

Study of dual-fuel (diesel + natural gas) particle matter and CO₂ emissions of a heavy-duty diesel engine during transient operation

The aim of this study is to describe the impact on particle matter and CO₂ emissions of converting an existing heavy-duty diesel engine for on-highway truck applications to a dual-fuel engine (diesel + natural gas), especially in transient operation. A dual-fuel engine with homogeneous gas charge injection in the intake line before turbocharger was considered. The results showed the feasibility of this kind of technology for transient operation reaching a significant reduction of particle matter plus a decrement in CO₂ emissions at the expense of a small decrement of brake fuel conversion efficiency and an increment of unburned hydrocarbons in the exhaust gases.

Key words: dual-fuel, natural gas, diesel engine, heavy-duty, transient, particle matter, CO₂

1. Introduction

Nowadays there is significant interest in converting diesel to NG for heavy-duty engines used in commercial vehicle applications due to the growing availability of NG in Europe which opens the way to using it in long-distance transport. This also reduces Europe's dependence on liquid oil fuels and the risk for the future stability of supply resulting from a significant price increase. Another important advantage is that the use of NG can be favorable to decreasing or mitigating CO₂ emissions and has potential for reducing toxic exhaust specific emissions such as smoke and PM.

The mentioned diesel to NG engine conversion can be total or partially performed. In the case of total conversion, the combustion system is changed in order to work from diesel to otto-cycle; this conversion can include several modifications such as changing the combustion chamber in order to obtain a new compression ratio and introducing spark plugs. The technical process and the main technological limits that apply to existing European diesel engine for truck application currently used in long-distance transport converted to a dedicated NG engine is widely studied in references [1, 2].

In the case of partial engine conversion heavy-duty dual fuel (HDDF) engine is used [3–7]. A HDDF engine is conceived to simultaneously operate with diesel fuel and a gaseous fuel, both fuels being metered separately, where the consumed amount of one of the fuels relative to the other one may vary depending on the operation. Two types of technology are considered according to the way of gas injection: High Pressure Direct Injection (HPDI) and the Homogeneous Gas Charge Injection (HGCI). In the first one diesel and NG are injected directly into the combustion chamber at higher injection pressure using a special injector with a dual-concentric needle design or by using two separate injectors [8, 9]. This allows for small pilot quantities of diesel fuel and large quantities of NG to be delivered at high pressure to the combustion chamber. The major advantages of this type of HDDF engines are that there is no limitation by knocking at

high loads, their HC emissions are low and they can replace more than 90 % of the diesel fuel (by energy). The main disadvantages for this kind of technology are the introduction of the new injection system which can lead to some important engine cylinder head modifications for retrofitted engines, a direct communication with OEM controller software could be necessary and that such engines only work with LNG, which means a cryogenic system should be installed. This technology is outside the scope of this work.

In the case of HGCI, gaseous fuel is injected into the intake line of the engine (Fig. 1) and premixed with air or EGR during the intake and compression stroke. Ignition of the charge is managed by injection and auto ignition of diesel fuel using the original injection system of the baseline engine. Substitution rates of 50 % to 75 % can be achieved depending on several factors including duty cycle and the level of integration with the base diesel. The most important advantages of this technology are the simplified engine control with no direct communication with OEM, the non-intrusive technical modifications in the engine structure, the fact that it can be used with CNG and LNG indistinctively, which is a key factor in places where gas infrastructure and availability play an important role, and the low cost for the implementation in retrofitted engines with a considerable time reduction during development phase comparing to HPDI. The main disadvantages of HGCI technology are that the upper load range is typically limited by knocking, higher CO and HC emissions which lead to using methane catalyst in the exhaust pipeline and that the transient response of the engine could deteriorate as substitution rates increase especially during transient operation.

Bearing in mind the benefits and drawbacks for HDDF engine operation the objective of this work is established. The aim of the present work is to study the impact in PM and CO₂ emissions of converting existing HD diesel engine for on-highway truck applications to a HDDF engine when Homogeneous Gas Charge Injection in the intake line before turbocharger is used.

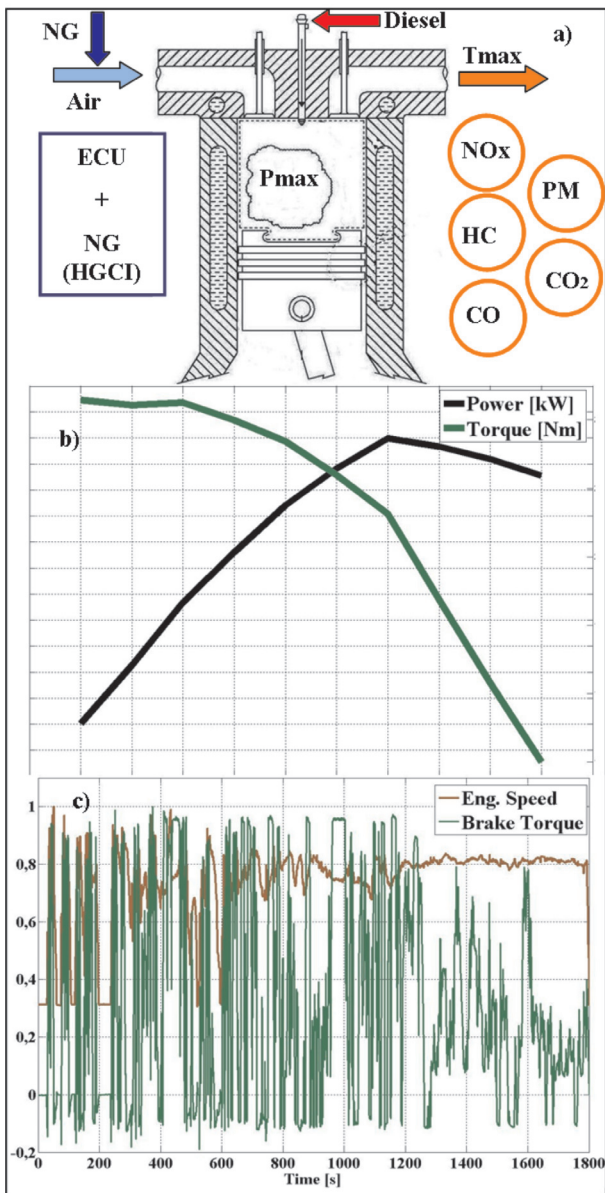


Fig. 1. Hddf engine concept (HGCI): a) emissions and thermodynamics, b) performance, c) transient behaviour

2. Experimental set-up

The experimental tests were carried out in a turbocharged intercooled heavy-duty DI diesel engine for on-highway truck applications. The engine used in this work was a four-stroke, six cylinder diesel engine equipped with an EUI fuel injection system. Table 1 summarizes the main engine specifications. The engine was fully instrumented with fast response thermocouples and mean pressure sensors, which were placed in convenient zones. The temperature of all the fluids: inlet air, cooling water, lube oil, EGR and fuels were measured by PT100 and K-type thermocouples. In addition, in order to obtain the maximum in-cylinder pressure and the maximum inlet turbocharger turbine during diesel operation AVL Indimodul 621 indicating system and the exhaust manifold fully instrumented with temperature and pressure sensors were used respectively.

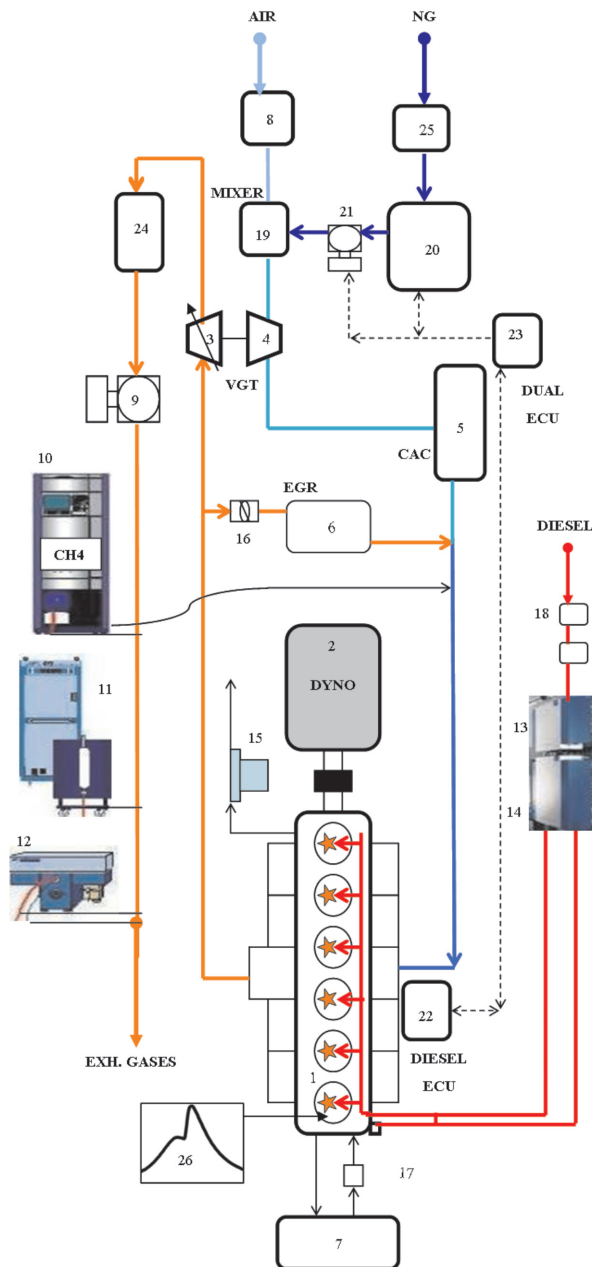
Table 1. Main engine characteristics

Type	Turbocharged – intercooled DI Diesel engine with Variable Geometric Turbine (VGT) and High Pressure cooled EGR
No Valves	4 each cylinder
No Cylinders	6 in-Line
Bore x Stroke	134 x 168 mm
Compression Ratio	16.8:1
Injection System	Electronic Unit Injector (EUI)
Power level	>340 kW @ 1800 rpm
Torque level	> 2000 Nm @ 1200 - 1500 rpm

The test cell was equipped with an active dynamometer with correspondingly maximum values of: power (450 kW), torque (2869 N·m) and speed (4500 rpm). The dynamometer is equipped with digital high-precision torque transducer HBM T12 and is linked with a PC via AVL PUMA OPEN 1.3 software to program and run the stationary and transient tests automatically. A scheme of the experimental set-up employed in this work is shown in Fig. 2.

Table 2. Main properties of the working fuels

Diesel properties		
Cetane number	52.8	[-]
Density @ 15° C	836	kg/m ³
Viscosity @ 40° C	2.891	mm ² /s
Sulphur	3.0	mg/kg
Carbon	85.91	% weight
Hydrogen	13.43	% weight
LHV	42.917	MJ/kg
NG properties		
Methane	92.0347	%Mole
Ethane	7.0087	%Mole
Propane	0.6620	%Mole
Butane	0.1052	%Mole
Pentane	0.0020	%Mole
Isobutane	0.0754	%Mole
Isopentane	0.0073	%Mole
Nitrogen	0.1047	%Mole
Carbon dioxide	0.0000	%Mole
LHV	10.5952	kWh/m ³
Density	0.7743	Kg/m ³



1	Diesel engine	14	AVL 753C fuel temp. control
2	Active dynamometer	15	AVL 442 blow-by meter
3	Turbocharger turbine (VGT)	16	EGR control valve
4	Turbocharger compressor	17	Coolant flow meter
5	Charge air cooler (CAC)	18	Fuel filters
6	EGR cooler	19	NG and air mixer
7	Engine coolant cooler	20	NG press regulator system
8	Air mass flow meter	21	NG throttle valve
9	Exhaust back pressure valve	22	Diesel ECU
10	Horiba 7500 DEGR + CH ₄ module	23	Dual ECU
11	Horiba MDLT 1300 T	24	Methane catalyst
12	AVL 439 opacimeter	25	NG measurement device
13	AVL 735S fuel mass flow meter	26	AVL indimodule

Fig. 2. General schema of the engine installation at the test cell

The test cell can work with different types of fuels such as gasoline RON 95/98, diesel, NG, LPG, reference and special

fuels (leaded gasoline for example). Major characteristics of the diesel and NG fuels used in this work are shown in Table 2. Diesel fuel consumption was measured by AVL 735S continuous measurement system (Coriolis principle) and diesel fuel temperature control was achieved by AVL 753C equipment.

NG fuel consumption was measured by a BRONKHORST gas mass flow meter with a measurement range of 1–150 kg/h. The main parts of the HGCI system are: an air-NG mixer, a gas throttle electric valve and the NG pressure regulator system (a shut-off electric valve, a high pressure limiter and a low pressure regulator). These components are continuously communicated with dual ECU.

The intake air consumption was measured by Sensyflow P thermal mass flow meter with a measurement range of 0–4000 kg/h. The test cell is also equipped with humidity and intake air temperature control system.

For the measurement of the typical gaseous emissions (NO_x, CO₂, CO, HC) exhaust gas analyzer HORIBA 7500DEGR was used which was extended with a CH₄ module for its measurement during dual-fuel mode. It is important to mention that methane catalyst was only installed after all the diesel baseline tests were done.

To construct the 1D model for diesel baseline engine which is explained in the next section, CO₂ in the intake manifold was also measured to calculate EGR rate according to equation (1) where the ambient [CO₂]_{atm} concentration remains constant at 0.04 %.

$$EGR = \frac{[CO_2]_{intake} - [CO_2]_{atm}}{[CO_2]_{exhaust} - [CO_2]_{atm}} \quad (1)$$

The PM specific emission during standard emission cycles was measured by micro dilution tunnel HORIBA MDLT-1300T. However, for HDDF engine calibration tasks in the test bench a partial flow Opacimeter AVL 439 was used to measure the smoke opacity of a sample of the exhaust flow. The opacity values obtained from diesel baseline engine were used as a reference.

HDDF engine operation was controlled by means of IDIADA engineered dual ECU. The management of the Diesel quantity by controlling diesel ECU and the gaseous injected quantity by controlling gas throttle electric valve operation is done by this controller. This ECU also performs monitoring tasks during diesel and dual-fuel engine operation through CAN bus J1939. The Dual ECU was also tested in an automatic HiL test bench for simulation of the hardware environment and to put it through all the possible scenarios it might experience in real world environment. Several tests according to ISO16750 were also performed.

3. Methodology

To reach the objective of the present work two major activities were planned as stated in Table 3: the first of them was to perform dual-fuel engine calibration which comprises a virtual (1D model) and test bench calibration in order to have the final calibration maps for the dual

ECU. The second major activity was to develop worldwide standard emission cycles especially in transient operation to check engine behavior and compare brake fuel conversion efficiency, energy distribution and exhaust specific emissions values with those previously obtained in diesel fuel operation only.

3.1. Calibration activities

The main target of virtual calibration based on 1D simulation model was to obtain a baseline calibration for the engine in dual-fuel mode for a quicker and less expensive development process. It is important to mention that this is a feedback loop process as can be seen in Fig. 3.

First, the 1D model was constructed using GT-power software which is the industry-standard engine simulation tool. This tool takes account of wave dynamic manifold phenomena via robust solution of the Navier-Stokes equations and the transient performance response of the engine that could be an important issue when HGCI technology is used. After that, the model was extensively validated in diesel mode with experimental data from the engine test bench.

Once the model was validated the next step was to simulate and optimize strategies and control devices functioning

in dual-fuel mode. In this work due to the engine original technology (EGR + VGT) the EGR deactivation strategy when the engine works in dual-fuel mode was selected in order to have a simple electronic control by dual ECU. This strategy was extensively tested in the 1D model before it was finally on the test bench.

Table 3. Experimental matrix

Tasks	Subtasks	Dual-fuel Mode	Target
Calibration tasks	Virtual calibration (1D model)	Engine mapping (no idle)	Fast engine development EGR deactivation Gas throttle control Baseline calibration
	Test bench calibration		Diesel replacement ratio Stationary calibration Transient adjustments Final calibration maps
Emission cycles	ESC	9 modes (25-75% Load)	Dual transitions PM
	FTP	Crowded freeway (300 s)	Transient behaviour CO ₂ & PM Brake fuel conversion efficiency
	ETC	Motorway (300 s)	Transient behaviour Fuel energy distribution

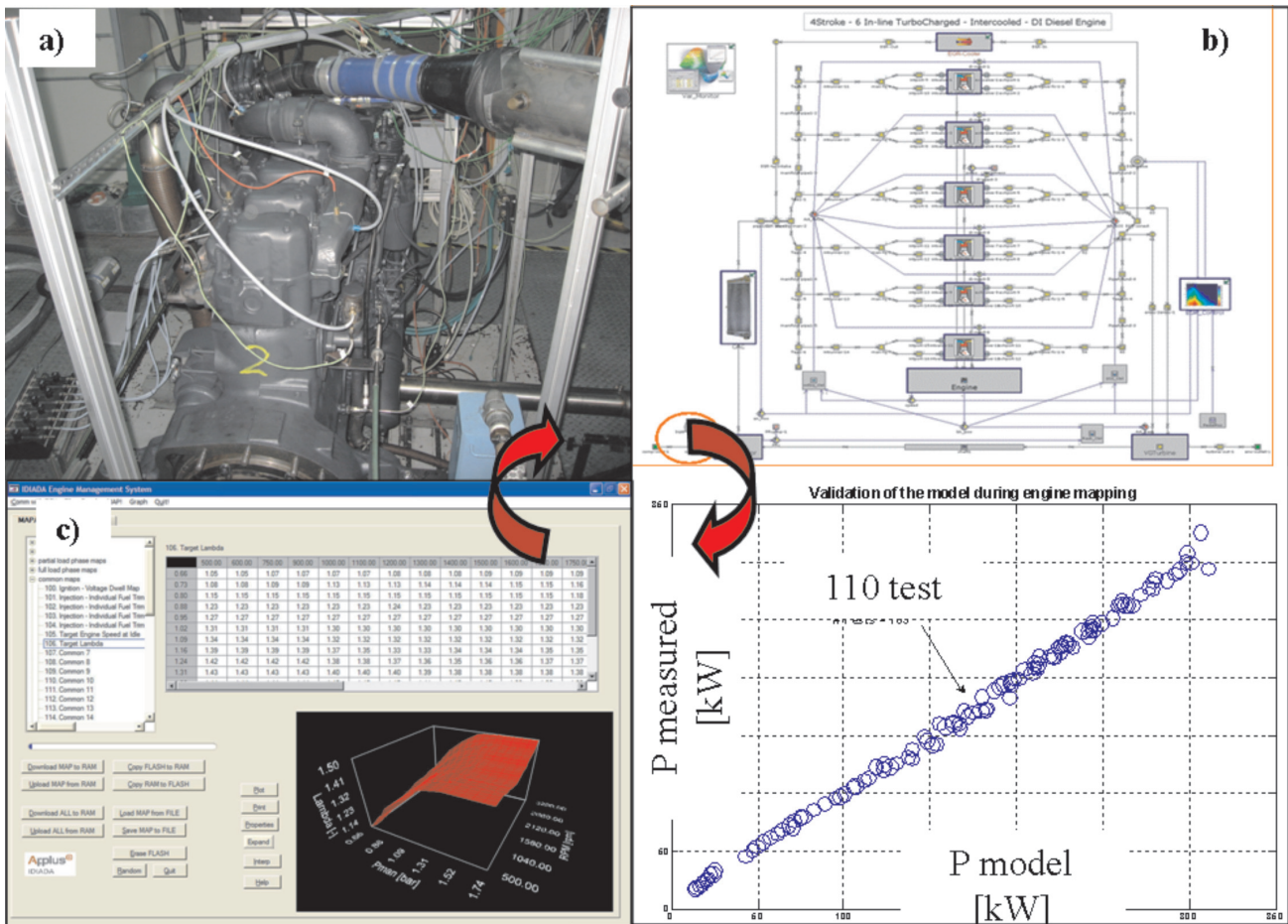


Fig. 3. General methodology for engine calibration: a) testing, b) modelling and validation, c) baseline calibration

Some other important targets reached by means of the 1D engine model were to infer the engine dynamic behavior in dual-fuel mode testing several substitution rates via Simulink coupling (Fig. 4a) and to select an optimized gas throttle electric valve diameter (Fig 4b). The response of this valve was tested in different stationary and transient scenarios in the model or even in extreme conditions such as when the engine is suddenly forced to leave dual-fuel mode to diesel operation or vice versa.

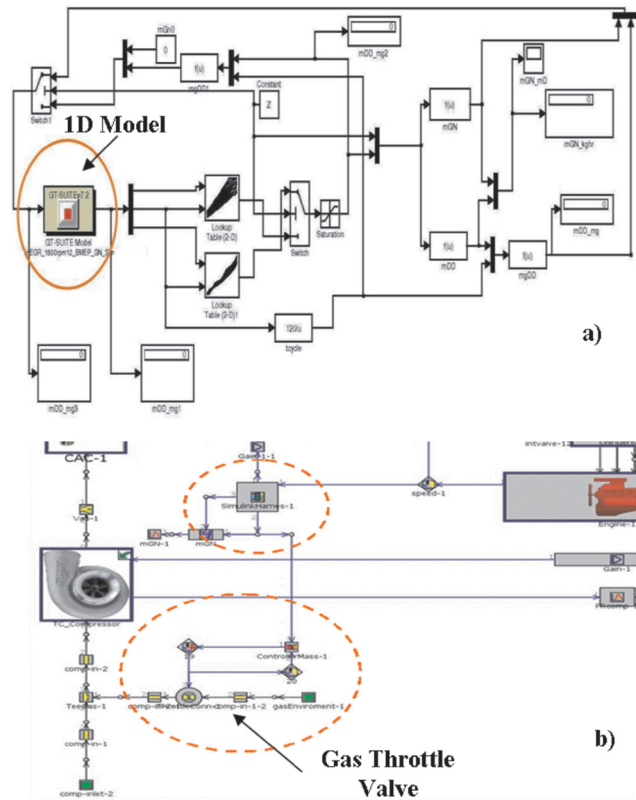


Fig. 4. Modelling dual-fuel mode operation: a) dynamic control with Simulink, b) natural gas throttle valve operation

On the other hand, the test bench calibration activities were done based on the previous baseline calibration from engine model. This baseline calibration was downloaded to dual ECU (Fig. 5) and each point was slightly and conveniently modified at the test bench to match engine performance and emissions.

The stationary calibration was performed in order to reach the maximum Gas Energy Ratio (GER) until the engine's thermodynamic limits (Fig. 6a). Examples of these restrictions were maximum in-cylinder pressure (Fig. 6b), maximum rate of pressure rise and maximum temperature measured at inlet turbocharger turbine. Final dual-fuel mode calibration maps were achieved after the implementation of small adjustments via electronic gains and filters for engine transient operation. The calibration was broadly tested in steady state and transient conditions at the test bench to verify its functionality.

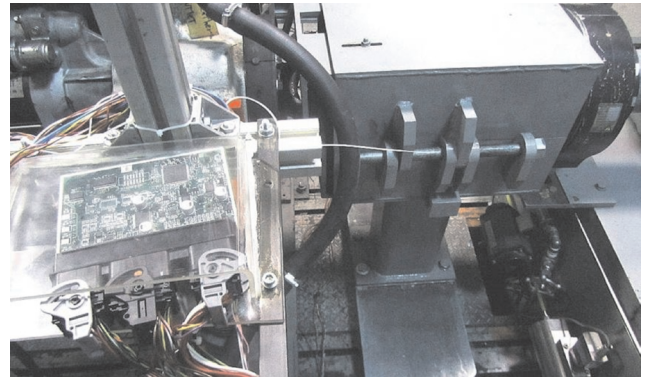


Fig. 5. IDIADA engineered dual ECU during test bench engine calibration

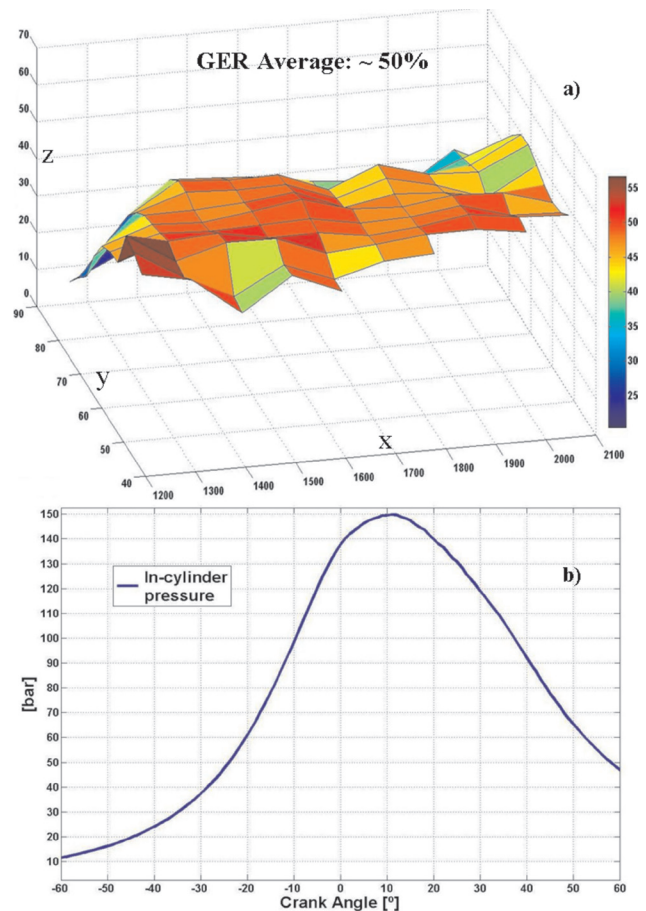


Fig. 6. Final calibration results: a) average GER, b) maximum in-cylinder pressure

Chemical energy supply, GER, and the brake fuel conversion efficiency in dual-fuel mode were calculated by equations (2), (3) and (4) respectively.

$$E_{Dual} = m_{NG} \cdot LHV_{NG} + m_{DD} \cdot LHV_D \quad (2)$$

$$GER = 100 \cdot \left(\frac{m_{NG} \cdot LHV_{NG}}{E_{Dual}} \right) \quad (3)$$

$$\eta_{Dual} = \frac{W_{Effective}}{E_{Dual}} \quad (4)$$

the total sets of its GHG emissions. Hence CH₄ emissions must be under control by methane catalyst.

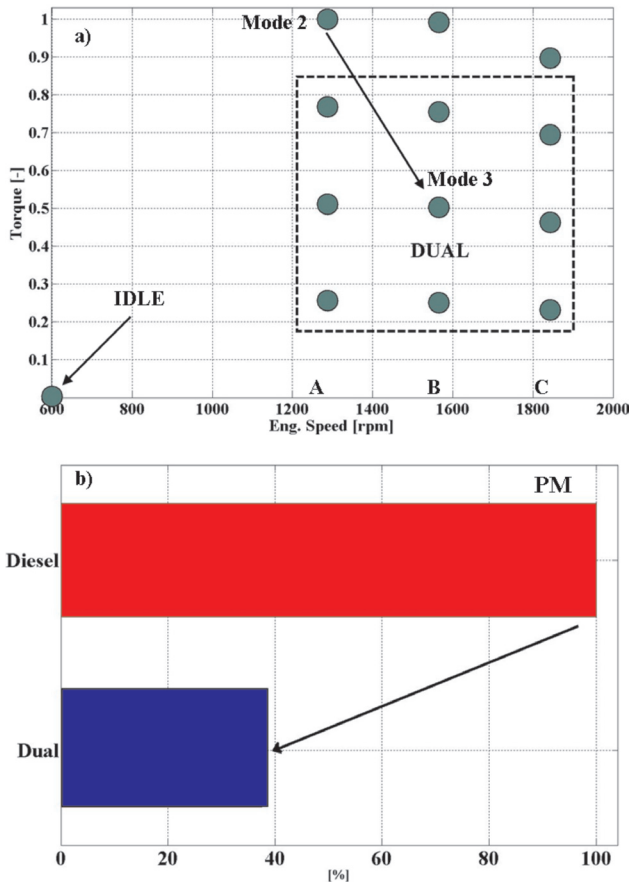


Fig. 8. ESC test: a) dual-fuel modes, b) PM comparison

Figure 10a shows the comparison of the results of NO_x, PM and CO₂ obtained after performing a complete sequence of FTP (cold + hot start) in diesel and dual-fuel mode. It can be seen from the figure that PM and CO₂ specific emissions decreased in dual-fuel mode 33.04 % and 3.64 % respectively. The reduction in CO₂ might have occurred because methane has very low C/H ratio which produces low CO₂ emission per energy content and the decrement in PM could be because of the short carbon chains present in the methane. On the other hand NO_x specific

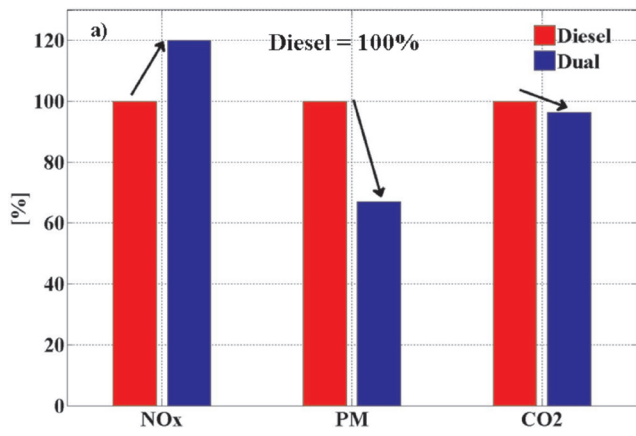


Fig. 10. FTP results comparison: a) NO_x, PM and CO₂ emissions, b) CO, NMHC emissions and fuel conversion efficiency

emissions increased 20.01 % in the test. Even though nitrogen content in the NG (Table 2) could have contributed to increase NO_x emissions, the authors consider that EGR deactivation strategy in dual-fuel mode might be the most influential factor for the result. This behavior is not expected in non-EGR diesel engines as it is reported in references [7, 11].

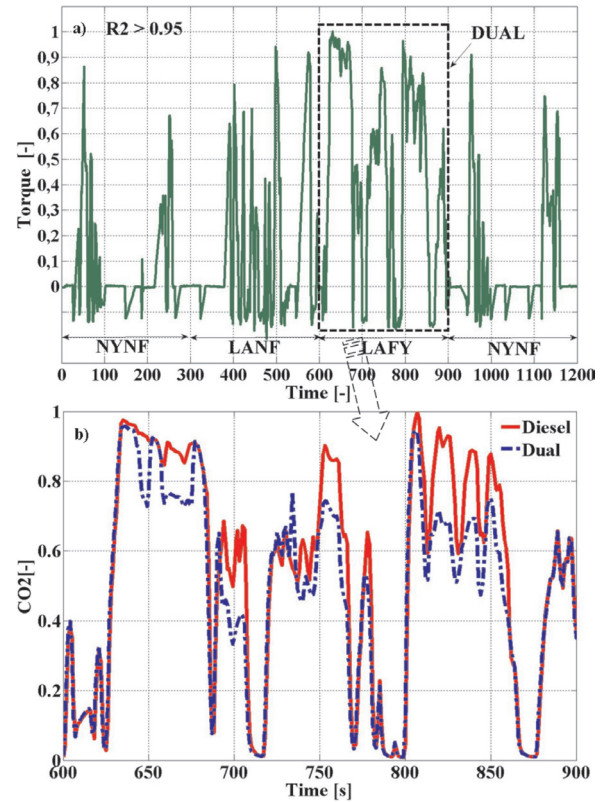
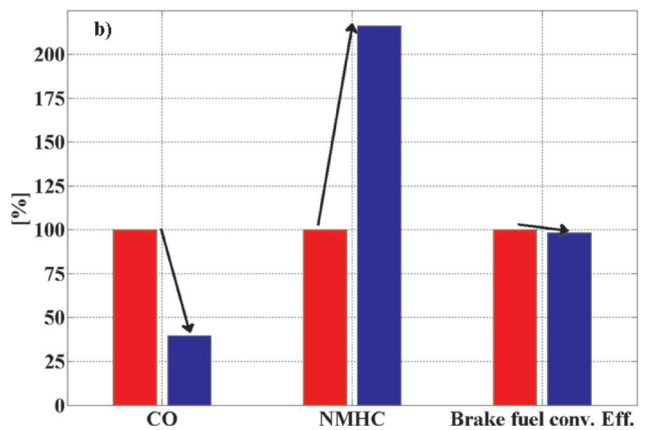


Fig. 9. FTP result: a) dual-fuel mode (crowded freeway), b) CO₂ comparison

Figure 10b shows the comparison of the results of CO, NMHC, and brake fuel conversion efficiency in the same condition mentioned in the above paragraph. The CO specific emissions decreased by 60.46 % but the NMHC increased by more than 100 %. The reduction of the CO was mainly due to methane catalyst. The relevant increment in NMHC could be improved by the implementation of the start of



diesel injection management in the dual ECU or selecting a better methane catalyst with a higher HC efficiency conversion for working in transient operation. On the other hand, the brake fuel conversion efficiency only suffered a small decrement of 1.69 % during the test.

4.3 Transient test: ETC

The transient operation of the engine working in dual-fuel mode during the last 300 s of the motorway part from ETC test is shown in Fig. 11a (dotted line). In that condition engine speed was around 1600 rpm, meanwhile engine torque changed from motoring up to almost 80 % of the load according to ETC demanded values. The engine’s thermodynamics, stability and good general behavior during the test demonstrate again the feasibility of this technology in transient operation.

Figure 11b shows the diesel and NG dynamic fuel consumption during the same last 300 seconds of the ETC test mentioned before. The figure pointed out the fast and coordinated response of the two fuels injected by the gas throttle electric valve and the diesel EUI which are governed by dual ECU to match engine fuel requirement during transient operation.

The exploded pie graph shown in Fig. 11b shows that 52 % of the total dual-fuel energy calculated by equation (2) is supplied by NG. This result corroborates the average GER around 50 % previously commented in the methodology section of this work (Fig. 6a) and also indicates the capability of the system for improving user economy during freeway-like transient conditions above all in places where diesel and NG prices have a substantial difference.

5. Conclusions

An experimental investigation was carried out in order to better understand the transient behavior of an existing heavy-duty diesel engine for on-highway truck applications when it is converted to a HDDF engine by means of the use of homogeneous natural gas injection in the intake line before turbocharger and the impact of the conversion on particle matter and CO₂ emissions. The most interesting aspects are remarked below.

- The feasibility of dual-fuel technology, especially the use of Homogeneous Gas Charge Injection in the intake line, was demonstrated during transient operation. Two worldwide standard cycles were used and in both of them dynamic behavior and transient engine response were matched to the original diesel baseline.

- The potential for PM specific emissions reduction working in dual-fuel mode was corroborated in stationary and transient engine operation. This result shows the capability of such engines to achieve a considerable decrement of one of the most current toxic contaminants in diesel engines.
- A decrement in CO₂ specific emissions in transient operation was also observed in dual-fuel mode compared with diesel only, which can contribute to diminishing the greenhouse emissions produced by this type of gas.
- The main drawbacks of this kind of technology found in this study were a considerable increment in NMHC specific emission and a small decrement of brake fuel conversion efficiency.
- NO_x specific emissions in dual-fuel mode were increased during transient operation mainly due to EGR deactivation strategy. This result is not expected for non-EGR HD diesel engines with or without SCR system.

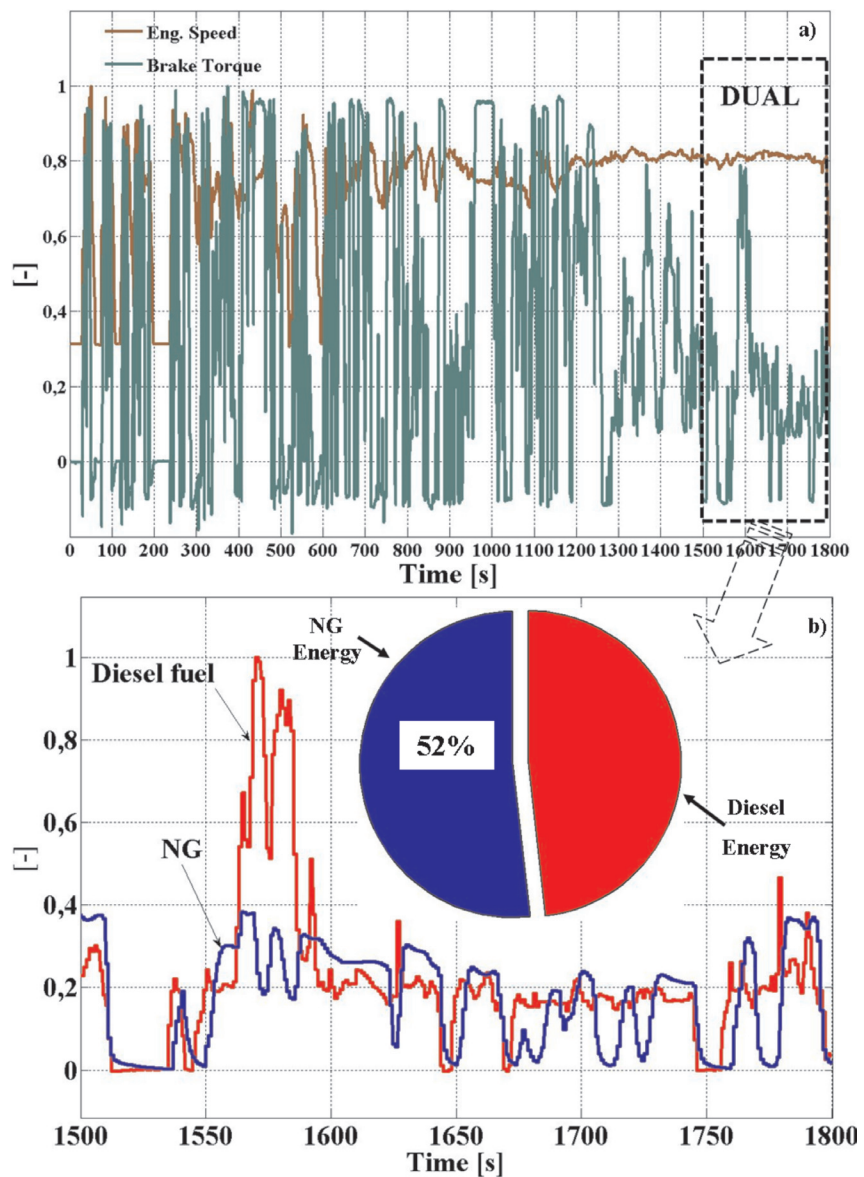


Fig. 11. ETC test: a) dual-fuel mode (motorway), b) NG and diesel energy distribution

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Nomenclature

CAN	Controller Area Network
CH ₄	Methane
C/H	Carbon Hydrogen ratio
CNG	Compressed Natural Gas
CO	Carbon monoxide
CO ₂	Carbon dioxide
DI	Direct Injection
ECU	Electronic Control Unit
EGR	Exhaust Gas Recirculation
EUI	Electronic Unit Injector

GHG	Greenhouse Gas
GUI	Graphical User Interface
HC	Unburned Hydrocarbons
HD	Heavy-Duty
HiL	Hardware in the Loop
LNG	Liquefied Natural Gas
LPG	Liquefied Petroleum Gas
NG	Natural Gas
NMHC	Non-Methane Unburned Hydrocarbons
NO _x	Nitrogen Oxides
OEM	Original Equipment Manufacturer
PM	Particle Matter
RON	Research Octane Number
SCR	Selective Catalytic Reduction
U.S.	United States
VGT	Variable Geometric Turbine

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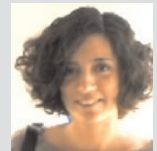
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