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Full Length Article

An experimental and numerical study on diesel injection split of a natural gas/diesel dual-fuel engine at a low engine load

Amin Yousefi[a], Hongsheng Guo[b,*], Madjid Birouk[a,*]

[a] Department of Mechanical Engineering, University of Manitoba, Winnipeg, Manitoba R3T 5V6, Canada
[b] Energy, Mining and Environment Portfolio, National Research Council Canada, 1200 Montreal Road, Ottawa, Ontario K1A 0R6, Canada

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ABSTRACT

Natural gas/diesel dual-fuel combustion is currently one of the most promising LTC strategies for the next generation of heavy-duty engines. While this concept is not new and it has been deliberated lengthily in the past two decades, several uncertainties still exist. A major shortcoming of this concept is associated with its low thermal efficiency and high level of unburned methane and CO emissions under low engine load conditions. The present paper reports an experimental and numerical study on the effect of different injection strategies (single and two pulses injection of pilot diesel fuel) on the combustion performance and emissions of a heavy duty natural gas/diesel dual-fuel engine at 25% engine load. The results of single diesel injection mode showed that advancing diesel injection timing from 10 to 30 °BTDC reduced unburned methane and CO emissions by 62% and 61% and increased thermal efficiency by 6%; however, NO_x emissions increased by 74%. In order to achieve NO_x – CH_4 and NO_x – CO trade-off and increased thermal efficiency at low load conditions, the effect of split injection strategy was experimentally and numerically examined. The results of split injection mode revealed that split injection strategy considerably increases the in-cylinder peak pressure compared to that of single injection (10 °BTDC). The results showed also that the heat release produced by the first injection of diesel fuel considerably increased the in-cylinder charge temperature before the start of the second injection. The flame zone of the split injection mode is markedly higher than that of the single injection due to larger heat release produced during the first injection which promotes the combustion of the second one. When the first injection timing is close to the second injection timing, the MPRR of split injection mode is higher than that of single injection (10 °BTDC). However, further advancing of the first injection timing continuously decreased the MPRR. OH radical analysis showed that for advanced first injection timings (38–50 °BTDC), the overall growth rate of OH radical becomes slower and its distribution is narrower as indicated by the wider non-reactive blue zones compared with those observed at a late first injection timing in the initial stages of combustion. However, OH radicals gradually grow during last stages of combustion in the expansion stroke, indicating that a more premixed combustion takes place in these cases. For very advanced first injection timing of 55 °BTDC, the OH distribution is similar to that of the single injection mode with lower OH intensity at initial stages of combustion and they barely grow during the late expansion stroke. At this condition, the ignition of premixed mixture is mainly controlled by the second diesel fuel injection. The trade-off between NO_x – CH_4 and NO_x – CO is achieved when applying split injection. Compared to single injection (10 °BTDC), the first injection timing of 50 °BTDC decreased unburned methane and CO emissions by 60% and 63%, respectively, and increased the thermal efficiency by 8.9%. However, NO_x emissions were maintained at the same level as single injection mode (10 °BTDC).

Abbreviations: ADIT, After Diesel Injection Timing; AMDIT, After Main Diesel Injection Timing; ASOC, After Start of Combustion; ASOI, After Start of Injection; BMEP, Break Mean Effective Pressure; BTDC, Before Top Dead Center; BTE, Break Thermal Efficiency; CA, Crank Angle; CFD, Computational Fluid Dynamic; CI, Compression Ignition; CO, Carbon monoxide; DIT, Diesel Injection Timing; EGR, Exhaust Gas Recirculation; EOI, End of Injection; EVO, Exhaust Valve Opening; HCCI, Homogeneous Charge Compression Ignition; HRR, Heat Release Rate; ITE, Indicated Thermal Efficiency; LHV, Lower Heating Value; LTC, Low Temperature Combustion; MPRR, Maximum Pressure Rise Rate; NO_x, Nitrogen Oxides; SOC, Start of Combustion; SOI, Start of Injection; TDC, Top Dead Center; THC, Total Hydrocarbon; UHC, Unburned Hydrocarbon; VCR, Variable Compression Ratio

* Corresponding authors.
E-mail addresses: Hongsheng.Guo@nrc-cnrc.gc.ca (H. Guo), madjid.birouk@umanitoba.ca (M. Birouk).

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1. Introduction

The usage of petroleum as an energy source is expected to decrease due to limited global oil reserves, its negative impact on the environment, and stringent emissions regulations. This will affect compression ignition (CI) diesel engines which play significant role in transportation and power generation, where there is a need for cleaner, more economical, and reliable alternative fuels. Conventional diesel engines also suffer from high soot emissions due to the over-rich regions in the core area of the fuel spray and high nitrogen oxides (NO_x) emissions as a result of high flame temperature of the stoichiometric fuel-air mixture at the periphery of the fuel spray. In order to reduce both soot and NO_x emissions, fuel-rich and high temperature stoichiometric regions should be avoided simultaneously [1]. An effective approach is to employ low temperature combustion (LTC) strategies which are featured by improved fuel atomization, mixture preparation, lower local equivalence ratios, reduced local temperature, and alternative fuels [2]. Homogeneous charge compression ignition (HCCI) combustion is one of such strategies characterized by early fuel injection, which promotes fuel premixed charge, long ignition delay, and short combustion duration. Ignition timing is kinetically controlled and therefore decoupled from the timing of the fuel injection event [3,4]. However, the lack of direct control of ignition timing and combustion phasing, higher unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions, as well as knock and misfiring under transient conditions, are the major drawbacks of HCCI combustion engines [5–8]. In contrast, some slightly more heterogeneous combustion strategies have been developed to overcome the majority of the aforementioned challenges. For example, the charge distribution is more heterogeneous than HCCI combustion as it consists of lean and rich regions at the time of ignition. Moreover, ignition timing is closely coupled to the fuel injection timing, though chemical kinetics still play an important role [3].

Natural gas/diesel dual-fuel combustion is one of these LTC strategies which allow a higher degree of combustion phasing control while maintaining low soot and NO_x emissions. In a dual-fuel engine, the primary method of fuel delivery is the port injection of natural gas which creates well-mixed charge of premixed fuel-air, while a small amount of diesel fuel is directly injected into the cylinder as the ignition source. Natural gas/diesel dual-fuel combustion tends to retain most and even surpasses occasionally the positive features of conventional diesel engines, and producing comparable power output and efficiency at different engine loads [9–11]. In addition, natural gas/diesel dual-fuel mode has attracted much interests due to other advantages, such as simple modification from a diesel engine and the flexibility in switching back to fully diesel mode [12]. Moreover, this combustion concept relies on natural gas as the major energy source, which yields lower carbon dioxide (CO_2) emissions due to the higher hydrogen to carbon ratio. However, there still exist some issues that are limiting the application of natural gas/diesel dual-fuel engines. One of these issues is the low thermal efficiency and higher unburned methane and CO emissions at low engine load conditions. At low load conditions, a natural gas/diesel dual-fuel engine is fed with a very lean air-fuel mixture which is difficult to ignite and burn, leading to significant levels of unburned methane and CO emissions. This is because, at very lean air-methane mixtures, portions of the charge which resides far away from diesel fuel spray escape the combustion process [13–15]. Numerous studies have addressed these issues by examining the effect of combustion boundaries, such as diesel injection timing and pressure [16–20], natural gas energy fraction [20–24], variable compression ratio (VCR) [25–27], natural gas injection timing [20,28], and the use of exhaust gas recirculation (EGR) [29,30]. Among these strategies, diesel injection timing change is of great interest and usually regarded as a critical factor which has influence on the combustion performance and emissions characteristics.

Various researchers have examined the effect of conventional diesel injection timing (i.e., 5–30 °before top dead center (BTDC)) on thermal efficiency and unburned methane and CO emissions of natural gas/diesel dual-fuel mode at low conditions [10,14,17,31–34]. For instance, Zhang et al. [34] examined the effect of diesel injection timing (DIT) sweep (DIT = 7–25 °BTDC with 2 °crank angle (CA) increment) on combustion performance and emissions of natural gas/diesel dual-fuel mode and found that total hydrocarbon (THC) and CO emissions reduced with advancing diesel injection timing. On the other hand, they noted increased NO_x emissions with advanced diesel injection timing due to higher in-cylinder temperature. Yang et al. [14] investigated various diesel injection timings (DIT = 5–29 °BTDC with 4°CA increment) in a natural gas/diesel dual-fuel engine at a low engine load. They reported that advancing diesel injection timing from 5 to 29 °BTDC significantly increased break thermal efficiency (BTE). Moreover, they observed that, with advancing diesel injection timing, THC and CO emissions notably decreased due to the relatively higher combustion rate and greater utilization of premixed natural gas at earlier injection timings. Similar to the findings in [34], NO_x emission was observed to significantly increase with advancing diesel injection timing in [14]. Wang et al. [33] also observed that advancing diesel injection timing from −5 to 22.5 °BTDC drastically increased NO_x emissions, while decreased THC emissions.

Based on the briefly reviewed literature above, it is revealed that advancing diesel injection timing in the range of conventional diesel injection timing (i.e., 5–30 °BTDC) improved thermal efficiency as well as unburned methane and CO emissions but generated higher NO_x emissions. To address this issue, various single pulse conventional diesel injection timings (10–30 °BTDC with 4°CA increment) are experimentally and numerically examined in the present paper using a natural gas/diesel dual-fuel combustion with 75% natural gas energy fraction under 25% engine load (break mean effective pressure (BMEP) = 4.05 bar). Afterwards, the effect of diesel injection split (two pulses injection) as a feasible method to decrease both NO_x and unburned methane emissions and increase thermal efficiency is examined under the same engine load condition. In particular, the effect of first pulse injection timing (28–55 °BTDC) with fixed split injection ratio of 60% and second pulse injection timing of 10 °BTDC on combustion performance and emissions of natural gas/diesel dual-fuel engine is investigated. This provides useful information for the optimization of diesel injection strategy for natural gas/diesel dual-fuel combustion at low load engine conditions.

2. Experiments

2.1. Test engine

The engine used in this investigation is a modified single-cylinder version of Caterpillar's 3400-series heavy-duty engine. More details about the experimental setup and engine configuration can be found elsewhere [35]. Table 1 lists the specifics of the engine and Fig. 1 depicts the schematic diagram of test setup.

Natural gas was injected into the intake port by a fuel injection manifold. Diesel fuel was directly injected into the cylinder using a prototype common-rail fuel injector system. The start of injection and injection pulse width for both diesel and natural gas were controlled by a driven system provided 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). The flow rates of diesel and natural gas were measured by two Bronkhorst mass flowmeters, respectively, and the flow rate of air was measured by a turbine mass flowmeter.

2.2. Fuels and their supply systems

Diesel fuel used in this study was a Canadian ultra-low-sulfur diesel (ULSD), and the natural gas used in this research was supplied by