injection timing does not improve the combustion of natural gas-air mixture in the central part of the combustion chamber but slightly increases the unburned methane trapped in the crevice region. This is the main reason that advancing diesel injection timing slightly increases the unburned methane emissions under medium to high load-low speed conditions. Overall, advancing diesel injection timing significantly increases thermal efficiency and decreases the unburned methane and GHG emissions under low load-low speed and medium load-high speed conditions. It improves the thermal efficiency under medium to high load-low speed conditions, but comes at the expense of increased methane and unchanged GHG emissions.

1. Introduction

Low temperature combustion (LTC) concept is recognized as a viable strategy to overcome the challenge of simultaneously suppressing the nitrogen oxides (NO_x) and soot emissions in compression ignition diesel engines. The two key features of LTC strategies consist of low combustion temperature and long ignition delay time [1]. Low temperature inhibits NO_x formation while a long ignition delay time promotes an enhanced mixing that reduces the propensity of soot formation by avoiding locally fuel-rich zones, and consequently ultra-low level of NO_x and soot below the current emissions limits may be achieved. LTC strategies have been demonstrated to result in high thermal efficiency through a combination of lean mixture, optimal combustion phasing near the top dead center (TDC), short combustion duration, and reduced heat transfer [2]. Recently, the LTC strategies have been examined by blending two fuels with different reactivity, i.e., dual-fuel combustion. Dual-fuel combustion uses direct injection of a high reactivity fuel to ignite a premixed low reactivity fuel and air mixtures [3]. Diesel is usually used as the high reactivity fuel because of its high cetane number. Natural gas is great candidate for the low reactivity fuel due to its high ignition temperature. In the dual-fuel combustion, the ignition process is initiated in the high reactivity

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Nomenclature

AMR	Adaptive Mesh Refinement
ASOC	After Start of Combustion
ATDC	After Top Dead Center
BMEP	Break Mean Effective Pressure
CAD	Crank Angle Degree
CFD	Computational Fluid Dynamic
CO	Carbon monoxide
COV	Coefficient of Variation
DIT	Diesel Injection Timing
EGR	Exhaust Gas Recirculation
EVC	Exhaust Valve Closing
EVO	Exhaust Valve Opening
GHG	Greenhouse Gas
IMEP	Indicated Mean Effective Pressure
IVC	Intake Valve Closing
IVO	Intake Valve Opening
$ISCH_4$	Indicated Specific CH_4
$ISNO_x$	Indicated Specific NO_x
$ISCO_2$	Indicated Specific CO_2
LTC	Low Temperature Combustion
NO_x	Nitrogen Oxides
RPM	Revolution per Minute
SOC	Start of Combustion
TDC	Top Dead Center

regions where the liquid fuel is directly injected. Owing to the premixed charge of lower reactivity fuel-air mixture, the combustion process is controllable, which sequentially progresses from the higher reactivity to the lower reactivity regions.

Dual-fuel mode not only is a fuel flexible approach, but also has the potential to lead to high efficiency clean combustion in compression ignition engines. In recent years, natural gas has drawn substantial interest as a low reactivity fuel in dual-fuel combustion and some of the original versions of diesel engine have been commercialized to operate as a dual-fuel engine based on premixed natural gas [4–6]. Natural gas/diesel dual-fuel combustion tends to retain most positive features of conventional diesel engines, and is capable of producing comparable power output and efficiency at different engine loads [7]. Moreover, this strategy can be achieved via the installation of a low cost port fuel injection system for the formation of a low reactivity mixture of air and fuel (e.g., natural gas), while, the stock diesel fuel injection system can be retained in the dual-fuel mode. However, the greatest challenge of natural gas/diesel dual-fuel engine is the high level of unburned methane emissions which contribute to the GHG by about 25 times greater than CO_2 over a 100 year period [8]. Past research has shown that advancing diesel injection timing reduces the unburned methane and CO emissions of natural gas/diesel dual-fuel engine under low-medium engine load conditions [9–12]. It is shown that advancing diesel injection timing under low load-low speed and medium load-high speed conditions significantly decreases the unburned methane and GHG emissions [13–16]. However, according to the experimental data of the present study, advancing diesel injection timing under medium to high load-low speed conditions increases the unburned methane emissions and does not change the GHG emissions. Finding the reasons behind this phenomenon is the main objective of the present study.

Accordingly, four typical cases of drive cycle in heavy-duty diesel engine are selected to investigate the effect of diesel injection timing on natural gas/diesel dual-fuel engine. Cases 1 and 2 represent the low load-low speed (Case 1, BMEP = 4.05 bar, RPM = 910) and medium load-high speed (Case 2, BMEP = 11.24 bar, RPM = 1750) conditions, respectively. Advancing diesel injection timing under these conditions significantly reduces the unburned methane and GHG emissions. Cases 3 and 4 represent the medium load-low speed (BMEP = 12.15 bar, RPM = 910) and high load-low speed (BMEP = 17.15 bar, RPM = 1120) conditions, respectively. The opposite scenario manifests under these cases where advancing diesel injection timing increases the unburned methane emissions. A computational fluid dynamic (CFD) model based on CONVERGE 2.4 software is developed to examine and hence understand the reasons behind the underlying phenomenon. Cylinder pressure, engine out emissions, and OH and CH_4 distributions are all analyzed in order to investigate the effect of diesel injection timing on natural gas/diesel dual-fuel engine under different engine load-speed conditions.

2. Methodology

2.1. Experimental setup

The engine used in this investigation is a modified single-cylinder heavy-duty engine. Detailed description of the experimental setup and methodology is reported elsewhere [17], and only a brief summary is provided here. Specifications of the engine are provided in Table 1. Natural gas fuel port injector was fed by a low-pressure line which included eight gas fuel injectors manufactured by Alternative Fuel Systems Inc. Diesel fuel, natural gas, and air flow rates were measured using a TRICOR, a Bronkhorst, and a turbine mass flowmeters, respectively. Engine loading was accomplished by an eddy-current dynamometer. The engine speed and load were controlled by an electronic control module and an AVL Digalog Testmate, respectively. The in-cylinder pressure was measured by a water-cooled pressure transducer (Krister Corp.) fitted inside the cylinder head and has a resolution of 0.2 crank angle degree. The averaged pressure signal for the calculation of the indicated mean effective pressure (IMEP) and heat release rate (HRR) were averaged over 100 consecutive cycles using an AVL real-time combustion analysis system. The emitted smoke was measured using a commercialized laser-induced incandescence system. The engine-out gaseous emissions such as CH_4, NOx, CO, and CO_2 were measured using a California Analytical Instruments' series 600 gas analyzers.

The experiments were conducted at four different engine load-speed conditions including low load-low speed (BMEP = 4.05 bar and 910 RPM), medium load-high speed (BMEP = 11.24 bar and 1750 RPM), medium load-low speed (BMEP = 12.15 bar and 910 RPM), and high load-low speed (BMEP = 17.6 bar and 1120 RPM). A diesel injection timing (DIT) test was conducted during the investigation for each case. For all cases, the intake temperature was kept constant at 40 °C during the experiments, which is close to the engine room temperature. The selection of this temperature was due to the fact that the engine test cell

Table 1
Test setup specifications.

Engine type	Single cylinder-CAT 3400
Bore × stroke	137.2 mm × 165.1 mm
Conn. rod length	261.62 mm
Displacement vol.	2.44 L
Compression ratio	16.25
Diesel fuel injector	Common rail injector
Number of nozzle hole×diameter	6 × 0.23 mm
Maximum speed	2100 rpm
Low idle speed	600 rpm
Rated power BMEP @1800 rpm	20.6 bar
IVO	−358.3 °ATDC
IVC	−169.7 °ATDC
EVO	145.3 °ATDC
EVC	348.3 °ATDC