

Emissions Characterization from Different Technology Heavy-Duty Engines Retrofitted for CNG/Diesel Dual-Fuel Operation

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ABSTRACT

This study was aimed at experimentally investigating the impact of diesel/natural gas (NG) dual-fuel retrofitting onto gaseous emissions emitted by i) legacy, model year (MY) 2005 heavy-duty engines with cooled EGR and no after-treatment system, and ii) a latest technology engine equipped with DPF and urea-SCR after-treatment systems that is compliant with 2010 US-EPA emissions standards. In particular, two different dual-fuel conversion kits were evaluated in this study with pure methane (CH_4) being used as surrogate for natural gas. Experiments were conducted on an engine dynamometer over a 13-mode steady-state test cycle as well as the transient FTP required for engine certification while gaseous emissions were sampled through a CVS system.

Tailpipe NO_x emissions were observed at a comparable level for diesel and diesel/ CH_4 dual-fuel operation for the 2010 compliant engine downstream the SCR. However, a reduction of $\sim 26\%$ in diesel exhaust fluid (DEF) consumption was measured for dual-fuel operation, corroborating with engine-out NO_x concentrations that were observed to be $\sim 32\%$ lower as compared to diesel-only operation, indicating that the addition of natural gas significantly decreased NO_x emissions formed during the combustion process. On the other hand, dual-fuel operation of the non after-treatment equipped, legacy engines showed conflicting NO_x emissions results. While NO_x emissions were observed to reduce for one of the legacy engines by $\sim 25\%$, dual-fuel operation of the second legacy engine resulted in a 6.6% increase in brake-specific NO_x .

Carbon dioxide emissions (CO_2) were observed to decrease across all steady-state engine load modes and by 3 to 8% over the transient FTP cycle with no specific trend towards an engine technology. Methane addition to the post-2010 engine, however, resulted in the highest overall CO_2 reduction for the given amount of CH_4 injected. Additionally, this study was focusing on evaluating possible greenhouse gas benefits (i.e. tank to exhaust) from diesel/NG retrofitting as a function of engine technology. It was observed that despite the reduction in CO_2 emissions the global warming potential (GWP) benefits of dual-fuel operation were outweighed by a significant increase in methane-hydrocarbon (MHC) emissions, specifically by factors of 1.6 and 2.3–4.8 for the post-2010 and legacy engines, respectively.

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INTRODUCTION

Natural gas (NG) has emerged as a promising energy source to power the transportation sector over the next decades due to its abundance and comparably less production intensive process from well-to-tank than associated with conventional liquid hydrocarbon fuels. Furthermore, natural gas's clean-burning properties and reduced carbon footprint [1, 2, 3, and 4] make it a fuel of choice to comply with upcoming stricter emissions and greenhouse gas (GHG) regulations and aid in attaining national ambient air quality standards. Diesel/NG dual-fuel operation has gained increased interest as it allows combining both improved combustion efficiencies, inherent to the diesel-cycle, with the low carbon properties of NG.

Dual-fuel systems can be broadly categorized into indirect natural gas injection into the intake tract and direct injection of NG into the

combustion chamber [5, 6, and 7]. The former is introducing a homogeneous air-gas mixture into the combustion chamber which is being compressed below auto-ignition conditions and ultimately ignited by the pilot diesel fuel injection around top dead center [2, 3, and 7]. The latter uses two separate injectors or proprietary injection systems such as the high pressure direct injection (HPDI®) technology from Westport, Inc. Direct injection does not allow the gas to premix with surrounding air leading to a more dominant diffusion mode [5], and thereby more efficient combustion with engine performance close to a diesel engine of similar size [8, 9]. With heavy-duty (HD) vehicle turnover rates on the order of two decades, however, [10] retrofitting existing diesel engines to diesel/NG dual-fuel operation is becoming an increasingly interesting option for fleet-owners to take advantage of NG's advantages over their cost of operation.

Dual-fuel retrofit technologies are developed as ‘add-on’ devices and intended to be mated to a wide range of engine technologies. Natural gas injection techniques vary depending on conversion kit type and include pre-compressor fumigation, post-intercooler injection, intake manifold injection, and sequential port injection [5, 6]. Their control algorithms are usually calibrated for a specific engine family using an empirical approach and natural gas substitution rates are primarily a function of engine operating conditions characterized by engine speed and torque demand as well as ultimately, drivability of the vehicle. Furthermore, control strategies may include interception and alteration of diesel injector signals or emulation of throttle signal to the original engine manufacturer (OEM) engine control unit (ECU) for the purpose of reducing the injected diesel pilot quantity and inject an energy-equivalent amount of natural gas to maintain the desired engine power output level [5]. In general, dual-fuel conversion retrofitting methods do not actively alter the existing engine control unit's operating strategies. Therefore, such a simplified retrofitting approach poses a unique challenge to the operational efficiency and control of highly integrated after-treatment systems available on modern heavy-duty diesel engines.

BACKGROUND

The primary benefits of converting compression-ignition (CI) as opposed to spark-ignition (SI) engines to dual-fuel operation is the inherently higher engine efficiency attainable with CI engines [6], and that conversion is possible without major changes to the base engine [6, 11]. Furthermore, indirect NG injection systems allow for the engine to be operated in conventional diesel-only operation if no natural gas is available as opposed to NG direct injection systems that are only capable of reduced power operation (i.e. ‘limp home’ mode) in case of an empty natural gas tank [3, 6]. On the other hand, challenges associated with dual-fuel operation include the difficulty to operate the engine in dual-fuel mode during idle and low engine loads, and that advanced emissions control systems are still required to meet stringent, current, and future proposed emissions standards [6]. Furthermore, due to high sensitivity for engine knocking, natural gas substitution rates are limited at full load operation [6] (i.e. knock limited gas replacement) and increased methane emissions are observed for dual-fuel operation as the combustion chamber of a diesel engine is not optimized for port injected premixed combustion, thus leading to methane slip from flame quenching and crevices [3, 6] as well as poor CH_4 conversion rates over the existing diesel oxidation catalysts [6]. The increase in methane slip from combustion translates into an effective decrease in combustion efficiency. An additional limiting factor includes high diesel injector tip temperatures for large NG substitution rates as the lower quantities of diesel pilot injection lead to reduced cooling of the injector tip [6, 12]. Therefore, control strategies often move towards increased diesel pilot injection quantities to allow for sufficient tip cooling, thus resulting in overall richer combustion [12].

Fundamental studies of dual-fuel combustion [11, 13] have identified the equivalence ratio of the premixed fuel (i.e. proportional to substitution rate of diesel by natural gas) to be a key parameter that significantly impacts the rate of heat release. Dronniou *et al.* [13] performed an optical investigation of fundamental combustion phenomena occurring during ignition of methane-air mixture with

pilot injection of diesel fuel for a wide range of equivalence ratios of the premixed charge on a single cylinder research engine. For low equivalence ratio conditions (i.e. low NG substitution rates), the combustion of the premixed charge of methane gas was found to be dominated by spray entrainment and mixture stratification of diesel fuel [13] with the dominant portion of combustion occurring in the diffusion phase [6, 11, 13, and 14] (i.e. no flame propagation or bulk ignition). It was concluded from imaging data that the consumption of premixed fuel in the central part of the charge is likely to be a significant challenge for dual-fuel combustion strategies that are operating on an overall lean equivalence ratio of premixed fuel (i.e. lower NG substitution rates) [13]. This observation was deemed significant as it could possibly explain the increase in unburned hydrocarbon emissions found by other studies [3, 6, 7, and 11 (to name only a few)] using dual-fuel combustion strategies under fuel-lean conditions.

On the other hand, for higher equivalence ratios, especially close to stoichiometric conditions (i.e. high NG substitution rates), time-resolved natural luminosity imaging revealed significant modifications in combustion behavior indicating evidence of flame propagation becoming the dominant governing factor of combustion [13]. Uniform high-temperature combustion reactions throughout the combustion chamber were observed provided that the equivalence ratio of the premixed fuel exceeded a certain threshold value [13]. Thus, it was concluded that combustion strategies operating at equivalence ratios near stoichiometric mixture conditions (i.e. high NG substitution rates) exhibit greater potential in terms of ensuring more complete combustion compared to low equivalence ratio strategies (i.e. lean conditions or low NG substitution rates) [11, 13].

The reported efficiency of retrofitted heavy-duty diesel engines is normally 8-12% lower compared to the original diesel-only base engine over transient test cycles [6]. Indeed, port fuel injection of gaseous fuel is found to significantly reduce the volumetric efficiency of the engine due to fuel vapors in the intake [3]. Furthermore, a reduction in the amount of fresh air, and thereby oxygen concentration, in the cylinder charge leads to an effective reduction in maximum rated power compared to diesel-only operation [3]. Brake-specific fuel consumption (BSFC) (i.e. including gaseous and liquid fuel) is reported to increase for port-fuel injected dual-fuel systems as a function of increasing NG content in the intake mixture [3, 15, and 16]. This is assumed to be mostly due to the lower charge temperatures and combustion rates, especially predominant for low load operation [15, 16].

The increased methane concentrations in the exhaust stream may pose an additional challenge for modern after-treatment equipped engines. CH_4 light-off temperatures for an aged oxidation catalyst are typically in the range of about 400-450°C [6]. On one hand, oxidation catalysts of base diesel engines are not optimized for methane conversion and exhaust temperatures remain typically below the aforementioned light-off range for the majority of engine operating conditions, thus, leading to poor CH_4 conversion rates [6]. On the other hand, high levels of engine-out hydrocarbon concentrations will put increased demand on the efficiency, and especially durability of the default OEM oxidation catalysts [6]. Conversely, vanadium-based selective catalytic reduction (SCR) catalysts Document Sharing Hub

temperature limit of approximately 580-600°C where they become unstable and might sustain non-reversible damage [6, 17]. Therefore, elevated exhaust gas temperatures after the methane oxidation catalyst could increase the risk of accelerated aging or damaging vanadium-based SCR systems [6]. As a result, dedicated lean burning CNG engines have been equipped with Cu-Fe-based zeolite catalysts because of their higher hydrothermal stability and desirable conversion efficiency at high exhaust gas temperatures [17, 18].

Furthermore, studies [6, 19, 20, and 21] have highlighted the significant impact changes in NG quality and composition may have on engine performance, durability and more importantly emissions characteristics. NG fuel quality can be characterized based on [6] i) energy content; ii) content of longer chain hydrocarbons; and iii) level of contamination. Generally, natural gas fuel is described by two parameters, the methane number (MN), which is a measure of knock resistance, and the Wobbe index (WI), which is a measure of fuel interchangeability [19, 21]. A comprehensive survey of natural gas fueling stations in Southern California [19] found MN and WI to vary in the range of 77 to 100 and 1135 to 1385, respectively (i.e. for comparison, pure CH₄ has MN = 108, and WI = 1357). Thus, Lyons *et al.* [19] concluded that the actual NG vehicles are being operating with might significantly differ from fuel used for engine certification (i.e. CARB Cert-fuel requires CH₄ fraction of 90.0±1.0 mole-%, equivalent to MN = 86).

Studies [19, 20, and 21] have reported NO_x, NMHC, and CO emissions to generally reduce as a function of increasing CH₄ content (i.e. higher MN) in the NG for dedicated lean-burn natural gas engines. However, for stoichiometric operating natural gas engines the same studies [19, 21] could not find a statistically significant trend in NO_x emissions due to changing fuel methane content. For lean-burn combustion, Karavalakis *et al.* [20] concluded the increase in longer chain HC (e.g. ethane and propene) in NG fuel to lead to increased adiabatic flame temperatures, and thus, resulting in higher NO_x. On the other hand, THC [19], CH₄, and formaldehyde [19, 20] were observed to increase as a function of increasing methane number for lean-burn engines, whereas no clear trend was found for THC [21] for stoichiometric operating engines. Particulate matter emissions did not show to be significantly affected by MN for both lean-burn and stoichiometric operating NG engines [19, 20, and 21].

Table 1. US-EPA CO₂ emissions standards for compression ignition engines for medium and heavy heavy-duty vocational (measured over FTP) and tractor (measured over SET) applications in [g/bhp-hr] [23].

Model year	Medium heavy-duty Vocational	Heavy heavy-duty Vocational	Medium heavy-duty Tractor	Heavy heavy-duty Tractor
2014-2016	600	567	502	475
2017 and later	576	555	487	460

With the heavy-duty vehicle fleet being identified as the second largest contributor to greenhouse gas (GHG) emissions among mobile source emissions within the United States [22], the USEPA and National Highway Traffic Safety Administration (NHTSA) established GHG and fuel efficiency standards for medium and

heavy-duty engines and vehicles [22]. The CO₂ standards shown in Table 1 are being introduced in a two tier process starting with model year (MY) 2014-2016 and 2017 and later engines, respectively [22, 23]. In addition to CO₂, the US-EPA also considers pollutants of CH₄ and N₂O as part of GHG emissions and defined a regulatory standard to limit both to 0.10 g/bhp-hr as measured over the transient Federal Test Procedure (FTP) engine certification cycle (i.e. applicable to MY 2014 and later CI engines) [23]. For medium and heavy heavy-duty engines that are certified for the primary use in tractors and other long-haul applications, GHG emissions are quantified over the steady-state Supplemental Emissions Test (SET) cycle, whereas engines that will be used for both tractor and vocational applications GHG emissions need to be measured over both SET and transient FTP cycle [23]. A more detailed discussion about the specifics of the US-EPA GHG regulation is given elsewhere [24].

Natural gas derived from fossil sources produces on the order of 20-30% less GHG emissions compared to gasoline or diesel fuel combustion, whereas biogas has the potential effectively reduce GHG emissions to near zero depending on feed stock and production process [6]. Indeed, the well-to-wheel GHG production strongly depends on gas production methods, modes of transportation and distribution, actual combustion efficiency as well as methane leakage rates during storage, transport, vehicle refueling and on-board storage [6, 25, and 26]. “Boil-off” and the resulting venting of natural gas vapors from cryogenic LNG tanks (i.e. in order to maintain constant pressure in the storage vessel due to ambient heat input), both at refueling stations and from vehicle on-board tanks, is of concern from a GHG emissions standpoint [6] as methane has a ~72 and ~25 times larger direct global warming potential (GWP) relative to CO₂ (i.e. CO₂eq) on a 20 and 100 year time period, respectively [27]. Furthermore, a recent study quantifying methane leakage rates from natural gas compressor stations and associated NG-fueled engines [25] reported methane loss rates between 4 and 22 kg/hr from five different compressor station sites.

The primary aim of this study is to improve understanding and quantify the impact of dual-fuel combustion using methane as the premixed fuel onto engine performance and emissions rates from legacy and modern after-treatment equipped diesel engines. Of special interest were SCR NO_x conversion efficiencies as a function of methane substitution rates. Furthermore, special attention was directed at tank-to-wheel greenhouse gas emissions and global warming impact relative to diesel-only operation for three different combinations of engine and dual-fuel conversion kits.

EXPERIMENTAL METHODOLOGY

All measurements performed for the study presented herein were conducted at the Engine and Emission Research Laboratory (EERL) at West Virginia University. The EERL is part of West Virginia University's Center for Alternative Fuels, Engines and Emissions (CAFEE) and the transient engine dynamometer test cell and associated emissions quantification instruments are designed and operated according to recommendations outlined in the Code of Federal Regulations (CFR), Title 40, Part 1065 [28]. This chapter will



summarize details of the evaluated test engines and NG ‘retrofit kits,’ engine dynamometer setup and laboratory instrumentation as well as emissions sampling systems.

Test Engines

In order to understand the impact of natural gas retrofitting onto engine performance and emissions rates as a function of engine and after-treatment technology, engine dynamometer testing was conducted on two pre-after-treatment technology and one state-of-the-art engine equipped with an advanced after-treatment system. The two legacy engines without after-treatment included a 12.8L Mercedes Benz MBE 4000 OM4606, and a 12.0L Mack AC-460P engine. Both were model year (MY) 2005 engines equipped with a high pressure cooled exhaust gas recirculation system (C-EGR), fuel direct injection with electronic unit injectors (EUI), variable geometry turbocharger (VGT), and were compliant with US-EPA 2004 heavy-duty emissions standards (i.e. NMHC+NO_x: 2.5 g/bhp-hr, PM: 0.1 g/bhp-hr).

Table 2. Test engine specifications

Engine	1	2	3
Manufacturer	Mercedes-Benz	Mack	Mack
Model	MBE 4000 OM4606	AC - 460P	MP8 - 505C
Model Year	2005	2005	2011
After-treatment System	None	None	DOC / DPF / urea-SCR
Displacement [L]	12.8	12.0	12.8
Bore x Stroke [mm]	128 x 166	124 x 165	131 x 158
Compression Ratio	17.75:1	16:1	16:1
Maximum Torque	1381 ft-lb @ 1080 rpm	1660ft-lb @	1810 ft-lb @ 1100 rpm
Maximum Power	386 hp @ 1986 rpm	480 hp @	505 hp @ 1800 rpm
Turbocharger	VGT	VGT	VGT
Injection System	EUI (2000 bar)	EUI	EUI (2400 bar)
EGR	High pressure cooled EGR	High pressure cooled EGR	High pressure cooled EGR
Emissions Certification	US-EPA 2004 NO _x /PM: 2.4/ 0.1 g/bhp-hr	US-EPA 2004 NO _x /PM: 2.4/ 0.1 g/bhp-hr	US-EPA 2010 NO _x /PM: 0.2/ 0.01 g/bhp-hr
NG Retrofit Kit	1	1	2
NG Injection Location	Sequential Port fuel injection	Sequential Port fuel injection	Pre-compr. intake fumigation

A model year 2011, 12.8L Mack MP8-505C engine was selected for testing as a representative candidate of current technology while additionally being an engine widely employed for Class-8 vehicle applications [29]. The engine was compliant with US-EPA 2010 criteria pollutant emission standards (i.e. NO_x: 0.2 g/bhp-hr, PM: 0.01 g/bhp-hr) and was equipped with a diesel oxidation catalyst (DOC), catalyzed wall-flow type diesel particulate filter (DPF), and aqueous-urea (i.e. diesel exhaust fluid (DEF)) based selective

catalytic reduction (SCR) system for NO_x abatement. Similarly to the MY 2005 engines, the Mack MP8 also featured a C-EGR system, electronic unit injectors, and VGT. Detailed specifications for the three test engines can be found in Table 2. For simplicity, the Mercedes-Benz, Mack AC, and Mack MP8 will be referred to as ‘Engine 1,’ ‘Engine 2,’ and ‘Engine 3,’ respectively, for any discussion of results hereinafter.

The tests were conducted using commercially available ultra low sulfur diesel (ULSD) fuel with a maximum sulfur content of 15 ppm. The CNG used for dual-fuel operation during this study was substituted with high purity (i.e. >99.5%) methane from compressed gas cylinders (i.e. Matheson Chemically Pure CH₄) in order to minimize the effects of varying natural gas composition and their ultimate impact on combustion characteristics and resulting emissions [19, 20, and 21]. The use of high purity methane was justified by the fact that the primary objective of this study was to investigate the impact of retrofitting different technology engines with dual-fuel conversion kits onto engine performance and emissions rates. Thus, by selecting a common fuel it was possible to ‘block-out’ any secondary influence of changing fuel properties. This was further important since testing of the three engines for this study extended over a time period of one year, hence, quality and composition of locally available NG might have undergone significant variations. Furthermore, the practice of using pure methane for investigation of dual-fuel combustion has been applied by other researchers [13, 30, and 31].

Dual-Fuel Conversion Kits

The primary purpose of dual-fuel conversion kits is the ability to retrofit a given diesel engine, with minimal to no alterations of the existing engine hardware and control strategy [6, 13], in order to allow for partial utilization of an alternative fuel source (e.g. natural gas) for engine operation that ideally results in reduced exhaust emissions, and more importantly, fuel and thereby cost savings for the vehicle/fleet operator. Additionally, the advantage of traditional dual-fuel conversion kits is that the engine could be easily switched to diesel-only operation in case of unavailability of natural gas [6, 13].

Two different dual-fuel conversion ‘retrofit kits’ were investigated during this study, namely, i) one with sequential port fuel injection (PFI) of natural gas (i.e. NG retrofit kit 1), and ii) one with natural gas being fumigated into the intake pipe at a pre-compressor location (i.e. NG retrofit kit 2) as depicted in Figure 1. The intake air fumigation-type kit 2 was applied to the after-treatment equipped Engine 3 (see numerical marker 3 in Figure 1), whereas the same PFI kit 1 was retrofitted to test Engines 1 and 2 (see numerical markers 1 and 2 in Figure 1).

The PFI kit comprises a NG injection rail that is placed between the existing intake manifold and cylinder head and houses two solenoid-activated injector nozzles per cylinder, and a NG controller unit. Additionally, two knock sensors are placed on the side of the engine block, one measuring audible knock from the rearward three cylinders (i.e. 4, 5, 6) and the other from the forward three cylinders (i.e. 1, 2, and 3). The NG controller unit determines the natural gas substitution rates based on manifold air pressure (MAP), manifold air

temperature (MAT), CNG temperature and a set of individually (i.e. for a given engine family) calibrated lookup tables (i.e. fueling maps). If audible knocking occurs, the substitution is immediately reduced. Additionally, the NG controller unit is connected between the OEM ECU and diesel fuel injectors allowing the NG controller to actively alter the OEM ECU diesel fuel injection duration. This is done in order to reduce the amount of diesel being injected with respect to the amount of CNG addition to the intake mixture on a fuel energy equivalent basis. The OEM ECU control over all other engine operating parameters, including exhaust gas recirculation and variable geometry turbocharger, are not being actively altered by the NG controller. A PFI-based dual-fuel system allows for higher substitution rates and faster cycle-to-cycle adjustment as natural gas is injected close to the intake port of the combustion chamber, thus drastically reducing the response time in comparison to a pre-compressor intake fumigation system such as 'retrofit kit' 2.

The second 'retrofit kit' under investigation comprises a natural gas fumigation module installed upstream of the compressor and an electronic control module (ECM). The ECM accesses publicly broadcasted controller area network (CAN) parameters as input to its controller algorithms and lookup tables (i.e. calibrated maps). Natural gas is added to the intake air as a function of engine parameters with no addition (i.e. 100% diesel operation) during engine 'cold' start and idle conditions. This 'retrofit kit' was designed as an entirely passive retrofit system that relied on the OEM ECU to reduce diesel fueling as a function of added energy to the intake air due to NG injection. No active alteration of original engine control parameters is being performed. The primary advantage of this system is in its simplicity regarding installation to the engine. However, the relatively long lag-time between injection of a quantity of NG and it being introduced into the combustion chamber poses difficulties for transient response and can additionally lead to large quantities of NG slipping through the engine when the throttle is being rapidly released (i.e. switching from load to no load demand).

Laboratory Measurement Setup

Engine testing for all three engines was conducted on an 800 hp General Electric® (GE) motoring/absorbing dynamometer capable of performing both steady-state and transient test cycles up to speeds of 2500rpm. The engine and after-treatment system (i.e. for *Engine 3*) were instrumented with K-type thermocouples and pressure transducers (Omega, Inc.) to quantify the thermodynamic states of different engine components and flow streams (i.e. exhaust, EGR flow, intake flow, coolant, etc).

Instantaneous diesel fuel flow measurements were performed for both diesel-only and dual-fuel operation using an AVL Fuel Mass Flow Meter (model 7351CST), operating based on the Coriolis principle, in conjunction with an AVL Temperature Control Unit (model 753CH). The fuel flow meter is capable of flow and density measurements with an accuracy of 0.12% (pt.). In addition to the AVL fuel flow measurement, fueling rates broadcasted via CAN by the ECU were also recorded. The flow rate of methane was measured by an Endress-Hauser Gas Flow Meter (model 8DFO8-AG15AAAABAON, max. error: $\pm 0.50\%$ of value) based on the Coriolis principle. Furthermore, carbon balance calculations based on recovered carbon fractions in the exhaust stream (i.e. CO_2 , CO, THC) were performed to corroborate direct fuel measurement.

Regulated gaseous emissions, including total hydrocarbons (THC), carbon monoxide (CO), oxides of nitrogen (NO_x) and particulate matter (PM), as well as carbon dioxide (CO_2), nitric oxide (NO) and methane (CH_4) were measured on a diluted basis using a full-flow subsonic venturi (SSV) constant volume sampling (CVS) system coupled with a variable speed blower. Gaseous emissions sampling was performed on a continuous basis using a Horiba® MEXA 7200D Automotive Emissions Analyzer System; with CO and CO_2 being detected by the non-dispersive infrared (NDIR) method (cold dry sample), NO_x (heated dry atmospheric sample) and NO (cold dry atmospheric sample) using the chemiluminescence (CLD) method; THC and CH_4 using heated-flame ionization detectors (HFID) with the latter being routed through a non-methane cutter before entering the HFID. All sample lines and probes were heated to prevent water condensation prior to analyzer inlet. Additionally, raw exhaust NO_x concentrations were measured downstream the turbocharger using a Horiba MEXA-720 zirconium-oxide sensor for the after-treatment equipped *Engine 3*, allowing to calculate real-time SCR NO_x conversion efficiencies.

PM characterization was performed using the gravimetric method as outlined in 40 CFR, Part 1065. A slip stream of diluted exhaust gas was sampled from the main dilution tunnel through a secondary dilution system and subsequently sampled onto 47 mm Pallflex® quartz-fiber filter (TX40) membranes (Pall Corporation). An environmentally controlled clean room (Class 1000, maintained at 21°C and 50% RH) was utilized for PM filter media handling, storage and weighing using a Sartorius microbalance with an accuracy of 0.1 µg.

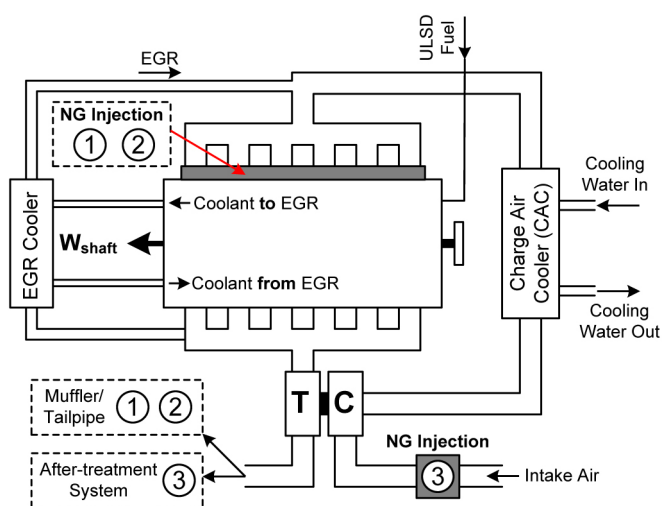


Figure 1. Schematic of natural gas injection systems. Numerical markers (i.e. 1-3) indicate location of NG addition for a given test engine.

Test Plan

Gaseous and particle phase emissions measurements were performed by operating the three test engines, retrofitted with the respective dual-fuel conversion kits, over steady-state and transient test cycles. Specifically, *Engine 1* and *2* were exercised over the 14 modes of the SET and the FTP transient heavy-duty engine certification test cycle [32], whereas *Engine 3* was tested over the 14 modes ramped modal cycle (RMC) [32] and FTP cycle. The RMC essentially utilizes the same engine speed/load set-points as prescribed by the SET with the difference that mode transition is part of data collection as opposed to the strictly modal data collection for the SET. For the purpose of results discussion in this study however, the ramped portions of the RMC were neglected and emissions rates were solely calculated for the stabilized portions of each mode. Thus, SET will be used to refer to the steady-state test for all three engines hereinafter.

Table 3 outlines the test matrix followed for all three engines and two ‘retrofit kits’ during this study. All results presented and discussed hereinafter will follow the same order in terms of engines (i.e. Engine 1, 2, 3 data). Each engine was tested first on diesel-only operation to establish a baseline reference and was subsequently operated in dual-fuel mode to compare the results. Between each hot-start test, a 20-minute ignition off soak period was observed allowing for the analyzers to be zeroed and spanned, the PM filter to be changed, and the engine/after-treatment system to reach a comparable thermodynamic state for each test. Each test sequence was initiated with a cold start test (i.e. SET or FTP depending on sequence) in order for the engine/after-treatment system to reach operating temperatures.

Table 3. Engine dynamometer test schedule

Engine	Retrofit Kit	Diesel mode	Dual-fuel mode
1	1	FTP / SET	FTP / SET
2	1	FTP / SET	FTP / SET
3	2	FTP / SET	FTP / SET

Emissions factors of CO₂, CO, NO_x, methane hydrocarbons (MHC) and non-methane hydrocarbons (NMHC) will be presented on a work-specific basis (i.e. g/bhp-hr) as integrated values for the transient test (i.e. FTP cycle) and as modal data for the steady-state tests (i.e. SET). All presented emissions data, unless otherwise specified, are based on diluted CVS measurements.

RESULTS AND DISCUSSION

The dual-fuel conversion ‘retrofit kits’ evaluated in this study were individually calibrated for the given engine platform/family they were retrofitted to. As described earlier, retrofit kit 1 involved active ‘interception’ and adjustment of the diesel injector signals by the NG controller unit, whereas retrofit kit 2 was a purely passive NG addition system relying on the OEM ECU to reduce throttle demand to the engine, and thereby ultimately reducing diesel fueling due to the added energy available in the intake air/methane mixture. Both approaches pose a unique challenge to the OEM engine control of modern, electronically controlled engines, and will influence the operating characteristics of control parameters, including exhaust gas

recirculation, VGT, load-specific injection timing and even multiple injection events. Control of these parameters heavily relies on look-up table (i.e. maps) based control algorithms that are predominantly a function of engine speed and torque. However, the engine torque parameter used in ECUs is a computed parameter that is based on diesel fuel injection quantities into the engine, which in turn are obtained from integration of modulated injector pulse-width along with the physical state of the fuel (i.e. pressure, temperature, etc.). Thus, if the actual fueling of the engine is altered due to a change in throttle demand, caused by the addition of natural gas in the intake, torque values might be ‘wrongly’ computed ultimately leading to changes in control parameters such as EGR, VGT, etc. as the erroneously estimated torque value will cause a shift within the respective look-up tables.

In order to reduce the impact of alterations to the OEM control parameters and maintain engine performance and ultimately vehicle drivability comparable to diesel-only operation, traditional ‘add-on’ conversion kits are not necessarily able to fully maximize the natural gas usage as could be achieved by a dedicated dual-fuel engine (i.e. OEM system) with the engine control parameters being specifically optimized for dual-fuel operation. Furthermore, installation of dual-fuel ‘retrofit kits’ on post 2010 model year engines with advanced after-treatment and on-board diagnostic (OBD) systems might pose additional challenges for seamless integration. In particular, addition of NG might slow the completion rate of OBD monitors, and thus alter the in-use monitoring performance ratio (IUMPR) [33] possibly triggering fault codes and leading to the illumination of the engine malfunction indicator lamp (MIL). Various approaches are suggested in order to prevent monitor enable conditions, including restriction of natural gas substitution to limited areas within the full engine operating range (i.e. centered around medium engine speed/load operation) or intermittent termination of NG addition allowing for OBD monitor completion [33].

Even though the majority of dual-fuel ‘retrofit kits’ are applied to outside-useful-life (OUL) and intermediate-age engines that are characterized by reduced certification requirements, integration of secondary controllers for dual-fuel conversion are likely to disrupt OBD performance [34]. Functionality and more importantly, accuracy of monitors for the air handling and fueling system, EGR control as well as after-treatment systems are expected to be impacted, and separate OBD strategies and calibrations might be required for retrofit systems [34]. Furthermore, with the introduction of GHG regulations [22] it is possible that future OBD requirements (i.e. beyond 2018) for example, will include monitoring of CH₄ emissions and conversion efficiencies of methane mitigation systems (i.e. catalysts). OEM involvement or cooperation is believed to be necessary, however, to fully maximize the potential benefits of natural gas dual-fuel conversion while complying with full OBD requirements as applicable by 2018 [33, 34].

Development and calibration (including population of look-up tables) of the NG controllers used in conjunction with the ‘retrofit kits’ presented in this study heavily rely on steady-state optimization of selected engine speed/load modes underneath the engine torque curve with additional ‘fine tuning’ for behavior and response over transient operation. Ultimately, the dual-fuel systems will be operated over