

Experimental Optimization of a Small Bore Natural Gas-Diesel Dual Fuel Engine with Direct Fuel Injection

Paul Fasching, Florian Sprenger, and Helmut Eichlseder
Graz University of Technology

ABSTRACT

Dual fuel combustion processes, which burn varying ratios of natural gas and diesel, are an attempt to reach high efficiencies similar to diesel engines while exploiting the CO₂ savings potential of natural gas. As shown in earlier studies, the main challenge of this combustion process is the high emission of unburned hydrocarbons during low load operation. Many publications have focused on a layout which utilizes port injection of natural gas and a direct injection of diesel to initiate combustion. However, previous studies indicated that a sequential direct injection of both fuels is more promising. It enables charge stratification of natural gas and air, whereby a remarkable reduction of the unburned hydrocarbon emissions was observed. This work develops this approach further, utilizing a low pressure direct injection of natural gas. The effects of main application parameters such as injection strategy, exhaust gas recirculation, air fuel equivalence ratio and the crank angle of 50 % fuel burned, etc. were investigated systematically on an engine dynamometer. For this purpose, an EU6 certified 4-cylinder diesel engine for passenger cars was modified. The measurements were conducted for two low load and two medium load operating points, which are particularly critical for the hydrocarbon emissions and characteristic for the NEDC test cycle. The aim was to optimize combustion in order to minimize the emission of unburned hydrocarbons. As a result, the total hydrocarbon emissions were reduced by 91 % to 2.2 g/kWh at the operating point $n = 1750 \text{ rpm} / \text{BMEP} = 5 \text{ bar}$. In this publication, the influence of the individual application parameters and their combined effect on the combustion is discussed.

CITATION: Fasching, P., Sprenger, F., and Eichlseder, H., "Experimental Optimization of a Small Bore Natural Gas-Diesel Dual Fuel Engine with Direct Fuel Injection," *SAE Int. J. Engines* 9(2):2016, doi:10.4271/2016-01-0783.

INTRODUCTION

The automotive industry is facing major challenges. With regard to the powertrain, the main motivations behind research are the tightening legislation of the pollutant emissions and the regulation of the carbon dioxide (CO₂) emissions in all key markets. CO₂ is the most important greenhouse gas (GHG) and a major driver of global warming. By limiting the CO₂ emissions, the authorities strive to limit the impact of transportation on our environment.

In Europe, after a phasing in over 3 years, the CO₂ output of new passenger cars has to be on average below 130 g CO₂/km in 2015 and below 95 g CO₂/km in 2020 [1] (which is equivalent to a consumption of approx. 4.0 l/100km or 59 mpg gasoline). In 2007 (2013), the average CO₂ output of new passenger cars across the EU was 159 (127) g CO₂/km [2]. While in Europe, the CO₂ targets are linked to the vehicle's weight, in the US the CO₂ limits are linked to the base area of each vehicle. The test cycles are also different (NEDC in Europe and a weighted average over 5 test cycles in the US). However, the trends are similar. In 2015, the average CO₂ emissions of new passenger cars and light trucks have to be below 263 g CO₂/mi (163 g CO₂/km) in the US [3]. By 2020, the limit will be reduced to 213 g CO₂/mi (132 g CO₂/km) and by 2025 vehicles will have to emit less than 163 g CO₂/mi (101 g CO₂/km) [4]. The

trends of Chinese and Japanese regulations are similar [5]. A comprehensive overview of the worldwide pollutant and CO₂ legislation is given in [5].

In order to fulfill these ambitious targets, OEMs need to invest in a mix of different technologies. The electrification of the powertrain and further improvement of the internal combustion engine are important measures, but the use of alternative fuels is also a possible way to reduce CO₂ emissions. An important alternative fuel is natural gas (NG) which consists mainly of methane (CH₄). The hydrogen-carbon-ratio (H-C-ratio) of methane is 4 (on a molar basis), while gasoline and diesel only have H-C-ratios of approx. 1.87 [6]. Therefore, less CO₂ is produced during the combustion of NG than during the combustion of gasoline or diesel. Under the assumption of equal engine efficiencies and complete combustion, methane enables a CO₂ reduction of approx. 25 % compared to gasoline or diesel engines [7]. NG passenger cars are not widely spread yet, although hardly any other single measure facilitates such a large reduction of real-world CO₂ emissions, is comparably cost-effective and short-term realizable.

This work focuses on a dual fuel (DF) combustion process which utilizes compressed natural gas (CNG) and diesel as fuels. In contrast to bivalent engines, which can switch between two fuels, dual fuel engines require *both* fuels simultaneously. One aim of NG-diesel dual

fuel engines is to combine the high efficiency of diesel engines and the CO₂ savings potential of NG. In order to maximize the CO₂ savings, the injected diesel mass needs to be as small as possible as depicted in Figure 1.

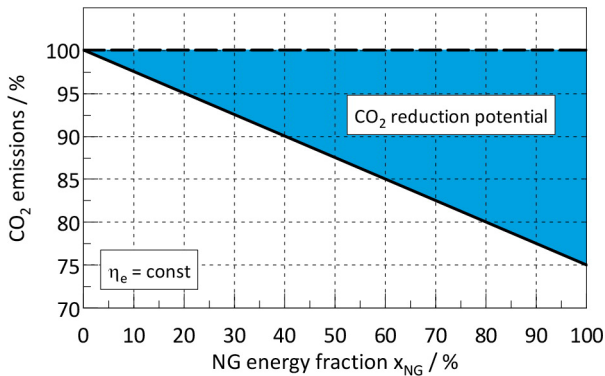


Figure 1. CO₂ emissions in dependency of the NG energy fraction under the assumption of equal engine efficiencies and complete combustion [8].

In several previous studies [9, 10, 11, 12, 13, 14, 15, 16, 17], a layout was analyzed which features port injection of CNG and a direct injection of diesel which ignites the air-fuel mixture. The challenge of this concept is a significant emission of unburned hydrocarbons (HC) during low load operation. In addition, the unburned HC consist mainly of CH₄, which is difficult for the exhaust gas aftertreatment system (EAS) to convert and is a particularly harmful greenhouse gas.

In order to overcome this problem, a sequential direct injection of both CNG and diesel was proposed, which enables charge stratification [8]. Preliminary measurements on the engine test bench indicated that the unburned HC emissions can be significantly reduced with this layout [8]. This work develops this approach further. A state-of-the-art 2 L passenger car diesel engine was extensively modified to integrate a low pressure direct injection of CNG (max. 18 bar) in addition to the standard diesel injection system. In order to maximize CO₂ savings, the injected diesel mass was reduced to a minimum. It is only used to ignite the air-fuel mixture.

In this work, the main application parameters which influence the thermodynamic characteristics and the engine-out emissions of the described combustion process were identified and investigated experimentally. Based on these results, the combustion process was optimized. Finally, the CO₂ emissions, the thermodynamic characteristics and the engine-out pollutant emissions of the optimized DF combustion process were compared with a state-of-the-art turbocharged gasoline engine of the same capacity and with the base diesel engine. The measurements were conducted for two low load and two medium load operating points.

In the first section of this paper, an overview of dual fuel concepts is given and definitions are introduced. Subsequently, the investigated concept is explained in detail. The experimental setup and the methodology are introduced. Afterwards, the influences of the most important parameters are discussed. These are the start of injection of NG (SOI_{NG}), exhaust gas recirculation (EGR), the crank angle of 50 % fuel burned (MFB50), the air-fuel equivalence ratio and the swirl flap position (SFP). Finally, the results of the optimized combustion

process are presented and compared to conventional gasoline and diesel engines. In summary and outlook, the conducted work is summarized and suggestions for further investigations are given.

OVERVIEW OF DUAL FUEL CONCEPTS

Three categories of engines can be distinguished with regard to the number of fuels used: monovalent engines, bi- or trivalent engines and dual fuel engines. Monovalent engines run on only one fuel as conventional gasoline or diesel engines. Bi- and trivalent engines operate with 2 and 3 fuels respectively, but only run on one fuel at a time. For example CNG passenger cars can often run on gasoline too. However, such a layout is a compromise as the engine cannot be optimized for one fuel. In contrast, dual fuel engines run on two fuels simultaneously. The investigation of several different fuel combinations has been documented in literature such as NG-diesel [9, 10, 11, 12, 14, 15, 16, 17], hydrogen-diesel [10, 12, 13], gasoline-NG [14], hydrogen-NG [12, 18] etc. In fact, the dual fuel combustion is not a new concept. For example, for the IFA W50, a truck built from 1965 - 1990, a conversion kit existed for CNG-diesel DF operation. In academia, the ignition mechanisms of DF combustion were experimentally investigated in a constant volume chamber as early as 1968 [19]. Today's focus of DF research is on adapting existing concepts to the current and future emission legislation and reviewing the applicability to truck and passenger car engines. This institute has also conducted several studies on dual fuel combustion [8, 9, 10, 11, 12, 13, 18, 19, 20, 21].

Subsequently, only NG-diesel dual fuel engines are considered. There are different motivations behind the development of dual fuel engines across the engine segments which lead to different concepts. Some combustion processes work with varying fuel fractions, others with a small and fixed injection quantity of diesel which substitutes a spark plug. In some cases, an operation without limitations is also possible in diesel-only mode. Other combustion processes only offer a 'limp home' functionality in diesel-only mode. An operation with NG-only is usually not feasible, as the diesel injection is needed to ignite the air-fuel mixture. The challenge of NG-diesel dual fuel engines is the high emission of unburned HC during low load operation. Complicating matters, the unburned HC consist mainly of CH₄ which requires temperatures of approx. 500° C to be converted by an oxidation catalyst [22, 23, 24, 25].

For large bore engines (ship engines), dual fuel combustion processes are state-of-the-art [26]. There, emission limits are met because large bore engines usually operate close to full load where HC emissions are less critical. The motivations for introducing dual fuel combustion in large bore engines are the low costs of NG, combined with the flexibility to run on heavy fuel oil if necessary. Furthermore, LNG tanker ships can directly combust the tank boil-off which remains unused otherwise.

For truck engines, there are a few aftermarket solutions available [27, 28], but only Volvo developed a LNG-diesel truck engine, which achieved EU5 emission standards, and introduced a small batch series to the market [29].

Dual fuel combustion processes are being investigated for small bore engines and different technologies are being reviewed. The motivation is mainly the need to reduce the CO₂ emissions of passenger cars. However, as passenger cars are operated in partial load to a large extent, the reduction of unburned HC emissions is of crucial importance. A concept reviewed by many authors is the port injection of NG combined with a direct injection of diesel [8, 9, 10, 11, 12, 13, 14, 15, 16, 17]. Different diesel injection strategies were proposed [16] and the influence of the NG fraction, EGR and the air-fuel equivalence ratio on the emission formation was investigated [8]. However, the unburned HC emissions remained unsatisfactorily high. Alternatively, layouts which feature a direct injection of both NG and diesel were reviewed. Thus, a stratification of the charge becomes feasible. Again there are two possible approaches: concepts which utilize a high pressure injection of NG and concepts with low pressure injection. For the high pressure injection an interesting approach exists for heavy duty applications. An injector was developed which is responsible for both the CNG and the diesel injection [30, 31]. Both concepts, low pressure and high pressure injection, seem promising regarding the achievable efficiencies and engine-out emissions [8, 31, 32]. The system pressure influences the feasible stratification and the cruising range of the vehicle as detailed subsequently.

DEFINITIONS

A few definitions which are used in this paper need to be introduced before the investigated concept is explained in detail. According to Equation (1) the air-NG equivalence ratio λ_{NG} only considers the NG-air mixture. It characterizes the stoichiometry of the charge in which the diesel spray is injected. It is an important parameter which influences the completeness of combustion.

$$\lambda_{NG} = \frac{m_{air}}{m_{NG} \cdot L_{stNG}} \quad (1)$$

Equation (2) defines the NG energy fraction x_{NG} which describes the fuel energy ratio of NG and diesel. In order to reach high CO₂ savings, high values of x_{NG} are desirable.

$$x_{NG} = \frac{m_{NG} \cdot LHV_{NG}}{m_{NG} \cdot LHV_{NG} + m_{Diesel} \cdot LHV_{Diesel}} \quad (2)$$

Equation (3) defines the EGR-rate. According to [7] it may be calculated from the inducted masses of fresh air (m_{Air}) and recirculated exhaust gas (m_{Exh}) or from the measured CO₂ concentrations. $[CO_2]_{Int}$ is the CO₂ concentration of the inducted charge (fresh air + recirculated exhaust gas), $[CO_2]_{Exh}$ is the CO₂ concentration in the exhaust gas and $[CO_2]_{Air}$ is the background concentration of CO₂ in the ambient air which is approx. 270 ppm.

$$EGR = \frac{m_{Air}}{m_{Air} + m_{Exhaust}} = \frac{[CO_2]_{Int} - [CO_2]_{Air}}{[CO_2]_{Exh} - [CO_2]_{Air}} \quad (3)$$

Equation (4) - (5) define the unburned species $\Delta\zeta_{IC}$ which represent the losses due to incomplete combustion. $\Delta\zeta_{IC}$ relates the energy which is contained in the unburned species of the engine-out emissions to the fuel energy. The subscript IC stands for 'incomplete combustion'. Eq.

(4) is valid for lean operation because only carbon monoxide (CO) and the total hydrocarbon (THC) emissions are considered. For stoichiometric operation, hydrogen (H₂) also needs to be considered (Eq. (5)). The stated THC emissions include the CH₄ emissions which account for the main part of THC emissions in NG operation.

$$\Delta\zeta_{IC} = \frac{m_{CO} \cdot LHV_{CO} + m_{THC} \cdot LHV_{THC}}{m_{NG} \cdot LHV_{NG} + m_{Diesel} \cdot LHV_{Diesel}} \quad \text{for } \lambda > 1 \quad (4)$$

$$\Delta\zeta_{IC} = \frac{m_{CO} \cdot LHV_{CO} + m_{THC} \cdot LHV_{THC} + m_{H_2} \cdot LHV_{H_2}}{m_{NG} \cdot LHV_{NG} + m_{Diesel} \cdot LHV_{Diesel}} \quad \text{for } \lambda \leq 1 \quad (5)$$

THE ORIGIN OF THE UNBURNED HC EMISSIONS IN NG-DIESEL DUAL FUEL COMBUSTION

In order to understand the effect of charge stratification on the dual fuel combustion process, the reason for the high HC emissions of port injection layouts need to be explained. In [8] the NG-diesel dual fuel combustion process with port injection is explained in detail. Since this publication is written in German, the findings which are relevant in the context of the NG direct injection are summarized below. The interested reader may refer to [8] for further results.

Figure 2 illustrates the model representation of the dual fuel combustion with port injection. It displays a cross-section of the combustion chamber close to firing TDC. After induction, at the end of the compression stroke, there is a homogeneous and compressed NG-air mixture in the combustion chamber. Its stoichiometry equals λ_{NG} . The upper flammable limit of NG-air mixtures is $\lambda_{lean} \approx 2$ [33]. A small amount of diesel is then injected. It influences the local stoichiometry. Inside the liquid core of the diesel spray $\lambda_{local} = 0$ while towards the cylinder walls λ_{local} still equals λ_{NG} . In between these limits there is a gradient of the local air-fuel ratio. After the diesel spray ignites, whether a flame front can develop and propagate through the combustion chamber depends on the stoichiometry of the NG-air mixture. If $\lambda_{NG} > \lambda_{lean}$ the flame extinguishes at $\lambda_{local} \approx \lambda_{lean}$ and the remaining NG is exhausted as unburned HC. A flame front can only develop and propagate towards the cylinder walls if $\lambda_{NG} < \lambda_{lean}$. This correlation is observed in Figure 3. It states measurements of a NG-diesel dual fuel combustion process with *port injection*. The operating point was $n = 1750$ rpm / BMEP = 5 bar and the NG fraction x_{NG} was set to 85 % which represented a diesel injection of 2.7 mg/cycle. The diesel rail pressure was 400 bar. The experimental setup is explained in [8]. Starting at wide-open throttle, the intake manifold pressure was gradually lowered by the throttle valve. At wide-open throttle, the intake manifold pressure p_{int} was 1030 mbar and the air-NG equivalence ratio λ_{NG} was 2.14 due to the low load. It exceeds the upper flammable limit λ_{lean} . Therefore, the unburned species $\Delta\lambda_{IC}$ reached 9.7 %. By throttling, λ_{NG} was subsequently lowered and stoichiometric conditions ($\lambda_{global} = 1.0$ and $\lambda_{NG} = 1.14$) were reached at $p_{int} = 700$ mbar. The unburned species decreased by 75 %. However, the ignition delay rose significantly at the same time because the self-ignition conditions for the diesel spray worsened due to the lower in-cylinder pressure. This affected the combustion negatively: the combustion duration increased and the indicated

efficiency reduced. In order to keep MFB50 constant at 11 °CA, the SOI of diesel (SOI_{Diesel}) had to be advanced gradually from -9 °CA to -18 °CA as displayed in Figure 3.

Apart from throttling, another way to reduce λ_{NG} is EGR. However, the same trends were observed. The air-NG equivalence ratio λ_{NG} and the unburned species reduced with increasing EGR, but the combustion degraded. The combination of EGR and throttling also did not improve the combustion stability of close to stoichiometric air-fuel mixtures. At lower loads (e.g. BMEP = 3 bar) the air-NG mixture was even leaner and λ_{lean} was exceeded even at throttled operation and with EGR.

This relation explains the high HC emissions at low load operation of DF combustion processes which utilize port injection of NG. There is a need to reduce the air-NG equivalence ratio, but at the same time the conditions necessary for a self-ignition of the diesel spray prevent the required reduction of λ_{NG} .

In addition, in Figure 3 it can be observed that more than 90 % of THC emissions on a molar basis consist of CH_4 . This confirms that the unburned species result mainly from NG.

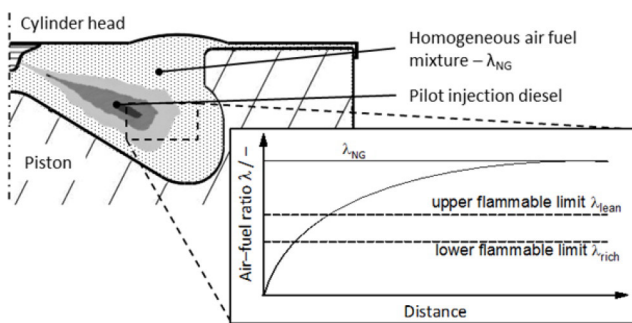


Figure 2. The model representation of a dual fuel combustion process with NG port injection; cross section of the combustion chamber at firing TDC [8].

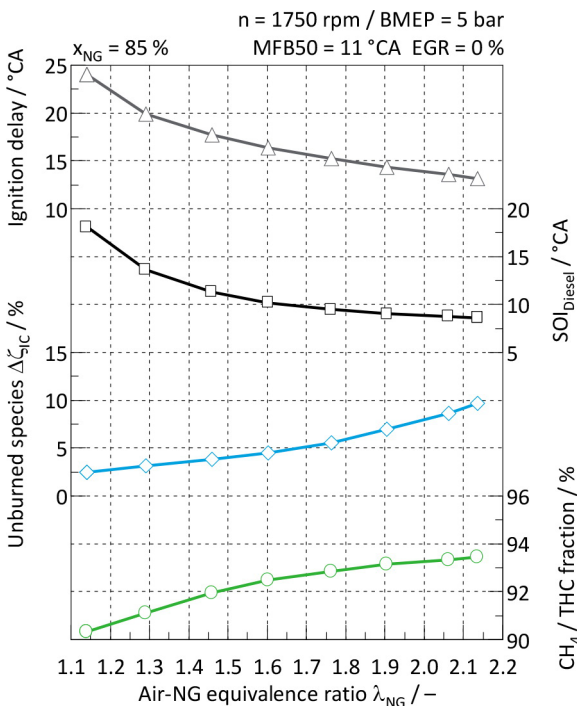


Figure 3. Engine characteristics of a NG-diesel dual fuel combustion process with NG port injection in dependency of the air-NG equivalence ratio.

THE INVESTIGATED CONCEPT

By introducing direct injection, the charge remains globally lean. Thus the conditions necessary for the self-ignition of the diesel spray are maintained. However, due to stratification, the local air-fuel ratio is lowered which is beneficial for a complete combustion as demonstrated in Figure 3. The aim was to center the NG around the diesel spray, with only air close to the cylinder walls and in the crevices. This is expected to reduce the unburned HC emissions significantly. Another advantage of direct injection is the additional charge motion, which is introduced by the injection spray and which benefits combustion. Furthermore, the volumetric efficiency increases which benefits the low-end torque.

A low pressure system was chosen because high system pressures affect the cruising range of CNG vehicles negatively. The maximum pressure of a vehicle's CNG tank is approx. 200 bar. However, the complete tank content cannot be utilized to propel the vehicle. Once the tank pressure equals the injection pressure, refilling is required. If the system pressure was 50 bar, 25 % of the energy content would remain unused (when assuming ideal gas). Therefore, a maximum system pressure of 18 bar was chosen for this work. The disadvantage is that the latest possible injection of NG is limited by the cylinder pressure as demonstrated in Figure 4. The pressure ratio across the injector nozzle was required to be supercritical until the end of injection (EOI), to ensure a stable NG injection. Typical injection durations for NG were approx. 2 ms (20 °CA) at 16 bar rail pressure for the load point $n = 1750 \text{ rpm} / \text{BMEP} = 5 \text{ bar}$. Thus, with the chosen setup at low load operation, the latest possible SOI_{NG} was approx. 70 °CA before TDC, which enabled moderate charge stratification. At higher loads, the SOI_{NG} had to be advanced. In Figure 4 the injection signal of the diesel injection is displayed as well. In this operation point SOI_{Diesel} was at -13.5 °CA. The diesel rail pressure was set to 400 bar and the corresponding injection duration was 0.35 ms.

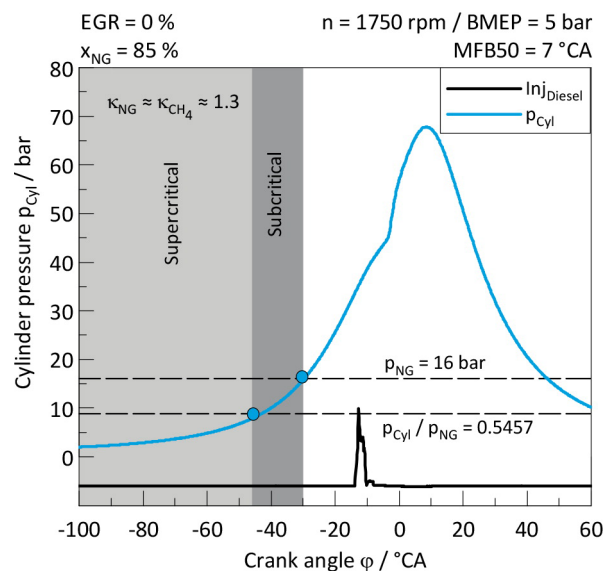


Figure 4. The relation between the injection pressure of NG p_{NG} , the cylinder pressure p_{Cyl} and the implications on the end of injection (EOI).

Different and varying fuel energy ratios are conceivable for dual fuel combustion processes, depending on the operating point. The motivation behind this investigation was the CO₂ savings potential of a NG-diesel dual fuel combustion process. Therefore, attempts were made to reach high NG fractions. Diesel injection was reduced to a minimum and was only utilized to ignite the air-fuel mixture. The minimum diesel quantity was limited to 2.4 mg/cycle by the injection system. Depending on the load point, this resulted in NG energy fractions between 80 % and 95 %. Compared to a small bore, NG spark ignited (SI) gasoline engine, the ignition energy is higher by several orders. In addition, the chosen layout offers the advantage of enabling diesel-only mode as well.

EXPERIMENTAL SETUP

The experimental setup is depicted in Figure 5. The base engine was a state-of-the-art 2 L diesel engine that complies with EU6 emission limits. Table 1 states the engine parameters of the base engine. In order to introduce the CNG, outward-opening injectors with a tapered nozzle tip were used. The injectors were in a late prototype stage and were provided by an external supplier. The maximum rail pressure was 18 bar. Figure 6 demonstrates the chosen integration. The CNG injectors were located centrally in the combustion chamber and the diesel injectors were placed eccentrically and slightly tilted. This layout with central CNG injector instead of a central diesel injector had to be chosen due to mechanical constraints. A specifically casted and machined cylinder head was needed to accommodate the injectors. The inlet ports and the diesel common rail system remained unchanged, but one exhaust valve per cylinder had to be removed in order to add the CNG injectors. The compression ratio and the piston geometry (ω -bowl) of the base engine were maintained. The serial diesel injectors with 8 holes were replaced by injectors with 6 holes, reduced flow rate and an asymmetric spray to account for the altered position.

The CNG injectors were actuated by a separate control unit and a power amplifier. The diesel injection and all other application parameters such as EGR, the position of the variable nozzle turbine (VNT), etc. were controlled by the engine control unit (ECU). Natural gas with a methane content higher than 95 vol% was used for all measurements. The measurement equipment installed at the test bench is listed in the appendix.

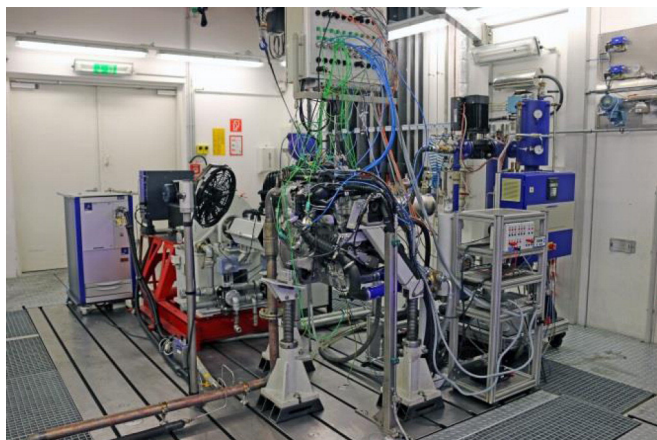


Figure 5. Experimental setup on the engine test bench.

Table 1. Engine parameters of the base diesel engine.

Cylinder	4
Displacement volume	2.0 Liter
Compression ratio	16.5 : 1
Number of valves	4
Rated power	140 kW / 4000 rpm
Rated torque	400 Nm / 1750 rpm
Technology	Common-rail injection system with 2000 bar VNT turbocharger with intercooler High pressure EGR with and without cooler Swirl flap Throttle valve EU6 certified



Figure 6. Layout of the combustion system in TDC with the central CNG injector and the inclined diesel injector.

METHODOLOGY

5 load points were chosen for the conducted measurements. Three of them are characteristic for the NEDC (New European Driving Cycle) as depicted in Figure 7. In addition, one medium load and one full load operating point were chosen:

- n = 1500 rpm / BMEP = 3 bar
- n = 1750 rpm / BMEP = 5 bar
- n = 2000 rpm / BMEP = 11 bar
- n = 1750 rpm / BMEP = 15 bar
- n = 2000 rpm / BMEP = max.

For the sake of brevity, only the results of the load point n = 1750 rpm / BMEP = 5 bar are stated in this publication. This operating point was chosen because, as mentioned before, the challenges of the dual fuel combustion process are mainly revealed in low load operation. During the measurement campaign, the influence of the following parameters was analyzed:

- Start of injection of NG (SOI_{NG})
- EGR rate
- Air-NG equivalence ratio λ_{NG}
- NG energy fraction x_{NG}
- MFB50
- Swirl flap position (SFP)
- EGR temperature
- Diesel rail pressure
- CNG rail pressure



Documents

Document Sharing Hub