Measurement of Cycle-to-Cycle Variations of Emissions on a Single-Cylinder Diesel Engine with Miller Timing

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Measurement of Cycle-to-Cycle Variations of Emissions on a Single-Cylinder Diesel Engine with Miller Timing

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Master of Science in Mechanical Engineering

presented by

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Zürich, Switzerland 2015
For my family, the guardians of my galaxy

MASTER THESIS

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Cover: MTU test bench at the ETH Zurich LAV laboratory.
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Zürich, October 2015

Antonio Živolić
Contents

List of Figures ix
List of Tables xiii

1 Introduction 3
  1.1 Literature Overview ........................................... 4
  1.2 Miller Inlet Valve Timing ..................................... 6
  1.3 Ignition Delay and Premixed combustion ....................... 6
  1.4 Characteristic Mixing Rate .................................... 7
  1.5 Fluctuation Intensity & Pressure Rise Rates .................. 9
  1.6 Pressure Variations Reduction ................................ 11

2 Experimental Procedure 13
  2.1 MTU-396 Single Cylinder Test Engine .......................... 13
  2.2 Fast Sampling Valve ........................................... 15
    2.2.1 Cambustion DMS500 ........................................ 17
    2.2.2 CAMBUSTION CLD500 Fast NO ................................ 18
    2.2.3 AVL Photoacoustic Soot Sensor ............................ 19
  2.3 Measurement Set-Up ........................................... 19
  2.4 Measurement Protocol .......................................... 20
  2.5 Experimental Data Processing ................................. 21
  2.6 NO Signal Interpretation ...................................... 22
    2.6.1 Sensitivity Analysis & Validation ........................ 23

3 Results 25
  3.1 Fast Sampling Valve Characterisation .......................... 25
  3.2 NO Results ................................................... 30
    3.2.1 Effect of Pilot Injection on Cyclic NO Emissions .......... 31
    3.2.2 Nitrogen Oxyde Emission Correlations ....................... 32
    3.2.3 Fluctuation Intensity influence on NO ...................... 34
    3.2.4 Intake Temperature Influence on NO emissions ............. 36
    3.2.5 EGR Effect on NO through Ignition Delay .................. 37
    3.2.6 Effect of Load on NO Emissions through Fluctuations Intensity 39
    3.2.7 Effect of Start of Injection on the ISFC - NO Tradeoff ...... 43
    3.2.8 Effect of Injection Pressure variation on the ISFC Vs. NO Tradeoff ........................................ 46
<table>
<thead>
<tr>
<th>Section</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.3  Soot Measurement Results</td>
<td>47</td>
</tr>
<tr>
<td>3.3.1 DMS500 - Direct &amp; FSV Configuration</td>
<td>48</td>
</tr>
<tr>
<td>3.3.2 FSV Openings Analysis</td>
<td>49</td>
</tr>
<tr>
<td>4    Conclusion</td>
<td></td>
</tr>
<tr>
<td>4.1  Experimental Observations</td>
<td>56</td>
</tr>
<tr>
<td>4.2  Interpretation of Results</td>
<td>57</td>
</tr>
<tr>
<td>4.3  Outlook</td>
<td>57</td>
</tr>
<tr>
<td>bibliography</td>
<td>59</td>
</tr>
</tbody>
</table>
## List of Figures

1.1 International Maritime Organisation limitations on NO emissions in a worldwide. With Tier I and Tier II already in effect and Tier III expected to be in effect in January 2016 [1]. ................. 3  
1.2 Marked in green are the member countries of the IMO organisation, where the IMO regulations are enforced [1]. ................. 4  
1.3 Miller intake valve timing schematic, typical for many modern marine diesel engines. Courtesy of motorship.com .................. 6  
1.4 Miller valve timing effect on NOx Emissions as expected from theory (red line) and as observed by [13] (blue line). ................. 6  
1.5 Typical heat release rates for 3 different inlet temperatures. Engine conditions: 1050rpm, DOI = 2.3ms, SOI = 10°BTDCA, bmep = 12bar ........................................... 7  
1.6 Typical heat release integral for 3 different inlet temperatures. Engine conditions: 1050rpm, DOI = 2.3ms, SOI = 10°BTDCA, bmep = 12bar ........................................... 7  
1.7 Typical mixing rates for three different inlet temperatures. Engine conditions: 1050rpm, DOI = 2.3ms, SOI = 10°BTDCA, bmep = 12bar ........................................... 9  
1.8 Variations in the pressure trace for a high fluctuation intensity operating condition. $T_{in} = 20^\circ C$, 581 cycles ........................................... 10  
1.9 Variations in mixing rate for a high fluctuation intensity operating condition. $T_{in} = 20^\circ C$, 581 cycles ........................................... 10  
1.10 Different vibration mode shapes with accompanying mode factors $(\rho_{m,n})$ for a cylindrical chamber ........................................... 10  
1.11 Maximum rate of pressure rise on knock intensity [30] ................. 11  
1.12 Fast Fourier Transform of the average cycle pegged .................. 11  
2.1 Front view of the MTU engine - visible in this photo is the FastNO sampling head, which is installle don a tripod and connected directly to the exhaust of the MTU for cycle resolved NO measurement ................. 14  
2.2 Fast Sampling Valve installed on the MTU exhaust and connected to the DMS500 ........................................... 15  
2.3 Battery for the FSV magnet actuation ........................................... 16  
2.4 Current amplifier for the FSV magnet actuation ........................................... 16  
2.5 Exploded view of the FSV ........................................... 16  
2.6 Micro-Epsilon Proximity Sensor for the FSV ........................................... 17  
2.7 Miro-Epsilon charge signal amplifier for the proximity sensor ................. 17  
2.8 Pressure vessel setup with diesel supply and Airsense connected ................. 18  
2.9 Pressure vessel characterisation - NO measurement equipment ................. 18
<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.10</td>
<td>DMS500 operating principle - Electrometers detect soot particles which are deposited upon them. Courtesy of CAMBUSTION Ltd.</td>
</tr>
<tr>
<td>2.11</td>
<td>CAMBUSTION Fast NO Analyzer</td>
</tr>
<tr>
<td>2.12</td>
<td>AVL Photo Acoustic Soot Sensor (PASS)</td>
</tr>
<tr>
<td>2.13</td>
<td>Measurement setup on the MTU engine exhaust pipe, with the Fast NO and the DMS500 sampling head on the left and right, respectively, and the FSV in the middle. This measurement setup allows for simultaneous cycle-resolved NO and soot measurement, while having remote access to FSV opening, purging and proximity sensor signals from the control room in order to verify the operation of the employed devices.</td>
</tr>
<tr>
<td>2.14</td>
<td>Possible ranges of the NO signal to extract the representative value of NO for an individual cycle. The reference used is the TDC at 360 °CA</td>
</tr>
<tr>
<td>2.15</td>
<td>Sensitivity of the NO emissions vs cylinder pressure on the different °CA ranges used for calculating the representative NO value</td>
</tr>
<tr>
<td>2.16</td>
<td>Sensitivity of the NO emissions vs cylinder pressure on the different °CA ranges used for calculating the representative NO value</td>
</tr>
<tr>
<td>2.17</td>
<td>Deviation of of FastNO Vs GOLEM measured soot averages</td>
</tr>
<tr>
<td>3.1</td>
<td>Pressure vessel setup with diesel supply and Airsense connected</td>
</tr>
<tr>
<td>3.2</td>
<td>Pressure vessel characterisation - NO measurement equipment</td>
</tr>
<tr>
<td>3.3</td>
<td>FSV behaviour under a back-pressure of 1.5 bar with the pressure vessel filled with 500ppm NO gas</td>
</tr>
<tr>
<td>3.4</td>
<td>FSV behaviour under a back-pressure of 2 bar with the pressure vessel filled with 500ppm NO gas</td>
</tr>
<tr>
<td>3.5</td>
<td>FSV behaviour under a back-pressure of 2.5 bar with the pressure vessel filled with 500ppm NO gas</td>
</tr>
<tr>
<td>3.6</td>
<td>FSV behaviour under a back-pressure of 3 bar with the pressure vessel filled with 500ppm NO gas</td>
</tr>
<tr>
<td>3.7</td>
<td>NO signal sample from a measurement point with pilot injection. Measurement conditions: 1050rpm, $m_{fuel} = 9$kg/h, bmep = 12, DOI = 1.97ms, $p_{inj} = 1000$bar, SOI = 10deg BTDC, $T_{in} = 20$ C, $p_{in} = 2.9$ bar, $p_{in} = 2.2$ bar</td>
</tr>
<tr>
<td>3.8</td>
<td>NO signal sample from a measurement point without pilot injection. Measurement conditions: 1050rpm, $m_{fuel} = 9$kg/h, bmep = 12, DOI = 1.97ms, $p_{inj} = 1000$bar, SOI = 10deg BTDC, $T_{in} = 20$ C, $p_{in} = 2.9$ bar, $p_{in} = 2.2$ bar</td>
</tr>
<tr>
<td>3.9</td>
<td>Correlation between NO and smoothed peak pressure - 581 cycles. The expected trend is perfectly linear, spread only due to measurement error</td>
</tr>
<tr>
<td>3.10</td>
<td>Correlation between NO and fluctuation intensity. The red trend line shows the expected correlation when using two pressure sensors instead of one</td>
</tr>
<tr>
<td>3.11</td>
<td>Correlation between NO and characteristic mixing rate during the diffusion combustion - 581 cycles</td>
</tr>
<tr>
<td>3.12</td>
<td>Distribution of fluctuations intensities for 581 cyles with Beta distribution fit</td>
</tr>
</tbody>
</table>
List of Figures

3.13 Individual cycles of a measurement point at $T_{in} = 20^\circ C$, color coded according to their fluctuation intensity deviation from the average. Green - fluctuations below 1 standard deviation from the mean of all the cycles. Red - fluctuations above 1 standard deviation from the mean of all the cycles. Blue - fluctuations within 1 standard deviation from the mean of all the cycles. Dashed lines are mean values of respective groups. .................................................. 35

3.14 Individual cycles of a measurement point, color coded according to their fluctuation intensity deviation from the average. Green - fluctuations below 1 standard deviation from the average of all the cycles. Red - fluctuations above 1 standard deviation from the average of all the cycles. Blue - fluctuations within 1 standard deviation from the average of all the cycles. Light colors are $T = 80^\circ C$, dark colors are $T = 20^\circ C$ .......................................................... 36

3.15 Effect of the intake temperature on the absolute NO emissions of the MTU engine .................................................. 37

3.16 Effect of the intake temperature on the STD of NO emissions of the MTU engine .................................................. 37

3.17 Different oxygen concentrations correspond to different EGR levels which influence the combustion characteristics. Higher EGR brings a more uniform combustion with lower deviation of cycle-to-cycle pressure traces .................................................. 38

3.18 Effect of EGR amount on the average fluctuation intensity ........ 38

3.19 Effect of EGR amount on the maximum fluctuation intensity .... 38

3.20 Effect of EGR amount on STD of NO emissions ................. 39

3.21 Effect of EGR amount on the amount of NO emissions .......... 39

3.22 Effect of load variation on fluctuation intensity, with the low load conditions showing unstable combustion and false readings .... 40

3.23 Effect of varying load on the STD of peak pressure, with low loads showing combustion instability .......................... 40

3.24 CO emissions for varying load conditions, with high CO values for unstable combustion .............................................. 40

3.25 Effect of load on absolute NO emissions .......................... 41

3.26 Effect of load on the STD of the NO emissions .................... 41

3.27 Influence of the load on the absolute NO Emissions of the engine, with separately considered groups of cycles in a certain range of fluctuation intensity .................................................. 43

3.28 Influence of the load on the specific NO Emissions of the engine, with separately considered groups of cycles in a certain range of fluctuation intensity .................................................. 43

3.29 HRR integrals for different load conditions, with the high load conditions showing markedly less ignition delay and a less steeper heat release slope .................................................. 43

3.30 HRR rates for different load conditions, with the diffusion combustion proportion visibly increasing with higher load ................. 43
3.31 Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages .................................. 44
3.32 Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages .................................. 44
3.33 Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages .................................. 44
3.34 Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages .................................. 44
3.35 Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages .................................. 45
3.36 Individual cycle NO vs ISFC tradeoff for a SOI sweep, the trend extends along the usual curve ........................................... 45
3.37 Influence off the high and low fluctuation intensity cycles on the fuel consumption vs NO tradeoff ..................................... 46
3.38 Heat Release rate for early and late SOI. The difference in diffusion combustion proportion is visible ..................................... 46
3.39 Tradeoff for different injection pressures at hot intake conditions. Points are individually measured cycles ................................. 47
3.40 Tradeoff for different injection pressures at cold intake conditions. Points are individually measured cycles ................................. 47
3.41 SSD with FSV vs Direct Measurement 1-4 .................................. 48
3.42 SSD with FSV vs Direct Measurement 5-8 .................................. 48
3.43 SSD with FSV vs Direct Measurement 9-12 .................................. 48
3.44 DMS Direct configuration deviation 1-4 .................................. 49
3.45 DMS Direct configuration deviation 5-8 .................................. 49
3.46 DMS Direct configuration deviation 9-12 .................................. 49
3.47 Dilution ratio for all 12 measurement points in the campaign ........... 49
3.48 Concentration weighted by number of particles for single openings of the FSV during a measurement series ................................. 50
3.49 Concentration weighted by mass of particles for single openings of the FSV during a measurement series ................................. 50
3.50 SSD signals of multiple consecutive FSV openings during a measurement series ................................................................. 51
3.51 Mass signals of multiple consecutive FSV openings during a measurement series ................................................................. 51
3.52 Interpolation method used for obtaining the relevant SDD for a measurement point. The method uses an artificial grid refinement of the 3 most prominent measurements for a given cycle and then creates a smoothed average of the 3 signals ............................................. 52
3.53 Correlations between the geometrical soot particle sizes and relevant combustion parameters. The red line divides the relevant measurement points (size above 20 nm) from the small particles, which are assumed to be created by the lubrication oil [37] ............................................. 53
3.54 Correlations between the soot particle number (mass) and relevant combustion parameters .......................................................... 54
## List of Tables

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>MTU-396 single-cylinder test engine technical specifications</td>
<td>14</td>
</tr>
<tr>
<td>3.1</td>
<td>Dilution of the FSV for different filling pressures of the pressure vessel</td>
<td>29</td>
</tr>
<tr>
<td></td>
<td>and FSV openign durations</td>
<td></td>
</tr>
<tr>
<td>3.2</td>
<td>Engine Measurement Conditions for the NO measurements on the MTU test engine.</td>
<td>30</td>
</tr>
<tr>
<td></td>
<td>The respective values which have been varied for different measurement</td>
<td></td>
</tr>
<tr>
<td></td>
<td>points are shown as ranges</td>
<td></td>
</tr>
<tr>
<td>3.3</td>
<td>Measurement Conditions for the direct soot measurement at the MTU engine,</td>
<td>47</td>
</tr>
<tr>
<td></td>
<td>using the CAMBUSTION DMS500, both directly connected</td>
<td></td>
</tr>
<tr>
<td></td>
<td>and with the FSV</td>
<td></td>
</tr>
</tbody>
</table>
Abstract

The aim of this thesis is to investigate cycle to cycle variations in the combustion process of diesel engines and quantify their effect on engine emissions. Additionally, the effect of cycle to cycle emission variations on the total amount of emissions produced has been shown. It is understood from theory that cycle to cycle pressure fluctuations in the combustion chamber produce "ringing" cycles with an oscillatory pressure trace, affecting the combustion process and the associated emissions on a cycle to cycle basis.

In order to create enhanced cycle to cycle variations and experimentally measure the thermodynamic engine parameters and the emissions produced, a specifically designed experimental engine has been employed. This engine is able to achieve extreme Miller conditions, where the intake valve closes before bottom dead centre (BDC), and therefore results in an decrease in end of compression temperatures. The end of compression temperature is further decreased by decreasing the cylinder intake temperature through use of a charge air cooler. This decreased temperature increases further the fluctuation intensities in the cylinder and allows for measurement of emissions under strongly fluctuating conditions.

The strongly fluctuating conditions showed a higher nitrogen oxide emissions (NO) in the context of this work, adversely affecting the total average tailpipe emissions. This is understood to happen due to the higher pressures and temperatures of highly fluctuating cycles, which created much more NO emissions than regular non-fluctuating cycles. Furthermore, since the NO creation reactions are exponentially dependent on the in-cylinder temperature, the contribution of high fluctuating cycles to the total measure NO amount is significant. Additionally, an effort has been made to simultaneously quantify soot emissions and relate them to cycle to cycle variations as well. However, even though the experimental capabilities have been developed, inconclusive measurement data has been obtained. Therefore, the influence of fluctuation intensity on soot emission is still unclear, with only some indications on possible effects, as expected from theory. Finally, the results of this thesis suggest that without considering the emission contribution of highly fluctuating cycles, the average emission values would be wrongly estimated.

**Keywords:** diesel emissions, cycle to cycle variations, nitrogen oxide, soot, ignition delay, fluctuation intensity, cycle resolved measurement
Pollutant emissions of internal combustion engines have long been a source of concern for researchers and industry, who have been trying to minimise these side effects of the combustion process, while improving engine efficiency. The impending challenges for diesel engine development comes from the highly restrictive International Maritime Organisation (IMO) Tier III regulations scheduled to take effect after the 1st of January 2016, with the limits seen in Figure 1.1 and the IMO member countries presented in Figure 1.2. Consequently, significant effort has been put recently into reduction of especially nitrous oxide \((NO_x)\), while maintaining a competitive fuel consumption and respective \(CO_2\) emissions of diesel engines.

![Figure 1.1: International Maritime Organisation limitations on NO emissions in a worldwide. With Tier I and Tier II already in effect and Tier III expected to be in effect in January 2016 [1].](image)

Throughout the years, two main methods of emission reduction have emerged:

1. In-cylinder - reduce the amount of pollutants formed during combustion
2. After treatment - reduce the amount of emissions in the exhaust stream

**Motivation** While both approaches have been successfully applied, this text focuses on the research of in-cylinder effects on emission formation. Specifically, the effects
of combustion with a low End of Compression (EOC) in-cylinder temperature have been studied and the respective emissions analysed. The EOC temperature is mainly influenced by the intake valve timing, charge reactivity and the intake air temperature, for a given compression ratio. Thus, in the following sections this parameters will be introduced and the effect of their variations explained.

In Chapter 1 a brief description of these parameters is given and their effect explained along with an overview of the state-of-the-art in this research area. Especially the effects of increasingly premixed combustion in diesel engines are analysed, which result from an increased ignition delay (ID). In Chapter 2 the methodologies and the experimental setup are presented, with the used equipment and data analysis tools and techniques. Important for the goal of this thesis was to develop appropriate measurement and data analysis methods to be able to post-process individual measurements. The results of the NO and soot measurement campaigns are presented in Chapter 3, where the obtained data has been analysed and presented, along with corresponding argumentation. Finally, Chapter 4 provides a conclusion of the current work, with a summary of the main insights and recommendations for further work.

Figure 1.2: Marked in green are the member countries of the IMO organisation, where the IMO regulations are enforced [1].

1.1 Literature Overview

Pollutant emission variability is directly related to cyclic combustion variability, and therefore it is important to accurately experimentally identify this variability by the variation of cylinder pressure evolution from one cycle to another [2] as a result of the both stochastic and deterministic combustion related effects [3],[4]. Although, such cyclic variations in combustion have been intensively studied in SI engines, there is a lack of similar studies done for diesel engines. Therefore some of the works in the literature are related to SI engines, but the fundamental phenomena still apply to diesel engines. One of the most relevant results comes from a number of authors, which argue that the cycle to cycle combustion variability has a negative effect on engine performance, fuel consumption as well as engine emissions [5],[6],[7],[8].

Having considered that pollutant formation is directly related to the combustion pro-
cess, it is clear that the mixing rate, burning rate, the peak combustion temperature and residence time in a critical temperature window are some of the main causes for engine emissions variability. As these parameters are linked with the combustion process, cyclic combustion variability not only limits the engine performance but also results in fluctuations of engine emissions [9], [10]. It is also well-known that engine performance and emissions correlate with a non-linear relationship; a small change in indicated mean effective pressure (IMEP) may lead to a remarkable variation of NO emissions [11]. Therefore, cycle-to-cycle variations lead to higher mean NO emission levels than what one would predict without taking NO variability into account [12]. Furthermore, [13] observed that the ignition delay leads to a series of phenomena which affect the NO pollutant creation parameters and therefore correlates with the emission amount.

Owing to the rapid fluctuations of emissions, fast response analyzers are required for the measurement of the cycle resolved emission levels. Such equipment, which is characterised by response times at the order of few milliseconds is suitable for transient emissions studies [14], [15] as well as for the investigation of cycle to cycle emissions amplitude [16]. The chemiluminescence detector (CLD) is the standard method for the measurement of NO levels [17]. The photons, which are emitted by the reaction between the sampled NO and ozone, are detected by a photo multiplier tube. The output CLD voltage is proportional to the NO concentration.

In order to measure the cyclicly resolved soot emissions, a Fast Sampling Valve (FSV) has been used, which has been developed at the Aerothermochemistry and Combustion Laboratory (LAV) at ETH Zurich by Pascal Wilhelm in 2010 [18]. The design is based on works and suggestions by [19], [20] and [21], who published works on combustion chamber sampling and gas sampling methods, as well as on ideas based on [22], [23] and [24] who published studies on particulate formation in diesel engines and direct in-cylinder sampling. Furthermore [25] and [26] used fast sampling valves to study soot formation and oxidation.

In this study, the combustion and emissions variability of a medium speed diesel engine is experimentally investigated. An integrated piezoelectric transducer is employed for cylinder pressure measurement, while fast response analyzers are located at the exhaust manifold for the definition of the cycle-resolved NO emissions. The test protocol includes various operating conditions with different engine speed and load, start of injection timings and inlet temperature conditions. An in-house code for heat release analysis of the measured pressure traces has been used and an add-on code has been developed to analyse the cycle to cycle emissions of the engine. The final results provide an insight into the reasons of combustion and emissions variability, building upon the existing work presented above.

**Objective** The aim of this project was to perform cycle-resolved emission measurements on an experimental diesel engine, in order to quantify cycle-to-cycle emission variations and understand how are they connected to the cycle-to-cycle variations in the combustion process.
1.2 Miller Inlet Valve Timing

The Miller Inlet Valve timing is an existing concept which has been successfully applied to production diesel engines, where the inlet valve closes much earlier than the bottom dead centre (BDC) and thus allows for additional expansion of the in-cylinder gases before compression, as described in Figure 1.3. That results in a decrease in EOC temperature, along with a slightly increased efficiency. However, in order to keep the same power density, a higher intake pressure is needed to fill the cylinder with the same air mass as without using Millertiming. This decreased EOC temperature relates also to a decrease in combustion temperatures and has thus been used as an effective measure to reduce NO emissions, since NO emissions are strongly dependent on the combustion temperature. However, it has been observed by [13] and others that by advancing the Inlet Valve Closing (IVC), the NO emissions increase again after a certain crank angle, as explained in Figure 1.4. This text aims to provide further insight in explaining this phenomenon by analysing how the cycle to cycle variations influence the total NO emissions under such extreme Miller conditions.

![Figure 1.3](image1.png) ![Figure 1.4](image2.png)

Figure 1.3: Miller intake valve timing schematic, typical for many modern marine diesel engines. Courtesy of motorship.com

Figure 1.4: Miller valve timing effect on NOx Emissions as expected from theory (red line) and as observed by [13] (blue line).

1.3 Ignition Delay and Premixed combustion

Under long ignition delay conditions, diesel engines have been observed to operate with highly variable heat release rates between consecutive combustion cycles, resulting in a highly variable cycle-to-cycle pressure trace. This effect is observed mainly due to the strong impact of high combustion rates in the premixed part of the diesel combustion process, which means a higher fraction of the total available fuel for combustion is burned in the premixed combustion phase.

Besides the extreme Miller timing, the other important starting point for the chain
of effects that affect cycle to cycle emission is the intake air temperature. The intake
temperature has a direct effect on the EOC temperature and thus on the reactivity
of the mixture and the ignition delay. These effects can be best observed in Figure
1.5 where the heat release rates for three different intake temperatures have been
plotted. Two main effects can be seen from this figure, firstly that the combustion
is significantly delayed under low intake temperature conditions, and secondly the
low temperature conditions increase the proportion of premixed combustion amount
while decreasing the diffusion combustion amount. Furthermore, Figure 1.6 shows the
heat release integrals for the same conditions, where again the ignition delay of the
low intake temperature conditions is apparent but also, the slope of the lower intake
temperature conditions suggest a stronger and faster combustion upon ignition.

![Figure 1.5: Typical heat release rates for 3 different inlet temperatures. Engine
conditions: 1050rpm, DOI = 2.3ms, SOI = 10°BTDC, bmep = 12bar](image1)

![Figure 1.6: Typical heat release integral for 3 different inlet temperatures. Engine
conditions: 1050rpm, DOI = 2.3ms, SOI = 10°BTDC, bmep = 12bar](image2)

### 1.4 Characteristic Mixing Rate

In order to quantify the diffusion combustion and the effect of combustion parameters
on the diffusion combustion, the mixing rate has been introduced, as explained in
[13]. The mixing rate is a parameter of the diffusion combustion, which describes its
speed of mixing and therefore burning, and is deduced from the HRR. The apparent
characteristic mixing time ($\tau_{mix}$) is a combustion parameter which can be used as a
measure of the apparent mixing during diffusion combustion. Given that during the
diffusion phase of combustion, the reaction rate is limited by the mixing of fuel with
air, the evaporated fuel can be related to the burned fuel using the characteristic
mixing time as a measure of the fuel/air mixing.

Following the above mentioned phenomena, the mixing rate is influenced by the
fluctuation intensity and thus a highly fluctuating cycle will exhibit a faster diffusion
combustion. In this context the mixing rate is defined as:
1. Introduction

\[
\frac{dQ_{\text{diff}}}{d\theta} = \frac{1}{\tau_{\text{mix}}} \cdot Q_{\text{available}}
\]  

(1.1)

\[
\tau_{\text{mix}} = \frac{Q_{\text{available}}}{dQ_{\text{diffusion}}} = \frac{m_{\text{fuel Evaporated}} - m_{\text{fuel Burned}}}{dm_{\text{fuel Diffusion}}} 
\]  

(1.2)

In the calculation of the characteristic mixing time, the mass of the evaporated fuel which is available at any given point is approximated using the \( d^2 \) law, which states that:

\[
d^2 = d_{0,\text{drop}}^2 - \beta \cdot t
\]  

(1.3)

where \( d_{\text{drop}} \) is the instantaneous droplet diameter, \( d_{0,\text{drop}} \) is the initial droplet diameter, assumed to be equal to the Sauter mean diameter and calculated using the Kamimoto correlation, and \( \beta \) is the evaporation coefficient. For the purposes of the analysis in this text \( \beta \) was kept constant for cycles with constant in-cylinder and injection conditions.

The parameter \( \tau_{\text{mix}} \) is a spatially global measure of air-fuel mixing during diffusion combustion, influenced by the level of turbulence in the spray and flame region, and by the rate of air entrainment into the spray plume not induced by turbulence. With the former affected by turbulence-enhancing parameters (injection pressure, background turbulence etc.) and the latter by varying of the shape of the spray. At constant charge and injection conditions, changes in \( \tau_{\text{mix}} \) are expected to be mainly due to increases in turbulence levels near the flame region, as only minor alterations in the spray itself are expected. The effect on the speed of combustion can be seen directly on Figure 1.5, which shows the mixing rates for the same conditions. From this figure it is evident that the absolute values of the mixing rate of the diffusion combustion are larger for the lower intake temperature conditions. That means that the higher proportion of premixed combustion at low intake temperature conditions influences the diffusion combustion by increasing its mixing rate. It is argued that this faster mixing during the diffusion combustion happens due to the increased pressure fluctuations generated by the premixed combustion, and thus results in higher combustion peak pressures and temperatures. Additionally, Figure 1.9 shows overlapped mixing rates for 581 consecutive cycles in a strongly fluctuating condition, proving that that at such conditions the speed of diffusion combustion from cycle to cycle is varying greatly.
1. Introduction

Figure 1.7: Typical mixing rates for three different inlet temperatures. Engine conditions: 1050rpm, DOI = 2.3ms, SOI = 10°BTDC, bmep = 12bar

1.5 Fluctuation Intensity & Pressure Rise Rates

The conditions of interest analyzed in this text are the ones which exhibit high cycle to cycle fluctuations, meaning a low intake temperature. Therefore the typical cylinder pressure traces for these conditions are shown in Figure 1.8, where pressure traces from 581 consecutive cycles have been overlapped. The figure shows the physical fluctuation of the pressure trace ('ringing') in the cylinder, with a natural frequency associated with the geometry of the cylinder, which can be described by the following formula from [27]:

\[ f_{m,n} = \frac{C \cdot \rho_{m,n}}{\pi \cdot B} \]  (1.4)

Where \( f_{m,n} \) is the specific vibration frequency for mode \((m, n)\) in [Hz], \( C \) is the local speed of sound [m/s], \( \rho_{m,n} \) is the vibration mode number and \( B \) is the cylinder bore [m]. The same authors [27] argue that the combustion chamber gases show the highest oscillation intensities in the first vibration mode, i.e. in the radial direction of propagation of the pressure waves. The vibration mode numbers are dependent on the cylinder geometries and are given in Figure 1.10.
1. Introduction

By observing further Figure 1.8 it can be seen that the variation in pressures are up to 25 bar or about 20% variation. This remarkable fluctuations are induced by very rapid pressure rise rates, coming from the strong premixed combustion contribution. Past research has already shown that rapid pressure rise rates due to rapid heat release have a direct influence on the fluctuation intensity or ringing of CI engines [28]. Further authors investigated the effect of rapid pressure rates on fluctuation intensity both computationally and experimentally [29], [30], showing that the maximum rate of pressure rise and ringing intensity are correlated. The result from [30] can be seen in Figure 1.11, where experimental measurements done with varying exhaust gas recirculation (EGR) rates, and therefore varying mixture reactivity, show an exponential correlation between the maximum rate of pressure rise and the fluctuation intensity.

For the purposes of this work, the fluctuation intensity has been quantified by taking a Fast Fourier Transform (FFT) of the cylinder pressure signal and taking the value

![Figure 1.8: Variations in the pressure trace for a high fluctuation intensity operating condition. $T_{in} = 20^\circ C$, 581 cycles](image)

![Figure 1.9: Variations in mixing rate for a high fluctuation intensity operating condition. $T_{in} = 20^\circ C$, 581 cycles](image)

![Figure 1.10: Different vibration mode shapes with accompanying mode factors ($\rho_{m,n}$) for a cylindrical chamber](image)
of the peak between 3000 and 4000 Hz, corresponding to the vibration modes of the measurement engine. A typical FFT for an average pegged cycle can be seen in Figure 1.12 with the relevant peak used for the quantifying the fluctuation intensity indicated by the arrow.

![Figure 1.11: Maximum rate of pressure rise on knock intensity [30]](image1)

![Figure 1.12: Fast Fourier Transform of the average cycle pegged](image2)

### 1.6 Pressure Variations Reduction

According to the above presented material [13] summarised the methods for reduction of in-cylinder pressure oscillations as follows:

- Reduce the amount of premixed combustion by reducing the ignition delay - through reducing the intake air temperature, using a higher cetane number fuel [31], using pilot injection to increase in-cylinder temperatures [32] or using a combustion chamber heating device.

- Reduce of the amount of premixed combustion by increasing the fuel injected during ID, using a lower injection pressure or shaping of the injection rate.

- Reduce the premixed combustion rate by mitigating the air-fuel mixing or influencing the mixture reactivity by using EGR [30]

Furthermore, in special combustion designs such as PCCI and HCCI operation, which operate under long ID conditions, a reduction in pressure variation can be obtained by increasing the homogeneity of the mixture or reducing reactivity through EGR and lean combustion. Finally, piston bowl designs also play a role in the pressure oscillations frequency as demonstrated by [29] and [33].
1. Introduction
2

Experimental Procedure

In this chapter the details of the various experiments undertaken to quantify the effects of the cycle-to-cycle combustion and emission variations will be discussed. The experiments were undertaken in the LAV laboratory of ETH Zurich, where an experimental MTU single-cylinder engine was made available for the purposes of this project. First the test bench and the devices used will be described and then their use in the various measurement campaigns will be explained, with a brief explanation of their working concept. Finally, the full experimental setup will be presented.

2.1 MTU-396 Single Cylinder Test Engine

The MTU single-cylinder test engine is a highly modified experimental engine, currently installed at the ETH LAV laboratory in the department of mechanical engineering at ETH Zurich. A front view of the engine setup can be seen in Figure 2.1, as seen from the control room. It is based on a MTU 396 series engine, with a modified fuel injection system (Ganser Common Rail). The specific test bench configuration allows for highly controllable inlet and outlet conditions, which allow simulating very varied combustion conditions. For the inlet side, the achievable inlet pressure is up to 5 bar and the inlet temperature from 17 to 100 degrees Celsius. Furthermore, the exhaust conditions are completely decoupled and independently controlled using a exhaust gas throttle. It needs to be mentioned that even though the original engine design comprised 4 valves, one exhaust valves was removed to enable the installation of measuring devices, the OLP sensor in this case. While this might generally result in reduced scavenging capabilities, in the case of the MTU test engine any such drawbacks are avoided due to the independent setting of the inlet and outlet conditions.

The relevant technical specifications of the engine can be seen in Table 2.1, while it is also important to mention the camshaft design of the engine. The camshaft allows for extreme Miller valve timing, with a very early intake valve closing, so that it is well suited for achieving low end of compression temperatures, important for obtaining fluctuating conditions, while maintaining the power density due to the high available boost pressure, provided artificially by a pre-compressed air storage tank.
2. Experimental Procedure

Table 2.1: MTU-396 single-cylinder test engine technical specifications

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td># Cylinder</td>
<td></td>
<td>1</td>
</tr>
<tr>
<td>Bore</td>
<td>mm</td>
<td>165</td>
</tr>
<tr>
<td>Stroke</td>
<td>mm</td>
<td>185</td>
</tr>
<tr>
<td>CR</td>
<td></td>
<td>13.7</td>
</tr>
<tr>
<td>Speed range</td>
<td>rpm</td>
<td>800-2100</td>
</tr>
<tr>
<td>Valves/Cyl</td>
<td></td>
<td>4 (3 effective)</td>
</tr>
<tr>
<td>Max injection pressure</td>
<td>bar</td>
<td>1600</td>
</tr>
<tr>
<td>Max cyl pressure</td>
<td>bar</td>
<td>155</td>
</tr>
<tr>
<td>Max injections per cycle</td>
<td></td>
<td>3</td>
</tr>
<tr>
<td># Injector orifices</td>
<td></td>
<td>8</td>
</tr>
<tr>
<td>Inlet pressure</td>
<td>bar</td>
<td>1-5</td>
</tr>
<tr>
<td>Inlet temperature</td>
<td>°C</td>
<td>17-100</td>
</tr>
</tbody>
</table>

Figure 2.1: Front view of the MTU engine - visible in this photo is the FastNO sampling head, which is installed on a tripod and connected directly to the exhaust of the MTU for cycle resolved NO measurement
2.2 Fast Sampling Valve

The Fast Sampling Valve (FSV) is an in-house gas sampling device developed during a PhD project at the LAV laboratory of ETHZ by Pascal Wilhelm in 2010 [18], and has been previously used for measuring cycle resolved exhaust gas samples. The FSV can be seen in Figure 2.2, installed on the exhaust pipe of the MTU test bench.

The basic design of the fast sampling valve resembles an injector-principle device, which uses pressurised diesel and a magnet to obtain very short opening durations and therefore sampling of very short intervals (theoretically below 1ms). The FSV requires a pressurised diesel supply for operation, which is provided through an external diesel pump, with operating pressure of 700 bar. Furthermore, it requires electrical power to actuate the magnet which pulls up a secondary valve, which consequently opens the main valve. The electrical power is provided via a DC source - a Pb-Ni battery, which can be seen in Figure 2.3. The battery is connected through to the magnet through a current amplifier device seen in Figure 2.4, which actuates the magnet. The actuation’s been coupled with the measurement test bench to ensure timely openings and automatic control.

Figure 2.2: Fast Sampling Valve installed on the MTU exhaust and connected to the DMS500
2. Experimental Procedure

An exploded view with the nomenclature of the main components of the FSV can be seen in Figure 2.5, where the FSV is disassembled during servicing. It is worth noting in this figure that the sampling head can be modified in length according to measurements needs, while the exhaust pipe adapter is used to attach the FSV to the MTU engine. The diesel inlet adapter is used to supply the FSV with pressurised diesel, while the outlet returns the diesel to the tank after it has passed through the fast sampling valve. The O-ring in the picture is a critical component which makes sure that the diesel side of the FSV is separated from the exhaust gas (sampling) side, in order to avoid diesel penetration in the sampled gas, which can result in possible contamination of the measurement devices. Therefore, the O-ring should be replaced at regular intervals. The purge inlet of the FSV allows for purging of the system with pressurised air and is used accordingly after every measurement. Additionally, the pretension pins and springs of the FSV adjust the pretension force which keeps the valve closed. This must be carefully adjusted as insufficient pretension will result in leakage of the exhaust gas in the FSV, while an over-tension can result in the FSV not opening.

**Figure 2.3:** Battery for the FSV magnet actuation

**Figure 2.4:** Current amplifier for the FSV magnet actuation

**Figure 2.5:** Exploded view of the FSV
Lastly, a proximity sensor placeholder is present in the figure, where a Micro-Epsilon proximity sensor was installed. The proximity sensor itself and the respective signal charge amplifier can be seen in Figures 2.6 and 2.7. The proximity sensor allowed for an accurate quantification of the FSV opening duration, timing and opening distance. Making sure that the FSV opens at the right time and stays open long enough (within a certain opening setting) to sample sufficient sampling gas to obtain a meaningful measurement. The fast sampling valve has been tested using a pressure vessel filled with NO gas, in order to characterise its performance, and the results of this analysis follow in the FSV Results section.

![Figure 2.6: Micro-Epsilon Proximity Sensor for the FSV](image)

![Figure 2.7: Miro-Epsilon charge signal amplifier for the proximity sensor](image)

### 2.2.1 Cambustion DMS500

In order to perform fast resolved soot measurements using the FSV, the CAMBUSTION DMS500 measurement devices has been used, as can be see in Figure 2.8, where the DMS500 is installed at the MTU test-bench. This device provides aerosol particle size measurement from 5nm to 1µm and fast response, with 10Hz data data recording and 200ms $T_{10-90\%}$. The device operates by gravimetrically correlating particle mass, meaning that the bigger particles, corresponding to a larger mass will travel further inside the electrometer and thus will activate the electrometer detectors further away from the inlet. A diagram of the basic operating principle of the system can be seen in Figure 2.10. Here a cross-section of the device measurement volume is shown, where the particles come in from the left and enter along the unipolar corona charger, transported by HEPA filtered sheath air flow towards the electrometers. Consequently, the particles will deposit alongside the sequential electrometers depending on their respective size (mass) and thus will excite the excited electrometers to produce a signal, signifying certain particle size has been detected. Consequently, the device outputs the sizes of both the nucleation and accumulation modes of the aerosol particles, as can be seen in Figure 2.9, where the DMS500 user interface is shown with the accumulation being the still liquid particles and accumulation being the solid particles.
2. Experimental Procedure

2.2.2 CAMBUSTION CLD500 Fast NO

In order to measure cycle resolved NO emission from the MTU engine, the CAMBUSTION CLD500 FastNO (sometimes known as an fNOx or fast NOx analyser) device was used. This device is specially suited for fast measurement, as it has specially designed sampling heads, where the NO oxidation reactions happen immediately in the vicinity of the probe, allowing a time response as low as $2\text{ms} \, T_{10-90\%}$. This allows the CLD500 to distinguish between two adjacent firing cycles, and even offer information about the variation in NO concentration during a single exhaust stroke. The device in mode of operation can be seen in Figure 2.11, with two NO measurement channels attached. The device has a limitation in the standard configuration, so that it is designed only for very low exhaust pressure conditions (typical for SI engines). This limitation was overcome with the support of CAMBUSTION Ltd. which provided additional capillary tubes, to be inserted in the measurement probe, which constrict the gas flow in the sampling head and thus reduce the pressure, so that the device can operate under a designed back pressure. The diameter of this hollow capillary add-on which was used was 0.13 mm, allowing for an good measurement signal even with engine back pressures of up to 2.6 bar.
2.2.3 AVL Photoacoustic Soot Sensor

In order to measure average tailpipe soot emissions, the AVL PASS Photo-acoustic Soot Sensor (PASS) was used, to measure the soot values in $mg/cm^3$. This device makes use of a modulated laser beam, acoustic standing wave and a microphone to measure soot concentration, it can namely detect only solid particles. The AVL PASS can be seen in Figure 2.12, as installed on the MTU test bench.

![CAMBUSTION Fast NO Analyzer](image1)

![AVL Photo Acoustic Soot Sensor (PASS)](image2)

2.3 Measurement Set-Up

The mentioned devices were all installed on the MTU test bench engine and operated together, in order to obtain simultaneous measurements for both NO and soot. Figure 2.13 shows the complete measurement setup with all the device attached. The FSV is attached via a fastening cage to the MTU exhaust pipe, and is connected to the diesel pump for pressurised diesel supply. The magnet of the FSV is connected to the respective battery and charge amplifier, as well as the D-Space control software for generating the opening signal. The purging outlet allows for remote purging of the FSV between measurements and can also be operated from the control room. Additionally, the CAMBUSTION DMS500 sampling head is attached to the FSV.
2. Experimental Procedure

sampling outlet, in order to capture cycle-to-cycle soot emissions. Finally, the CAMBUSTION CLD500 Fast NO is attached directly to the engine exhaust pipe via its sampling head, since the fast response of the device allows for cycle resolved NO measurements by measuring the the gas in the exhaust pipe immediately after the exhaust valve. Furthermore, the ancillary equipment mentioned in the above sections needed by the devices shown in Figure 2.13 has been attached to the MTU test bench and connected for remote control from the control room.

![Measurement setup on the MTU engine exhaust pipe, with the Fast NO and the DMS500 sampling head on the left and right, respectively, and the FSV in the middle. This measurement setup allows for simultaneous cycle-resolved NO and soot measurement, while having remote access to FSV opening, purging and proximity sensor signals from the control room in order to verify the operation of the employed devices.](image)

2.4 Measurement Protocol

The measurement protocol used during the experiments in this project follow multiple measurement campaigns, depending on the parameters which were investigated. A variety of intake temperature conditions, load, exhaust gas recirculation, start of injection, injection pressure and others has been used, and will be specifically provided in the results chapter, accompanying the corresponding measurement results. It is however worth noting that all the measurements were performed using the same camshaft, meaning a strong Miller effect is present throughout the measurements, which helps generating a low end of compression (EOC) temperature and fluctuating conditions.
2.5 Experimental Data Processing

The data which were recorded by the high frequency data acquisition system were used for the post-processing procedure. The post processing process is divided into two categories, involving the crank angle based data and the cycle-averaged resolved data. The first category refers to magnitudes such as cylinder pressure and heat release calculation, while the second category is referred to overall cycle values such as IMEP and emissions. The total post-processing procedure is performed by an in-house code that was developed at the Laboratory for Aerothermochemistry and Combustion (LAV). A single zone heat release rate analysis is applied for the elaboration of the cylinder pressure trace. The net heat release rate is calculated by [34]:

$$\frac{dQ_{\text{net}}}{d\phi} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\phi} + \frac{1}{\gamma - 1} V \frac{dp}{d\phi}$$  \hspace{1cm} (2.1)$$

where $p$ is the measured cylinder pressure, $\gamma$ is the specific heat ratio and $V$ is the instantaneous cylinder volume.

For calculation of the heat transfer coefficient the Woschni formula was used:

$$h_c = 3.26 \cdot B^{0.2} \cdot p^{0.8} \cdot T^{-0.55} \cdot w^{0.8}$$  \hspace{1cm} (2.2)$$

Where $B$ is the stroke, $T$ is the temperature and $w$ is the average gas velocity in the cylinder. The average gas velocity can be calculated as:

$$w = \left[ C_1 \bar{S}_p + C_2 \frac{V_d T_r}{p_r V_r} (p - p_m) \right]$$  \hspace{1cm} (2.3)$$

where $V_d$ is the displacement volume, $\bar{S}_p$ the mean piston speed $p$ is the instantaneous cylinder pressure, $p_r, V_r, T_r$ are the working-fluid pressure, volume and temperature at some reference state (inlet valve closing or start of combustion), and $p_m$ is the motored cylinder pressure at the same crank angle as $p$.

Where $C_1$ and $C_2$:

- For the gas exchange period: $C_1 = 6.18$, $C_2 = 0$
- For the gas compression period: $C_1 = 2.28$, $C_2 = 0$
- For the combustion and expansion period: $C_1 = 2.28$, $C_2 = 3.24 \cdot 10^{-3}$

Finally, gross heat release rate, which represents the fuel chemical energy, is calculated as the sum of net heat release rate and heat losses:

$$\frac{dQ_{\text{gross}}}{d\phi} = \frac{dQ_{\text{wall}}}{d\phi} + \frac{dQ_{\text{net}}}{d\phi}$$  \hspace{1cm} (2.4)$$

21
2. Experimental Procedure

Regarding the post processing of the cycle resolved data, the $IMEP_{net}$ is given by the integration of cylinder work (Eq. 7). Although $IMEP_{net}$ was calculated in this study, we refer to this as IMEP, for simplicity.

$$IMEP = \frac{\int_{IVC}^{EVO} pdV}{V_d}$$  \hspace{1cm} (2.5)

The single-cycle IMEP is subsequently used in the ISFC Vs. NO tradeoff with the cycle resolved data. These findings will be presented in the Results chapter along with the other NO results.

2.6 NO Signal Interpretation

Upon recording the transient NO measurement, a method has been devised to meaningfully interpret the recorded NO data. In order to so, the NO data and the in-cylinder pressure data were superimposed to gain an insight on the effective delay of the NO measurement and the characteristics of the signal. From Figure 2.14 it can be seen that the NO signal is rather constant, with the biggest jump happening in the gas exchange phase, when a certain amount of scavenging happens, and thus the fresh intake air produces the 'dip' in the signal which can be seen around 600°CA. Furthermore, it is important to note that the NO signal superimposed on the cylinder pressure trace actually corresponds to the previous (n-1) cycle, and not the superimposed upon, as the device samples the exhaust gases, immediately after the cylinder, while the exhaust valve is closed.

An example of processing the traces of cylinder pressure and NO concentration (from the same cylinder) is given with reference to Figure 2.14. During the valve-closed part of the current cycle, the analyser measures the NO level of the gas remained in the exhaust manifold from the previous cycle. When exhaust valve opens (EVO) a portion of the (in-cylinder) gas violently exits the cylinder (blow-down) and clears the exhaust manifold from the previous cycle gas. However, a delay between EVO and analyser response is observed, attributed to the distance between the valve outlet and the sampling probe, as well as to the instrument response time [35]. Its is further noteworthy that the NO concentration may also vary from cycle by cycle for non fluctuating conditions due to the bulk gas motion and local turbulence [36].

During the closed part of the next cycle, the analyser signal is constant and corresponds to the NO level of the current cycle. This mean value is taken as the NO emissions of the current cycle, as shown in Figure 2.14.
2. Experimental Procedure

DCA

Cylinder Pressure [bar]

0
50
100
600
800
1000
1200

NO Emissions [ppm]

Figure 2.14: Possible ranges of the NO signal to extract the representative value of NO for an individual cycle. The reference used is the TDC at 360 °CA.

2.6.1 Sensitivity Analysis & Validation

Thus, in order to extract a representative NO value for each cycle, a certain range of °CA in which the measurement is done is selected, using as a reference the TDC or 360 °CA. The representative value is then simply defined as the average value of NO emissions in the selected range. Multiple ranges were explored, from 20 °CA to 200 °CA as shown in Figure 2.14, and the respective sensitivity of the results has been analysed. The sensitivity has been tested using the cylinder pressure measurement as the relevant representative value, which is closely linearly correlated with the NO emissions. Thus the sensitivity of this (NO emissions vs. cylinder pressure) correlation on different °CA ranges of interest of the NO signal (in terms of °CA) has been investigated. The results of the sensitivity analysis can be seen in Figure 2.15, where the NO emissions vs. cylinder pressure correlation can be seen for various °CA ranges from which the representative NO value was extracted. On the same plots linear fits of the different data sets are plotted, with the very small difference in slope of the linear fits indicating a large similarity in data sets. Additionally, Figure 2.16, shows the maximum deltas (differences) in the NO emissions produced by any combination of relevant range choices, from 20°CA to 200 °CA. The absolute values of this deltas reaches up to only 30ppm, which suggests that the maximum errors created by a different choice of NO relevant range are in the order of 3%.
Consequently, the conclusion reached was that the sensitivity of the NO emission measurement on the chosen range is minimal, and thus for the sake of convenience a \( ^\circ \text{CA} \) range of 60 \( ^\circ \text{CA} \) (30\(^{\circ}\)CA after and before TDC ). This analysis was performed with critical importance, as it represents a fundamental assumption, upon which all consequent analysis is based on. Therefore this range was used in the development of the computer code for emissions analysis, and all of the following results make use of the individual cycle representative NO emissions extracted through this method.

![Figure 2.15: Sensitivity of the NO emissions vs cylinder pressure on the different \(^{\circ}\text{CA}\) ranges used for calculating the representative NO value](image1)

![Figure 2.16: Sensitivity of the NO emissions vs cylinder pressure on the different \(^{\circ}\text{CA}\) ranges used for calculating the representative NO value](image2)

**Validation** Additionally, the averages of the NO measurements obtained in this way with the fastNO device were compared to the average tailpipe measurements done by the GOLEM analyser (Figure 2.17), validating the data analysis procedure.

![Figure 2.17: Deviation of of FastNO Vs GOLEM measured soot averages](image3)
3 Results

In this chapter the results of this thesis are presented and observations are made on the findings. Firstly, the characterisation of the FSV used for soot measurement is explained, where the performance has been assessed under varied back-pressure conditions. Secondly, the cycle resolved NO results from the MTU test engine measurements are presented and discussed, with a special focus on the contribution of highly fluctuating cycles on the overall NO emissions of the engine. Lastly, results of cycle resolved soot measurements are presented with the respective insights.

3.1 Fast Sampling Valve Characterisation

In this section the results of the performance characterisation of the FSV will be presented. This has been done so that the sampling performance of the device for different exhaust pressure conditions could be quantified and considered when using the FSV on a working engine. The FSV was tested using a sealed pre-vacuumed pressure vessel - a controlled environment with a 500ppm NO gas, for a number of pressures and FSV opening durations. The measurement set-up can be seen in Figure 3.1, which shows the FSV installed on the pressure vessel with all the required ancillary equipment.

![Figure 3.1: Pressure vessel setup with diesel supply and Airsense connected](image)

![Figure 3.2: Pressure vessel characterisation - NO measurement equipment](image)

The equipment comprises a vacuum pump for vacuuming the pressure vessel, a diesel supply pump for providing high pressure diesel for the actuation of the FSV, a supply of pure NO and a NO measuring device - in this case the AirSense measurement...
3. Results

device from V&F Instruments Inc. The AirSense device can be seen in Figure 3.2, along with the transient recorder used to record the measurement data. The AirSense device uses the principle of ion collision for gas detection, with primary gases being Xe, Kr and Hg, allowing for a theoretical response time of 10ms. Furthermore, in the same figure the signal generator for the FSV opening and the FSV current amplifier can be seen, as they are required for precise control of the FSV via an actuation magnet which opens the valve.

![Graphs showing NO concentration over time with different repetition counts and average max NO values.](image)

**Figure 3.3:** FSV behaviour under a back-pressure of 1.5 bar with the pressure vessel filled with 500ppm NO gas

In order to understand the needed opening duration of the FSV, to obtain a satisfactory measurement under different back-pressure conditions, a variation in opening duration has been performed as well. The opening durations used are 20ms, 30ms, 40ms and 60ms, with 10 measurement repetitions for the 20ms measurement and 20 repetitions for the 30ms, 40ms and 60ms measurements. The sampled gas from the FSV is then analyzed via the mentioned AirSense device. Figure 3.3 shows the behaviour of the FSV for the mentioned opening durations for a back-pressure of 1.5 bar. It can be seen from the figure that the consecutive repetitions of the measurement give very consistent results, which suggests that the FSV opening is...
well controlled and behaves as expected. Furthermore, for the rather short opening times the measured value does not reach a stable value, while for an opening of 40ms and onwards we obtain a good measurement with significant duration of the signal. Even though the dilution value of the 20ms opening measurement seems quite low, that is only because the first transient peak was incorrectly captured as the relevant value, while it is in reality considered too short to obtain a meaningful absolute value reading. Therefore a value of at least 40ms is recommended for measurements with a back-pressure around 1.5 bar.

Secondly, Figure 3.4 shows how the relevant signal range becomes significantly broader for a back-pressure of 2 bar, allowing for a longer and more stable signal obtained from the AirSense. Thus, the measurement is considered to be of a higher quality and more representative.

![Figure 3.4: FSV behaviour under a back-pressure of 2 bar with the pressure vessel filled with 500ppm NO gas](image)

Thirdly, Figure 3.5, with 2.5 bar represents an upper limit of the back-pressure expected during experimental measurements on the MTU test bench and is thus considered to be the most relevant for further experiments. From the figure it is clear that the FSV measurement remains very repeatable and consistent for the high
back-pressure (with the exception of 1-2 outliers in the 20 measurement repetitions for the 40ms and 60ms opening times. Additionally, it can be seen that the opening of 60ms provides very small added value to the dilution performance, thus an opening times of 40ms is recommended for this conditions.

Lastly, Figure 3.6 shows the behaviour of the FSV in the case of an extreme back-pressure of 3 bar, which is not expected during MTU test bench experiments, but is still provided here for completeness. From the figure it is clear that the valve performs very well under such high back pressure conditions, and reaches a broad steady signal even for short opening durations. Repeatability is also not a problem for such conditions, but what presents itself as an issue is the possibility of imperfect sealing/closure of the valve. So that for high back-pressure conditions the valve always lets in some gas and there is a gas measurement even for closed valve conditions. However, with sufficient valve purging between every measurement, this effect should be minimised, as the leakage os rather small and requires a lot of time to significantly influence the measurement.

**Figure 3.5:** FSV behaviour under a back-pressure of 2.5 bar with the pressure vessel filled with 500ppm NO gas
Table 3.1: Dilution of the FSV for different filling pressures of the pressure vessel and FSV opening durations

<table>
<thead>
<tr>
<th>Opening Duration</th>
<th>Vessel Pressure</th>
<th>1.5 bar</th>
<th>2 bar</th>
<th>2.5 bar</th>
<th>3 bar</th>
</tr>
</thead>
<tbody>
<tr>
<td>20 ms</td>
<td></td>
<td>7.6 %</td>
<td>10.4 %</td>
<td>9.2 %</td>
<td>6.0 %</td>
</tr>
<tr>
<td>30 ms</td>
<td></td>
<td>8.8 %</td>
<td>10.0 %</td>
<td>7.2 %</td>
<td>4.6 %</td>
</tr>
<tr>
<td>40 ms</td>
<td></td>
<td>8.4 %</td>
<td>9.8 %</td>
<td>5.8 %</td>
<td>4.0 %</td>
</tr>
<tr>
<td>60 ms</td>
<td></td>
<td>6.2 %</td>
<td>7.8 %</td>
<td>5.4 %</td>
<td>3.4 %</td>
</tr>
</tbody>
</table>

In summary, the obtained dilution values for the different conditions can be seen in 3.1, where the dilution behaves as expected - diminishing with increasing opening times and higher pressure vessel pressures. It must be noted that the 1.5 bar back-pressure conditions falsely show lower dilution than in reality because of the more-transient measurement conditions due to the shorter signal, especially the measurement for...
20 ms is not realistic as the false peak has been captured. From this analysis we conclude that the FSV valve performs well for sampling gases from a pressurised vessel, and is expected to perform satisfactory also for engine exhaust measurements. However, with soot measurements there are additional challenges due to the dilution and deposition on the measurement device. Therefore the suitability of the available FSV for particulate measurements can only be confirmed with once mounted on the final measurement setup at the MTU engine. There the particle dilution of the fast sampling device will be tested by performing soot emissions measurements on sampled gases.

### 3.2 NO Results

In this section the results of the cycle resolved NO measurements on the MTU test bench will be presented. As explained in the experimental methods chapter, the cycle resolved NO measurements were done for a variety of operating conditions of the engine, which were used to investigate various effects on the cycle to cycle emission. This section will first describe the effect of pilot injection on the cycle to cycle emission variations, then the fluctuation intensity effect on NO emissions will be explained on a single cycle basis, where other combustion parameters will be included. Afterwards the effect of intake temperature will be systematically analysed and its influence on the NO emissions quantified, with a special consideration for highly fluctuating cycles. Furthermore, the EGR effect on the relevant combustion parameters and ultimately NO emissions will be explained. Thereafter, the effect of load variation on the combustion parameters and the NO emission will be presented, with emphasis on the influence of fluctuation on the diffusion combustion. Lastly, an analysis of the Indicated Specific Fuel Consumption (ISFC) versus NO emissions tradeoff will be provided, visualising how the highly fluctuating cycles affect the tradeoff curve.

**Table 3.2:** Engine Measurement Conditions for the NO measurements on the MTU test engine. The respective values which have been varied for different measurement points are shown as ranges.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed</td>
<td>rpm</td>
<td>1050</td>
</tr>
<tr>
<td>Duration of Injection (DOI)</td>
<td>ms</td>
<td>1.1-2.3</td>
</tr>
<tr>
<td>Injection Pressure ($p_{in}$)</td>
<td>bar</td>
<td>1000-1500</td>
</tr>
<tr>
<td>Start of Injection (SOI)</td>
<td>BTDC$^\circ$</td>
<td>6-14</td>
</tr>
<tr>
<td>Intake Temperature ($T_{in}$)</td>
<td>$^\circ$</td>
<td>20-80</td>
</tr>
<tr>
<td>Intake Pressure ($p_{in}$)</td>
<td>bar</td>
<td>2.9 - 3.5</td>
</tr>
<tr>
<td>Exhaust Pressure ($p_{out}$)</td>
<td>bar</td>
<td>2.2-2.5</td>
</tr>
<tr>
<td>Brake Mean Effective Pressure (bmep)</td>
<td>bar</td>
<td>12</td>
</tr>
<tr>
<td>Fuel Consumption</td>
<td>kg/h</td>
<td>5-9</td>
</tr>
<tr>
<td>Air Consumption</td>
<td>kg/h</td>
<td>254</td>
</tr>
</tbody>
</table>
3. Results

3.2.1 Effect of Pilot Injection on Cyclic NO Emissions

In order to understand the behaviour and variation of the NO signal from consecutive cycles, an analysis has been made where the NO signal has been compared between two engine operating mode, with and without using pilot injection. This was done to better understand the effect of precursor fuel on the fluctuation intensity and consequently NO emission variation. The effect of the pilot injection can be seen in the following figures, where Figure 3.8 shows the NO signal without pilot injection present, resulting in rather high variations in the cycle-to-cycle NO emissions. The graphs show the pressure traces of 10 consecutive cycles taken from a transient recording of 144 cycles, superimposed on the recorded NO signal, while a third line shows the average NO value of the entire transient measurement series of 144 cycles. On the other hand, Figure 3.7 shows the same condition, but with an introduced pilot injection. It follows from the figure that the cycle-to-cycle variations in NO emissions are strongly attenuated, since the pilot injection contributes to greater combustion uniformity and less variations in the cycle-to-cycle pressure trace. This confirms that a pre-injection, as was stated in the Introduction chapter, is a very effective technique to reduce not only cycle to cycle combustion variation and achieve more constant performance and possibly less noise, but also to reduce the variation of cycle to cycle NO emissions.

![Figure 3.7: NO signal sample from a measurement point with pilot injection. Measurement conditions: 1050rpm, $m_{fuel} = 9$kg/h, bmep = 12, DOI = 1.97ms, $p_{inj} = 1000$bar, SOI = 10deg BTDC, $T_{in} = 20$ C, $p_{in} = 2.9$ bar, $p_{in} = 2.2$ bar](image-url)
3. Results

3.2.2 Nitrogen Oxide Emission Correlations

As shown in the previous subsection, each individual cycle produces a different amount of NO, with the amplitude of the variation being dependent on the operating conditions. In this subsection the individual cycle resolved NO emissions have been analysed in regard to relevant combustion parameters (explained in the Introduction chapter) and the respective correlations have been analysed. The starting point was the cylinder peak pressure, the increase of which creates favourable conditions for higher NO emissions, such as high temperatures and oxygen availability through high fluctuations. Therefore, as was expected from previous observations by [13], the NO vs smoothed peak pressures for individual cycles exhibit a linear correlation as shown in Figure 3.9. This result is expected to be perfectly linear in theory [34], but there is a certain spread present in the measurement results due to the measurement accuracies of both the CAMBUSTION fastNO analysis device and the cylinder pressure sensor. Furthermore, Figure 3.10 shows how does the NO emission related to the fluctuation intensity of individual cycles, which is a direct product of the ringing pressure trace. It is observed that the similar near correlation exists, however there is significant spread above the expected line, while there is no similar spread below. This result is believed to be caused by the measurement limitations, which is due to the fact that there is only one pressure sensor installed on the MTU.
3. Results

test engine. Therefore, the pressure fluctuations in the combustion chamber are not properly captured for all the cycles, as there are some cycles which might have the node of the vibration mode exactly in the vicinity of the pressure sensor. Therefore they would be falsely measured as low fluctuating, and they are represented by the spread on the top left of Figure 3.10, which shows low fluctuating cycles with high NO emissions. It is therefore expected that the fluctuation intensities can be better captured by using an improved setup with two pressure sensors at different locations (e.g., by using the higher value of the 2 pressure sensors). Thus, the expected results will lie on the red trend line as presented on Figure 3.10.

![Figure 3.9: Correlation between NO and smoothed peak pressure - 581 cycles. The expected trend is perfectly linear, spread only due to measurement error.](image)

![Figure 3.10: Correlation between NO and fluctuation intensity. The red trend line shows the expected correlation when using two pressure sensors instead of one](image)

Following the causality of the phenomena, where the fluctuation intensity enhances the characteristic mixing rate of the diffusion combustion, it can be seen that the mixing rate follows the above trend set by the peak pressure and fluctuation intensity and linearly scales with the NO emissions. A highly fluctuating cycle will result in a higher mixing rate and therefore faster diffusion combustion, generating more NO due to the higher temperatures produced.

Additionally, it is important to note that the distribution of the single cycles by fluctuation intensity roughly follows a Beta distribution function, and some statistical analysis has been done in order to group the cycles according to their fluctuation intensities. The distribution of the cycles in terms of fluctuation intensity can be seen in Figure 3.12, where the Beta fit is visible on a histogram of the number of cycles per each fluctuation level. The grouping of the cycles itself will be explained in the next subsection.

The remarkable result from the above mentioned figures is that an increase in cylinder pressure of about 10 bar or 10% in this case is responsible for an increase in NO from about 860 to 1500 ppm or about 75% percent. Suggesting that the strong fluctuation
induced by the rapid pressure rises can account for a significant total NO increase.

**Figure 3.11**: Correlation between NO and characteristic mixing rate during the diffusion combustion - 581 cycles

**Figure 3.12**: Distribution of fluctuations intensities for 581 cyles with Beta distribution fit

### 3.2.3 Fluctuation Intensity influence on NO

In order to better quantify the effect of the individual fluctuating cycles on the total engine tailpipe NO emissions, a characterisation of the cycles has been made according to their fluctuation intensity. Using the above mentioned Beta distribution, a mean fluctuation density has been defined, as well as a standard deviation of the data set. Using the standard deviation as a limit, three distinct groups have been defined as follows:

1. **Above 1\(\sigma\)** - group of cycles with a fluctuation intensity more than one standard deviation above the mean fluctuation value
2. **Below 1\(\sigma\)** - group of cycles with a fluctuation intensity more than one standard deviation below the mean fluctuation value
3. **Within 1\(\sigma\)** - group of cycles with a fluctuation intensity within one standard deviation of the mean fluctuation value

These three distinct groups of cycles have also their distinct influence on the NO emissions, as shown in Figure 3.13, where the three groups identified by the above criteria are colour coded and plotted against their NO emissions for a low intake temperature condition. The figure shows that the 'Below 1\(\sigma\)' group exhibits not only a low fluctuation intensity, but that the spread in NO emission of this group is quite limited, while the spread of both the 'Within 1\(\sigma\)' and 'Above 1\(\sigma\)' groups is quite large. However, this behaviour is changed when the intake temperature, conditions are altered, which can be directly seen in Figure 3.14, where two measurement sets for two different intake conditions are overlapped. From such a comparison it can be concluded that the higher intake temperature not only strongly diminishes the
overall fluctuation intensity, but also attenuates the spread in NO emissions of each of the three fluctuation groups. Indeed in the $T_{in} = 80^\circ C$ case, the amplitude of the NO emissions for the three respective groups is roughly identical. In the following subsection, the temperature influence on the fluctuation intensity groups will be discussed in more detail.

Figure 3.13: Individual cycles of a measurement point at $T_{in} = 20^\circ C$, color coded according to their fluctuation intensity deviation from the average. Green - fluctuations below 1 standard deviation from the mean of all the cycles. Red - fluctuations above 1 standard deviation from the mean of all the cycles. Blue - fluctuations within 1 standard deviation from the mean of all the cycles. Dashed lines are mean values of respective groups.
3. Results

Figure 3.14: Individual cycles of a measurement point, color coded according to their fluctuation intensity deviation from the average. Green - fluctuations below 1 standard deviation from the average of all the cycles. Red - fluctuations above 1 standard deviation from the average of all the cycles. Blue - fluctuations within 1 standard deviation from the average of all the cycles. Light colors are $T = 80^\circ C$, dark colors are $T = 20^\circ C$

3.2.4 Intake Temperature Influence on NO emissions

In this subsection the influence of intake temperature on the NO emissions will be discussed, specifically how the temperature influences each of the three fluctuation groups defined above. In order to make such an analysis, an experimental campaign has been done, in which an intake temperature sweep has been done. This allowed to quantify the influence of the intake temperature, while keeping other conditions constant and observe the behaviour of each of the fluctuation groups individually. Figure 3.15 shows such an intake temperature sweep influence on absolute NO emissions of respective cycles. Two main effects are important to recognise from this plot. The first one is that the general NO trend first decreases starting from a high intake temperature ($T_{in} = 80^\circ C$), and then increases after a certain threshold of about ($T_{in} = 50^\circ C$), following a quadratic trend. The second one is that the difference in absolute NO emissions between the groups increases considerably with decreasing inlet temperature. So that while at $T_{in} = 80^\circ C$, all the groups show roughly identical NO emissions, at a low intake conditions such as $T_{in} = 20^\circ C$ this difference becomes significant, with more than 200 ppm between the "Above 1σ" and "Below 1σ" cycle groups (red and green line, respectively).

The second relevant result in this analysis Figure 3.16, shows how the standard deviation of the NO emissions for the respective groups changes with intake temperature. It can be concluded that the STD of the 'Within 1σ' and the 'Below 1σ' groups is the same for high intake temperature conditions, while for the 'Above 1σ'...
group it is slightly higher, considering that for those conditions the absolute values of the standard deviation for all groups are rather low. Furthermore, with decreasing inlet temperature conditions, the absolute values of the standard deviations increase quasi-exponentially, with the standard deviations of the three groups deviating from one another. Additionally, at low intake temperatures, the 'Within 1σ' fluctuation group, which at high intake temperatures exhibited a STD close to the STD of the 'Below 1σ', exhibits a STD close to the STD of the 'Above 1σ'. This result confirms what was already discerned in Figure 3.13 and Figure 3.14, where the NO STD of the 'Within 1σ' for the low intake temperature condition was already observed to be similar to the 'Above 1σ' and for the high intake temperature condition similar to the 'Below 1σ' NO STD.

![Figure 3.15: Effect of the intake temperature on the absolute NO emissions of the MTU engine](image1)

![Figure 3.16: Effect of the intake temperature on the STD of NO emissions of the MTU engine](image2)

### 3.2.5 EGR Effect on NO through Ignition Delay

In this section the effect of EGR (Exhaust Gas Recirculation) on the ignition delay will be discussed. The EGR has long been a well-known method to reduce NO emissions through a combination of effects of reducing the combustion temperature and providing less available oxygen for NO production. Here the effect of EGR on various aspects of the combustion is presented, starting with Figure 3.17, which shows the effect of EGR on the standard deviation of the peak pressure, confirming that the EGR has a direct effect of decreasing the pressure fluctuations, and therefore creating a more uniform combustion. Furthermore, it is also discernible from the figure that the biggest benefit from using EGR come for the first step, while subsequent increasing amounts of EGR produce diminishing benefits. This is true not only for the STD of peak pressure but also for the subsequent parameters, with Figures 3.18 and 3.19, show similar effects of the EGR on the fluctuation intensity, since the fluctuation intensity is expected to be directly a consequence of the decreased STD of peak pressure. Thus, both the average and the maximum fluctuation intensity
exhibit a significant dependence on the EGR rate, with the initial introduction of EGR giving the largest relative decrease in fluctuation intensity.

![Figure 3.17: Different oxygen concentrations correspond to different EGR levels which influence the combustion characteristics. Higher EGR brings a more uniform combustion with lower deviation of cycle-to-cycle pressure traces.](image)

Figure 3.18: Effect of EGR amount on the average fluctuation intensity

Figure 3.19: Effect of EGR amount on the maximum fluctuation intensity

Most importantly, this EGR effects influence both directly and indirectly the NO emissions for the respective measurement points, with the resulting effects as shown in Figures 3.20 and 3.21. The resulting NO emissions STD are greatly attenuated by the introduction of EGR, as are the absolute values of the NO amount, suggesting that STD of the NO emissions an important parameter to predict absolute NO emissions.
3. Results

![Figure 3.20: Effect of EGR amount on STD of NO emissions](image1)

![Figure 3.21: Effect of EGR amount on the amount of NO emissions](image2)

3.2.6 Effect of Load on NO Emissions through Fluctuations Intensity

In this section the effect of load on the fluctuation intensity and NO emissions will be presented, with special consideration how do the pressure fluctuations influence the diffusion combustion and thus NO emissions.

The first result to understand the further analysis concerns the fluctuation intensity behaviour for varying load conditions, with load in the present example being represented with the duration of injection (DOI), all other operational parameters being constant. Thus, Figure 3.22 shows that the fluctuation intensity is rather constant for various loads, with a slight downward trend for high load cases. The very low load point of DOI = 1.1 ms is explained as a very unstable combustion point, where the fluctuation intensity is not correctly captured. This instability of the combustion can be seen by measuring the CO emission values, as shown in Figure 3.24, which shows that for the low load conditions there is a rapid increase of CO emissions, indicating an instability of combustion. Furthermore, the STD of the peak pressure shows an increasing trend for increasing load, upon which it reaches a given constant value and doesn’t increase further upon increasing the load. Again, the two very load points, show high standard deviations because of the combustion instability.
3. Results

![Graph](image1)

**Figure 3.22:** Effect of load variation on fluctuation intensity, with the low load conditions showing unstable combustion and false readings

![Graph](image2)

**Figure 3.23:** Effect of varying load on the STD of peak pressure, with low loads showing combustion instability

**Figure 3.24:** CO emissions for varying load conditions, with high CO values for unstable combustion

The effect of the load on average NO emission can be seen in Figure 3.25, where the NO emissions rapidly increase with the load, as expected from theory. Additionally, in Figure 3.26 it can be seen that not only the absolute value but also the standard deviation of the cycle resolved NO values increases with increasing load and then reaches a constant plateau and even slightly decreases. The explanation for this follows with the subsequent results.
In order to separate the effect of the load on the previously defined groups of cycles by fluctuation intensity the following diagrams were created, where multiple measurement points were used, one for each load condition. For each of this measurement point, the 581 cycles recorded were divided into the 'Above 1σ', 'Below 1σ' and 'Within 1σ' groups, and their respective averages have been found. These averages are presented in Figure 3.25, by the red, green and blue lines, while the black and magenta lines represent one single high and low extreme cycle from the 581 measured for each MP, respectively. In other terms, red, green and blue are averages of many cycles in the respective groups, while black and magenta represent one single cycle for each load condition.

Figure 3.25: Effect of load on absolute NO emissions

Figure 3.26: Effect of load on the STD of the NO emissions

The Figure 3.27, shows that for increasing injection duration (which translates to higher load) the NO values increase, as expected. What is of important to note is the 'critical point' at a load of about $DOI = 1.4ms$, until which the NO values are virtually identical for each of the three characteristic fluctuation groups. This behaviour is explained by the fact that up to this point, the load is low enough that the entire combustion process is composed almost entirely from a premixed combustion, as shown by the HRR in Figure 3.30. From this figure it is clear that for the low loads of $DOI = 1.1ms$ and $DOI = 1.25ms$ practically the entire HRR is formed by a premixed combustion. After the critical point of $DOI = 1.4ms$ in Figure 3.27, it can be seen that the NO emissions for the three characteristic fluctuation groups start deviating from each other, reach a given deviation, and then keep the same constant deviation of increasing load, while increasing the absolute NO emissions with higher load.

Even more insightful is Figure 3.28, which builds up on the previous results and shows the similar behaviour, but shows the NO emissions mass created per mass of fuel used. In this figure we observe the similar behaviour with the 'critical point' after which the emissions of the three characteristic fluctuation groups stabilise and gradually decrease with higher loads. This decreasing of NO mass emitted per mass of fuel used is explained by referring again to Figure 3.30, which shows also the
HRR for the measurement points above the "critical point". It should be noted that after this 'critical point' the diffusion combustion proportion of the combustion process increases steadily with increasing load. So that measurement points with DOI = 1.55ms, DOI = 1.85ms and DOI = 2.15ms show increasingly larger diffusion combustion parts, which are then affected by the fluctuation intensities generated by the premixed combustion part. This can also be seen from Figure 3.29, where the heat release rate integrals are plotted for the different loads, suggesting that the higher load cases will have a lower ignition delay, but also a lower slope and therefore lower premixed combustion and combustion intensity, which will eventually produce less fluctuation intensity.

This fluctuation intensities create high temperatures during the diffusion combustion and thus high NO in some cycles, so that the average NO emissions, reach a peak at a load of around DOI = 1.6ms, where there exist both significant premixed and some diffusion combustion. The downward trend after the maximum is then due to the fact that with further increasing load the premixed phase decreases in proportion, generating lower fluctuation intensities, and secondly due to the fact that any additional fuel added for a longer DOI will be added at the end of the diffusion combustion phase, where the conditions for NO creation are not supportive, since the temperatures are lower and the residence time of this fuel is short. This indeed means that even though this high load conditions are not optimal for fun efficiency, the NO mass created per fuel mass will decrease for high loads.

This result here presented confirms the hypothesis that the fluctuation intensities are created by the premixed combustion phase and have subsequently effect on the diffusion combustion phase in terms of influencing a certain number of cycles to create high NO emissions, as shown for the loads above the "critical point" of DOI = 1.4ms. On the other hand, with combustion regimes below DOI = 1.4ms the fluctuating intensities have no diffusion phase to influence, and therefore the NO emissions of all the cycles are virtually identical, excluding some random outliers.
3. Results

![Graph showing NO emissions as a function of DOI (milliseconds)]

**Figure 3.27:** Influence of the load on the absolute NO Emissions of the engine, with separately considered groups of cycles in a certain range of fluctuation intensity.

![Graph showing NO emissions per fuel used (g/kg) as a function of DOI (milliseconds)]

**Figure 3.28:** Influence of the load on the specific NO Emissions of the engine, with separately considered groups of cycles in a certain range of fluctuation intensity.

![Graph showing HRR integrals for different load conditions, with high load conditions showing less ignition delay and a less steep HRR slope](#)

**Figure 3.29:** HRR integrals for different load conditions, with the high load conditions showing markedly less ignition delay and a less steep HRR slope.

![Graph showing HRR rates for different load conditions, with the diffusion combustion proportion visibly increasing with higher load](#)

**Figure 3.30:** HRR rates for different load conditions, with the diffusion combustion proportion visibly increasing with higher load.

### 3.2.7 Effect of Start of Injection on the ISFC - NO Tradeoff

In order to quantify the effect of the SOI timing on the NO vs ISFC tradeoff a measurement campaign with a SOI sweep has been made, where the SOI has been varied from 6 to 14 degrees before top dead centre. During this campaign, single cycle measurements have been made and are plotted respectively in the individual Figures 3.31 to 3.34, with the orange line being a fit of the averages of each of the 5...
groups. This graphs shows how the individual cycles form a certain spread with the given tradeoff slope. With early SOI settings showing a near-horizontal spread, so that cycles with the same specific fuel consumption exhibit a large spread in possible NO emissions, partly due to the highly fluctuating cycles. On the other end, with very late SOI setting as in Figure 3.31, the possible variation of NO for a given efficiency is rather small, due to a shorter ignition delay and smaller fluctuations.

Finally, in Figure 3.36, the entire sweep has been overlapped, showing that the cycles for each condition in the sweep follow the expected Efficiency Vs NO Tradeoff.

Figure 3.31: Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages

Figure 3.32: Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages

Figure 3.33: Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages

Figure 3.34: Individual cycle spread on the ISFC vs NO tradeoff curve for a single SOI setting; line fits all averages
After having analyzed the sweep containing all the cycles, the same grouping of the cycles has been performed as in the previous results, where the cycles were separated into three groups according to the fluctuation intensity they exhibited. This grouping provides the insight into the negative contribution of the highly fluctuating cycles on the overall efficiency Vs. NO emissions tradeoff. It can be seen from Figure 3.37 that the highly fluctuating cycles perform worse than the low fluctuating cycles on the tradeoff, influencing the average emissions pulling them up towards a less beneficial tradeoff. Furthermore, the difference between the three different tradeoff curves (for each fluctuation intensity group) is very significant for late SOI timings (top left points analogous to Figure 3.36, while becoming less so for early SOI timings, where the engine efficiency is increased, but so are the NO emissions.

This indicates that at the low efficiency, late SOI conditions, the fluctuation intensity do not play a significant role, since the diffusion combustion is markedly shorter, as shown on Figure 3.38 and therefore there is a lack of time for the fluctuation intensities to influence the diffusion combustion. By consequence, the bulk of the NO will be already formed in the premixed combustion phase, and the additional NO produced in the diffusion phase by the effect of the fluctuation intensities will be smaller in magnitude. Therefore, that explains why the bottom right points in Figure 3.37 for early SOI are overlapping with only small differences between them.
3.2.8 Effect of Injection Pressure variation on the ISFC Vs. NO Tradeoff

A further study, which has been made on the ISFC vs. NO tradeoff concerns the behaviour under different injection pressure conditions. It is known from theory and best practices that a higher injection pressure will increase the engine efficiency for standard operating conditions, however it is worth analysing what happens to the tradeoff not only with varying injection pressures but also with cold intake conditions. Thus in Figure 3.39, a classical situation is plotted where the injection pressure increase enhances the engine efficiency, albeit at a cost of higher NO emissions. On the other hand, by looking at Figure 3.39, it can be deduced that for cold intake conditions the benefit of the higher injection pressures is rather limited. The spread in the x-direction (NO emissions) of the individually measured cycles does increase with lowering intake temperature, but the benefit of decreasing ISFC with higher injection pressure is minimal. Effectively resulting on average in an increase of NO emission of more than 50%, for an ISFC decrease of about 1%, as shown by the horizontal lines which indicate the mean ISFC values of the respective groups of measured cycles. Consequently, this analysis suggests that using a high injection pressure at low intake temperature conditions should be avoided for effective engine use.
3. Results

Figure 3.39: Tradeoff for different injection pressures at hot intake conditions. Points are individually measured cycles

Figure 3.40: Tradeoff for different injection pressures at cold intake conditions. Points are individually measured cycles

3.3 Soot Measurement Results

In this section the results of the soot measurement campaigns will be presented and the insights explained. As explained in the methods section, the soot measurements has been done by using the FSV device along with the DMS500 fast response particle analyser. The relevant engine conditions for the conducted measurements can be seen in Table 3.3, where a set of three different conditions has been used to produce different particle emissions.

Table 3.3: Measurement Conditions for the direct soot measurement at the MTU engine, using the CAMBUSTION DMS500, both directly connected and with the FSV

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Unit</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed</td>
<td>rpm</td>
<td>1050</td>
</tr>
<tr>
<td>Duration of Injection (DOI)</td>
<td>ms</td>
<td>1.65</td>
</tr>
<tr>
<td>Injection Pressure ($p_{in}$)</td>
<td>bar</td>
<td>1000</td>
</tr>
<tr>
<td>Start of Injection (SOI)</td>
<td>BTDC°</td>
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</tr>
<tr>
<td>Intake Temperature ($T_{in}$)</td>
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<tr>
<td>Intake Pressure ($p_{in}$)</td>
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<td>2.6</td>
</tr>
<tr>
<td>Exhaust Pressure ($p_{out}$)</td>
<td>bar</td>
<td>1.9</td>
</tr>
<tr>
<td>Brake Mean Effective Pressure (bmep)</td>
<td></td>
<td>9</td>
</tr>
<tr>
<td>Fuel Consumption</td>
<td>kg/h</td>
<td>6.6</td>
</tr>
<tr>
<td>Air Consumption</td>
<td>kg/h</td>
<td>120-150</td>
</tr>
<tr>
<td>Exhaust Gas Recirculation (EGR)</td>
<td>kg/h</td>
<td>80-110</td>
</tr>
</tbody>
</table>
3. Results

3.3.1 DMS500 - Direct & FSV Configuration

In order to quantify the dilution of the FSV, first a comparison has been made between the measured signal of the DMS500 directly attached to the exhaust pipe, and the signal from the DMS500 when it was used to sample through the FSV. With the first being called the "Direct" configuration and the second the "FSV" configuration. The results of this comparison in terms of Size Spectral Density (SSD) for three different engine conditions can be seen in Figures 3.41, 3.42 and 3.43. Where each measurement condition has been measured with the DMS500 "Direct" configuration and 4 repetitions of the "FSV" configuration. The signal for the "Direct" measurement is actually an averaged signal over the measurement time of about 1162 combustion cycles, while each of the repetitions in the FSV configurations is actually an averaged signal over 20-30 openings of the FSV, which was actuated during engine operation at stable conditions.

The figures show that the SSD of the FSV signal follow the trend of the direct measurement and exhibits a relatively constant dilution ratio of about 4 (seen more accurately for each measurement point in Figure 3.47). However, measurement issues come from the direct measurement instability itself. For example it can be seen from Figures 3.46 and Figure 3.44 that the DMS500 in "Direct" configuration measures cycles with high emission variation, as expected due to the increased mixture reactivity (low EGR rates). On the other hand, Figure 3.45 shows a low variation condition with high EGR rates of 110 [kg/h] and is there expected to be rather stable. However, there is significant emission deviations present (up to 50%) and thus is definitely not stable as expected. This suggests that the measurement device has insufficient resolution power/speed to accurately capture the particulate emission on a cycle to cycle basis and thus limits the possibilities of further measurement campaigns.

![Figure 3.41: SSD with FSV vs Direct Measurement 1-4](image1)

![Figure 3.42: SSD with FSV vs Direct Measurement 5-8](image2)

![Figure 3.43: SSD with FSV vs Direct Measurement 9-12](image3)
3. Results

**Figure 3.44:** DMS Direct configuration deviation 1-4

**Figure 3.45:** DMS Direct configuration deviation 5-8

**Figure 3.46:** DMS Direct configuration deviation 9-12

![Graphs showing particle density vs. diameter](image)

**Figure 3.47:** Dilution ratio for all 12 measurement points in the campaign

### 3.3.2 FSV Openings Analysis

An additional analysis has been made with further measurement campaigns, in the FSV configuration, where the measured values for each FSV opening were analysed. As the FSV was operated in sync with the engine control unit, and opened during the opening of the exhaust valve, it is expected to capture the representative values of the given cycle. Typical single cycle values can be seen in Figures 3.48 3.49, which show the particle concentration weighted by number and by mass, respectively. From this figures it can be seen that there are significant deviation between the cycle measure values, and they are not only attributed to the actual emission variation, but to the instability of the measurement setup. Especially the measured particle mass shows extremely large variations during the measurement series, which suggest that the measurements lack validity and the FSV setup is inconsistent. As mentioned in the methodology section, a proximity sensor has been used to assure an appropriate opening of the FSV, which was achieved according to the sensor. However, the particle measurement signal was still inconsistent.
3. Results

In order to get more insight into the measurement signals of the individual cycles the Spectral Size Density and the time resolved mass have been plotted for the individual cycles in Figures 3.50 and 3.51. Here it can be seen that the measurements are rather inconsistent, again, the particle number showing more stability than the mass measurement, which is highly inconsistent. Furthermore, it should be noted how the SSDs were plotted from the time-resolved signal of the DMS500 device. This is shown in Figure 3.52, which shows three most prominent signals (curves) for a given cycle, since the temporal resolution of the DMS500 allows for about 10 signal curves per cycle. This most prominent SSD lines are then fitted on an artificially refined grid and averaged. Consequently, an interpolant is plotted through the averaged values and that represents the representative value for the SSD.

This number and mass measurements of soot particles have been plotted with relevant combustion parameters in order to check for correlation between the combustion characteristics and the soot emissions. A number of different parameters have been selected and they can be seen in the below figures, where the first group in Figure 3.53 shows correlations with particle size, while the second group in Figure 3.54 shows the correlations with the particle number and therefore mass. Due to the previously mentioned limitations of the measurement setup, the results remain inconclusive, however there are some indications of trends, which will be discussed here and for which it is suggested to make further measurements in order to confirm or disprove them. For the purposes of discussing these results, only measurements points which show a particle size above 20nm were considered as relevant according to [37], even though all measurement signals are plotted, because this low particle size signals are understood to be coming from the lubrication oil particles.

From Figure 3.53(a), where the geometrical soot size is plotted against the smoothed peak pressure, a trend can be discerned, where an increase in peak pressure causes a decrease in particle size, which would agree with the expected result as higher peak
In terms of particle number, a similar analysis has been performed, where the total particle number has been plotted against the same parameters. The results of this analysis can be seen in Figure 3.54, where in 3.54(a) the effect of peak pressure gives a trend of increasing the soot particle total number. This result is expected as higher pressure creates higher temperatures and fluctuations as mentioned before, which facilitates soot oxidation and hinders soot agglomeration. Therefore, the larger particles will decrease their size by oxidation and the small particles will not agglomerate in a lower amount, resulting in a larger number number of smaller particles for higher peak cylinder pressures. Furthermore, 3.54(b) shows the direct effect of the increased peak pressure, the fluctuation intensity which is proportional to the particle number measured. This follows the fact that fluctuation intensity favours oxidation of soot particles by facilitated mixing with oxygen. Consequently, the fluctuations further enhance the mixing rather, resulting in a positive correlation between mixing rate and soot particle total number, as a faster more energetic combustion will positively affect the oxidation, giving a higher soot total number.
Figure 3.52: Interpolation method used for obtaining the relevant SDD for a measurement point. The method uses an artificial grid refinement of the 3 most prominent measurements for a given cycle and then creates a smoothed average of the 3 signals.

as shown in 3.54(c). Finally, 3.54(d) shows how the cycles with higher NO values produced have also a higher number of particles, albeit smaller as shown in 3.53(d). This final effect on emissions agrees with previous combustion parameter results and indicate that highly fluctuating cycles will produce higher NO emissions while at the same time producing a higher number of smaller soot particles. It is important to note in this section that this effects were observed for relatively smaller particle diameters (up to 25 nm), while measurement campaigns for higher soot amounts and larger particles showed great measurement instability and inconclusive results. Thus the results presented here are only an indication of the trends yet to be fully experimentally observed, and should not be taken as definitive evidence at this point. A large amount of uncertainty still remains deeply intertwined within the measurement procedure and equipment used, such that the gathered data used is of questionable validity. Therefore more accurate and faster response measurement techniques are recommended for future use and further validated data is needed for any strong conclusions.
3. Results

Figure 3.53: Correlations between the geometrical soot particle sizes and relevant combustion parameters. The red line divides the relevant measurement points (size above 20 nm) from the small particles, which are assumed to be created by the lubrication oil [37].
3. Results

![Graphs](image)

**Figure 3.54**: Correlations between the soot particle number (mass) and relevant combustion parameters.
4 Conclusion

In this section the results obtained during this thesis will be discussed and the relevant conclusions presented, summarising the main findings and the effects they have on the further diesel combustion research.

In past research lower combustion cycle temperatures have been promoted as an efficient method for improving the NOx-SFC tradeoff, especially by using extreme Miller timing and also pre-cooling techniques, since lower temperatures should theoretically produce lower NOx emissions. Continuing on previous work by [13], which showed that the NOx-SFC tradeoff benefits obtained through the reduction of cycle temperature are in reality restricted, this work experimentally proves that under such colder cycle conditions the cycle to cycle emissions of diesel engines exhibit a strong variation. This is related to the analogous strong variation in combustion parameters in the engine, which create this highly fluctuating cycles that have an adverse effect on NO emissions for diesel engines. Therefore, an understanding of why and how they are produced is important for reducing the overall tailpipe emissions, as required by regulatory agencies. The highly fluctuating cycles are created under combustion with low EOC temperatures and show a number of effects [13]:

- **Higher NO formation rate** - due to the higher temperatures in the cylinder created by higher pressures (isentropic compressions) produced by the fluctuation intensities. Also contribution from NO formed in the premixed combustion phase.

- **Higher in-cylinder temperatures** - result of better oxygen mixing, which is due to the pressure fluctuations and further spray penetration, as well as lower radiation heat losses, which are due to the increased soot oxidation at fluctuating conditions.

These above mentioned effects directly influence the measurements presented in this text, conducted on a medium-speed single-cylinder direct injection diesel engine with a simulated two-stage turbocharging system. The main results show that by decreasing the EOC and adiabatic flame temperature, a reversal in the decreasing trend of NO is obtained due to the high cycle to cycle fluctuations produced.

The experimental observations are presented in the following section, with the main points of interest. Afterwards an interpretation of the results is offered, with a discussion on which methods of NO reduction are recommended, accompanied with the outlook on how this work can contribute to further research and understanding.
4. Conclusion

of cycle to cycle variations in combustion and developing of methods to limit the emissions of diesel engines.

4.1 Experimental Observations

The experiments discussed in this text have been performed with extreme Miller valve timing and significant ID conditions, using also a pre-cooled charge air and show the following:

- ID and premixed combustion indirectly influences the cycle to cycle emission fluctuations, so methods to decrease the ID will successfully decrease cycle to cycle NO variations (fuel characteristics, pilot injections, EGR...)

- Pilot injections contributes significantly to the reduction of cycle to cycle variations in NO emissions, since they reduce the ignition delay and therefore the fluctuation intensities, which create higher NO values (Figure 3.7).

- The absolute value of NO emissions correlates linearly with the smoother peak cylinder pressure, the fluctuation intensity of the combustion cycle, and the maximum characteristic mixing rate of the combustion cycle (Figure 3.9, ??, 3.11).

- The distribution of the cycles under fluctuations conditions, ordered by fluctuating intensity follows a Beta distribution concentrated on the left (Figure 3.12).

- The average (Within 1σ) and highly fluctuating cycles (Above 1σ) have a high spread in terms of NO emissions for cold intake conditions - compared to the low fluctuating cycles (Below 1σ) (Figure 3.14).

- The intake temperature has a significant influence on the NO emissions, decreasing the overall NO emissions up to a certain temperature, while with further temperature decrease the trend is reversed. That means that the overall NO values are increased due to the significant contribution of the high fluctuating cycles under such conditions (Figure 3.15).

- The effect of EGR on reducing the variations of NO is very prominent, as well as reducing the total NO amount. The EGR exhibits diminishing benefits for high rates (Figure 3.20, 3.21).

- Even though the varying load has negligible effect on the fluctuation intensity produced, it does have significant effect on the cycle to cycle variation of NO produced (Figure 3.26).

- The fluctuation intensities contribute to higher NO creation only in the diffusion combustion part (Figure 3.28, with a lower amount of NO mass per fuel mass
4. Conclusion

used for higher loads as the additional fuel injected has less available time and
temperature for NO creation).

- Varying of the SOI timing has limited effect on the NOx-SFC tradeoff (Figure 3.36).

- High fluctuating cycles have a significant negative effect on shifting the NOx-
  SFC tradeoff. This effect is strong for higher efficiency, earlier SOI conditions
  and diminished for lower efficiency, later SOI (Figure 3.37).

- At low cycle temperatures, an increased injection pressure gives minimal
  benefits to fuel consumption, while substantially increasing the NO emissions
  produced (Figure 3.40).

4.2 Interpretation of Results

The above observations and findings, lead to a number of insights into the effect of
cycle to cycle variations in NO emissions and along with presented literature allow
for the following conclusions:

- Pilot injections are an effective method of reduction of injection delay and
  therefore cycle to cycle emission variations.

- The intake charge air temperature directly influences the ID and therefore
  the fluctuation intensity, indirectly dictating the level of cycle to cycle NO
  emissions.

- Highly fluctuating cycles have a higher influence on the overall NO emissions
  trend at lower charge air intake temperatures.

- EGR can be successfully used to reduce the cycle to cycle emission variations,
  because even though increasing the ID, EGR reduces the reactivity of the
  mixture and the premixed combustion proportion

- Increasing load does increase the cycle to cycle NO emission variation up to a
  certain maximum value, while further increases in load will increase the diffusion
  combustion proportion and reverse the trend. Consequently, there will be less
  premixed combustion available to generate the fluctuation intensities, which
  can then affect the diffusion combustion part and generate highly fluctuating
  NO emissions.

4.3 Outlook

The experimental measurements and findings presented in this text explain an effort

to understand better cycle to cycle emission variations and their creation. A number
of combustion parameters have been analysed and correlated with the emission
variations in order to quantify their influence. In terms of NO experimental findings,
the experimental campaigns showed meaningful data and insights which can be further allayed in more depth and eventually also included in a simulation model to better predict the cycle resolved and total NO emissions of rap engines. Some suggestions for further measurement campaigns could be to use two cylinder pressure sensors instead of one in order to resolved better the in-cylinder pressure fluctuations. Additionally, the measurement methods in this work focused on accurately measuring the cycle to cycle variations in emissions and not on the accurate absolute value measurements (although absolute values are in a acceptable range), so if accurate absolute measurements are desired it is suggested to use a more suitable NO measurement device.

In terms of soot experimental measurements, the current measurement setup exhibited lacklustre performance, necessitating further design improvements and different measurement methods, in order to provide more accurate cycle resolved soot data. In this aspect, it is suggested to develop a measurement methods which has a high sampling rate capability, with minimal dilution ratio, in order to capture the soot emission in a satisfactory manner.
Bibliography


