

Extending the NO_x Reduction Potential with Miller Valve Timing Using Pilot Fuel Injection on a Heavy-Duty Diesel Engine

Clemens Brückner, Panagiotis Kyrtatos, and Konstantinos Boulouchos
Swiss Federal Institute of Tech.

ABSTRACT

New emission legislations applicable in the near future to sea-going vessels, off-road and off-highway vehicles require drastic nitric oxides emission reduction. A promising approach to achieve part of this decrease is charge air temperature reduction using Miller timing. However, it has been shown in literature that the reduction potential is limited, achieving a minimum in NO_x emissions at a certain end-of-compression temperature. Further temperature reduction has shown to increase NO_x emissions again. Some studies have shown that this increase is correlated to an increased amount of premixed combustion.

In this work, the effects of pilot injection on engine out NO_x emissions for very early intake valve closure (i.e. extreme Miller), high boost pressures and cold end-of-compression in-cylinder conditions are investigated. The experiments are carried out on a 3.96L single cylinder heavy-duty common-rail Diesel engine operating at 1000 rpm and at constant global air-to-fuel ratio. Engine operating conditions with single and pilot injection are compared for a wide range of intake air temperatures and injection timings.

From the experiments, it is shown that at cold TDC conditions, further decreasing intake temperatures for single injection events leads to an increase in exhaust NO_x emissions at constant specific fuel consumption. For these points longer ignition delays and greater premixed proportions are observed. Splitting the injection event shows to counteracts this effect. This approach leads to a decrease of NO_x emissions with decreasing intake temperatures. Thus, the applicability of extreme Miller valve timing is extended, allowing its use for further reduction of NO_x emissions.

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INTRODUCTION

Although the field of research on internal combustion engines dates back to over a century, significant improvements can still be made. The increasing demand for mobility and transportation, as well as increasingly stringent emission regulations worldwide, have driven the development of internal combustion engines mainly towards the reduction of gaseous and particulate emissions, while enhancing the overall efficiency [1]. For the specific application of combustion in diesel engines, the main focus has been on the reduction of exhaust Nitric Oxides (NO_x) and Particulate Matter (PM), while maintaining or even improving fuel economy, and thus Carbon Dioxide (CO₂) emissions, which is known to be a greenhouse gas. New emission requirements for non-road and stationary power generation applications (EPA Tier 4), as well as for sea-going vessels (IMO Tier 3), becoming effective in 2015 and 2016¹ respectively, require further NO_x tailpipe emission reductions of up to 80% [2,3]. This radical reduction requires

additional efforts in research and development in order to achieve the requested NO_x levels without deteriorating the Specific Fuel Consumption (SFC).

To achieve this, two different approaches are possible:

1. Reducing the amount of NO_x formed during combustion by altering/optimizing the combustion process.
2. Reducing NO_x in the exhaust by means of after-treatment systems.

The work presented in this paper focuses on means of further in-cylinder NO_x formation reduction through means of very low end-of-compression temperatures.

A promising technology to achieve low in-cylinder temperatures, while keeping the SFC at a constant level, is closing the intake valve well before bottom dead center (generally known as Miller valve timing) in combination with high boost pressures [4,5,6,7]. Due to the early intake valve closure the charge air is expanded prior to compression,

1. Final adoption date pending by time of publication.

reducing the charge temperature at end of compression, and thus the adiabatic flame temperature. To counterbalance the power output losses resulting from the early valve closure, high boost pressures are required. Thus, this approach partially shifts the compression work from the piston compression stroke to the charging system, allowing to reject this part of the generated compression heat via inter- and after-cooler. The result is a reduction of the charge air temperature before fuel injection and consequently a reduced cycle temperature, which significantly reduces the thermal NO_x formation during and after combustion. It has recently been shown that these systems achieve similar power outputs as conventional state-of-the-art diesel engines with drastic reduction in NO_x emissions [4,7].

The success of this approach can be explained by the strong correlation between the resulting NO_x emissions and the adiabatic flame temperature in diesel combustion, that has been continuously reported in literature [8,9]. This correlation states that lower NO_x emissions should be expected with reduced adiabatic flame temperatures, due to the strong temperature dependency of the NO_x formation kinetics.

Contrary to expectations, recent investigations have reported in several cases that the adiabatic flame temperature correlation fails to capture the trend of NO_x emissions for diesel combustion [6,10,11,12,13,14]. For instance, Kidoguchi et al. [10] reported for a small-bore engine that a reduction of cetane number of a paraffinic fuel blend and retardation of Start of Injection (SOI) caused an almost two-fold increase in NO_x emissions, even though the adiabatic flame temperature remained nearly constant. Pierpont and Reitz [11] observed on a large-bore direct injection (DI) diesel engine that increasing injection pressure required significant SOI retardation to keep the engine-out NO_x constant. Consequently, a reduction in peak adiabatic flame temperature by shifting combustion towards the expansion stroke had barely any effect on NO_x emissions.

Kyrtatos et al. [6,12] have shown on a medium speed marine diesel engine that reducing the Top Dead Center (TDC) temperature by means of advancing the inlet valve closure (i.e. Miller valve timing), while keeping the intake air density constant, leads first to a reduction in NO_x emissions at constant SFC. Eventually, though, further temperature reduction leads to an increase in NO_x emissions. These conditions exhibit longer Ignition Delays (ID), allowing for more fuel being mixed prior to ignition and thus leading to greater magnitudes of premixed combustion. Moreover, the long ID allows for the spray to penetrate further and enhance air entrainment, resulting in an overall leaner premixed combustion that could favor NO_x formation.

From experiments with a single cylinder, DI optical diesel research engine, Musculus [13] has reported two trends contradicting the expected NO_x emission outcome predicted by the adiabatic flame temperature, that strongly correlate with the premixed combustion. It is reported that reducing the intake temperature, NO_x initially decreased and then increased

monotonically with premixed combustion. Equivalently, it is found that retardation of SOI eventually leads to an increase of NO_x after reaching a minimum for late injections, for operating conditions with large premixed ratios.

In addition to the findings described previously, many studies have shown that by decreasing the magnitude of premixed combustion, the NO_x emissions were reduced [10,11,13], suggesting a correlation between diesel premixed combustion and NO_x emissions.

Although the influence of the ratio of premixed combustion is not yet fully understood, several chemical and physical mechanisms can be found in the literature, which explain the apparent correlation between diesel premixed combustion and NO_x emissions. An overview is given in [13], briefly stating the following possible sources for increased NO_x emissions with increasing premixed combustion:

- **Compression heating.** Large portions of premixed combustion generally lead to a faster combustion, and consequently to higher in-cylinder pressures early in the cycle. Thus, it is argued that the reactants entering the diffusion flame are compressed to higher temperatures, leading to higher diffusion flame temperatures. It is stated in [13], however, that this effect is insufficient to explain the NO_x trend reversal. In addition, spray penetration is increased, leading to a larger flame surface and better use of the available oxygen in the combustion chamber. It has been reported that the air-fuel mixing rate during diffusion combustion is increased [12] due to the longer spray penetration prior to ignition, resulting in higher temperatures during diffusion combustion and thus more thermal NO formation.
- **NO_x formation in premixed combustion.** It has been suggested that at conditions with very long IDs the mixture might be lean enough (close to or leaner than stoichiometry) to form significant thermal and/or prompt NO in the premixed combustion products.
- **Reduced soot radiation cooling.** Musculus [14] argues that at operating conditions exhibiting large proportions of premixed combustion the radiative cooling due to soot is drastically lower due to the lower concentrations of particulates in the flame. Hence, the actual flame temperature is increased, leading to higher thermal NO_x formation.

Furthermore, an increase in pressure fluctuation intensity has been reported in literature with increasing ID [12,15], leading to high cycle-to-cycle variations with extreme ringing. It is shown that the high cyclic pressure fluctuations are reflected in the peak-to-peak cycle resolved NO emissions [12].

Based on the findings in literature presented, reducing the ID is expected to counteract the NO_x trend reversal, by reducing the magnitude of premixed combustion at very cold in-cylinder conditions.

One approach to reduce the ID is changing the injection strategy from a single main injection to split injection, through the application of a short pilot injection prior to the main injection. The pilot injection serves the specific purpose of providing ignition kernels, and thus to shorten the ID of the main injection. Consequently, the magnitude of premixed combustion is expected to be reduced. To this point, the use of a pilot injection, used to reduce the ID of the main injection, is seen as having potential to reduce this effect [16] and should thus allow further in-cylinder NO_x reduction when using extreme Miller valve timing, counteracting the NO_x trend reversal found in [6,12].

This study investigates the effects pilot fuel injection at very cold in-cylinder conditions on NO_x emissions and SFC at different Operating Conditions (OC), which are expected to extend the NO_x reduction potential with extreme Miller timing.

The paper begins by giving an overview of the experimental facility and the equipment used to acquire the measured data. This is followed by a description of the different operating conditions studied and by experimental observations of the NO_x -SFC trade-off trends and corresponding ratios of premixed combustion for increasing Miller degree. Finally, the influence of pilot injections at extreme Miller degrees and different loads is presented.

EXPERIMENT

Experimental Setup

A four-stroke, single cylinder, DI research diesel engine based on a MTU 396 series engine was used for the investigation. A schematic view of the research engine facility is shown in Figure 1. Due to the strong pressure fluctuations in the intake and exhaust path, which arise from the gas exchange dynamics of the single cylinder engine, large damping volumes have been installed upstream and downstream of the engine.

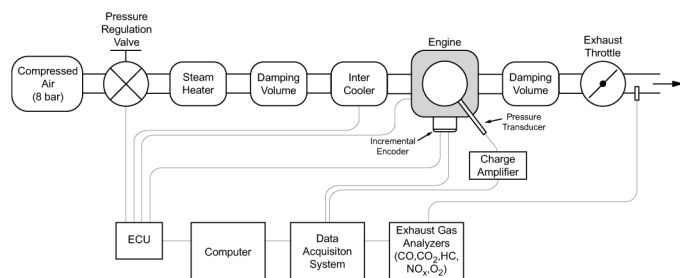


Figure 1. Schematic diagram of the experimental setup.

The engine is retrofitted with a Ganser common-rail direct injection system. The fuel injection system is composed of a solenoid injector with an 8-hole nozzle tip connected to a fully adjustable electronic power stage (dSPACE RapidPro Power Unit). The intake is supplied with pressurized air (up to 5 bar), which can be conditioned through heating and cooling from approximately 17°C to 100°C. The exhaust back pressure can be independently controlled using two fast, separately,

actuated exhaust gas throttles. The engine is connected to an eddy current dynamometer (Zöllner-Kiel AG, Type B-300 AC). A comprehensive overview of the engine and injector specifications is given in Table 1.

Table 1. Engine and Injector Specifications.

Engine type	MTU 396
Number of cylinders	1
Cycle	4-stroke
Displacement	3.96 liters
Stroke	185 mm
Bore	165 mm
Connecting rod length	341 mm
Compression ratio	13.77:1
Eff. compression ratio Miller Timing	11.65:1
Swirl number	0.57 ^a
Number of intake valves	2
Number of exhaust valves	1 ^b
Maximum cylinder pressure	155 bar
Engine speed range	800 – 1600 rpm
Intake pressure	≤ 5 bar
Intake temperature	17 – 100 °C
Exhaust back pressure	≤ 4 bar
Injection system	Direct Injection, Common Rail
Injector type	Solenoid, pilot-valve actuated
Number of holes and arrangement	8, uniformly spaced
Included angle	160 °
Maximum injection pressure	1600 bar
Nozzle orifice diameter	240 µm
Nozzle orifice L/D	6.25

^a Swirl number determination according to Tippelmann (SAE Technical Paper 770404, 1977) [17]

^b in this research diesel engine, one exhaust valve has been replaced with a water-cooled sensor adaptor for optical access or placement of a second pressure sensor.

It should be noted that the only auxiliary device powered by the engine is the high pressure pump for the injector. Cooling and lubrication are powered externally. The same applies for the boost pressure, which is obtained from a high pressure supply. Hence, a correction calculation for SFC is applied, which will be described later. In addition, one exhaust valve is replaced with a water-cooled sensor adaptor, which allows the placement of additional pressure or optical sensor.

The fuel consumption is measured using a dynamic fuel balance (AVL 733S).

Controlling of the test bench is performed using a dSPACE MicroAutoBox II programmable electronic control unit (ECU). The ECU controls fuel injection parameters such as injection pressure, injection duration and injection timing, as well as intake conditions (pressure and temperature) and exhaust back

pressure. Injection event timing is realized in reference to the incremental encoder (Hübner-Berlin POG 10 DN 900 TTL) transistor-transistor logic signal.

In-cylinder pressure is measured using a piezoelectric pressure transducer (Kistler 6041A). The pressure sensor is located between the intake and exhaust valve within the bowl region, allowing pressure measurements without influence of squish flow. The intake and exhaust pressures were measured using piezoresistive pressure transducers (Kistler 4075A10). Both amplifiers were mounted to a signal conditioning platform (Kistler SCP 2853A).

Exhaust gas emissions are measured with two continuous exhaust gas analyzers (ABB AO2000 Series; ECO PHYSICS CLD 82 M hrk1). The ABB analyzer includes a Non-Dispersive Infrared Absorption (NDIR) analyzer (URAS 26) to measure Carbon Monoxide (CO) and CO₂, a Flame Ionization Detector (MULTI FID14 NMHC) to measure unburned Hydrocarbons (HC), and a paramagnetic Oxygen (O₂) analyzer (MAGNOS 206). NO_x is measured with a Chemiluminescence Detector (CLD). Sampling is performed in the exhaust pipe at ambient pressure and the gases are transported to the analyzers in a heated line (190°C) to avoid condensation.

Operating Conditions

For this study the engine is equipped with a very early intake valve close timing, i.e. extreme Miller timing. An overview of the Baseline and Miller valve lift curves, as well as the exhaust valve lift curve, is given in Figure 2. It is apparent that in case of Miller timing the valve not only closes well before Bottom Dead Center (BDC), but also the maximum valve lift is drastically reduced compared to the Baseline case. The exhaust valve timing remained unchanged throughout this study.

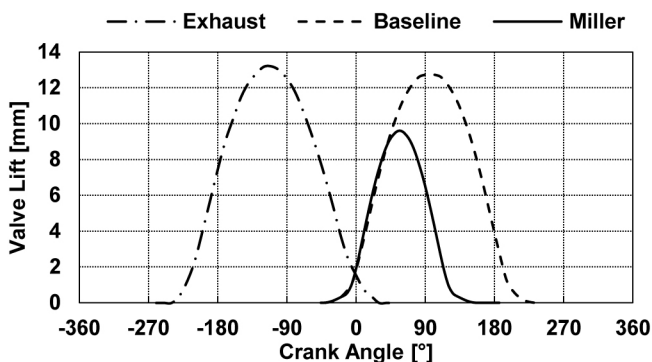


Figure 2. Engine valve timing.

Engine speed is fixed at 1000 rpm and boost pressure is adjusted depending on the intake temperature to keep the air density at intake constant. No Exhaust Gas Recirculation (EGR) is used for any of the measurements. The coolant temperature is set to 70°C. An overview of the operating conditions is given in Table 2.

Table 2. Overview of operating conditions.

Engine speed	1000 rpm		
Intake air temperature	20, 40, 80 °C		
Injection pressure	1000 bar		
Start of main injection	-10.4 to 3.6 °CA aTDC		
Setup 1 – Baseline vs. Miller			
Valve timing	Baseline		Miller
BMEP [bar]	8		8
Injection quantity [mg/stroke]	181		181
Global stoich. Air-to-Fuel ratio	3		3
Injection strategy	Single		Single
Setup 2 – Single vs. Pilot Injection			
Valve timing	Miller		
BMEP [bar]	8	13	16
Total inj. quantity [mg/stroke]	181	271	327.5
Global stoich. Air-to-Fuel ratio	2	2	2
Injection strategy	Single/Pilot	Single/Pilot	Single/Pilot
Injection dwell time [ms]	1	1	1
Rel. pilot fuel quantity [%]	21	14	11.6

Two experimental setups are used for this investigation, which are explained in the following.

The first setup consists of a comparison of Baseline and Miller valve timing at constant global stoichiometric air-to-fuel ratio of 3 and load, for single main injection (see Table 2, Setup 1). The load is chosen to correspond to a low-to-medium load operating condition for this type of engine. Due to the early valve closing of the Miller case, boost pressure is increased to achieve the same global air-to-fuel ratio at constant injection duration as in the Baseline case.

To study the effect of end-of-compression temperature on ignition delay, NO_x emissions, and Specific Fuel Consumption (SFC), the intake temperature is varied from 20°C to 80°C, and a sweep of Start of Main Injection (SOIm) was performed, while keeping injection pressure and duration constant.

In the second setup, the investigation concentrates on Miller valve timing at a global air-to-fuel ratio of 2 and three different loads (see Table 2, Setup 2). The fuel ratio chosen represents real-life operating conditions for engines of this type.

Modern, state-of-the-art 4-stroke, medium speed diesel engines achieve maximum Brake Mean Effective Pressures (BMEP) of up to 25 to 30 bar [4,18]. In order to investigate the effects of pilot spray under low to medium-to-high load operating conditions relevant for this type of engine, the operating points are set to BMEP of approximately 8 bar, 13 bar and 16 bar, respectively.

With this setup, the influence of pilot injection on ID, NO_x emissions and SFC is studied for intake temperatures from 20°C to 80°C and a sweep in SOIm. Injection pressure and, for