Control of particulate matter and nitrogen oxide emissions in diesel engines

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CONTROL OF PARTICULATE MATTER
AND NITROGEN OXIDE EMISSIONS IN
DIESEL ENGINES

A dissertation submitted to the
ETH ZURICH

for the degree of
Doctor of Sciences

presented by
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Dr. Alois Amstutz, co-examiner

2012
Quand tu veux construire un bateau, ne commence pas par rassembler du bois, couper des planches et distribuer du travail, mais réveille au sein des hommes le désir de la mer grande et belle.

Antoine de Saint-Exupéry
Acknowledgments

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December 12, 2012

Frédéric Tschanz
## Contents

Abstract ix

Nomenclature xii

1 Introduction 1
   1.1 Background and Motivation 1
   1.2 State of the Art 2
      1.2.1 Conventional Control of Diesel Engines 2
      1.2.2 Feedback Control of the Emissions 5
   1.3 Objectives and Contributions 6
   1.4 Structure of the Thesis 8

2 Experimental Facility 9
   2.1 Engine Test Bench 9
      2.1.1 Test Bench Configuration 10
   2.2 Analysis of the Combustion 12

3 Modeling & System Analysis 13
   3.1 Modeling of the Emissions and Sensors 13
      3.1.1 Emission Modeling; An Overview 13
      3.1.2 Modeling of the Engine-out PM and NO\textsubscript{x} 14
      3.1.3 Identification and Validation 18
   3.2 Control-oriented Analysis of the System 20
      3.2.1 NO\textsubscript{x} Control 21
      3.2.2 PM Control 23

4 Emission Control Structure 26
   4.1 Observer for the Engine-out Emissions 26
   4.2 Controller Design 29
   4.3 OP Range of the Controller 35
      4.3.1 Nominal Operating Range 35
4.3.2 Robust Operating Range .................................. 36
4.4 Validation of the Emission Controller ......................... 38

5 Combustion Control .............................................. 47
  5.1 Modeling and Controller Design .............................. 48
    5.1.1 Combustion Controller Synthesis ....................... 49
    5.1.2 Integration into the Emission Controller ............... 52
  5.2 Combustion Controller Validation .......................... 52

6 Certification Cycle Experiments ................................ 57
  6.1 New European Driving Cycle ................................. 57
  6.2 Federal Test Procedure 72 .................................. 63

7 Conclusions and Outlook ....................................... 69
  7.1 Conclusions .................................................... 69
  7.2 Outlook ......................................................... 71

Appendix ............................................................. 72
A Mean Value Engine Model ...................................... 72
  A.1 Submodels .................................................... 72
B Air-Path Control ................................................ 76
  B.1 Reduced Engine Model ....................................... 76
  B.2 Control Structure ............................................ 78
C Combustion Characteristics Estimation ........................ 84
  C.1 Hardware and Interfaces ................................... 84
  C.2 Process Functionality ....................................... 86
  C.3 Signal Processing ............................................ 89
D Learning Feedforward Control .................................. 94

List of Tables ..................................................... 96
List of Figures ..................................................... 96
Bibliography ........................................................ 102
Abstract

The pollutant emissions of particulate matter and nitrogen oxides are an issue of automotive diesel engines, which have to be addressed by internal and external measures. The focus in this work is on internal measures, namely on the integration of the engine-out pollutant emissions into a feedback control loop. The integration of feedback control for the emissions is an upcoming approach towards an innovative operation of automotive diesel engine systems. In particular, feedback control of the emissions can provide relaxation on the issues of the demanding process of calibration, the drift implicated deviation of the engine-out emissions, and the optimized matching of engine and aftertreatment operation.

This work presents a comprehensive control structure for the engine-out emissions of NO\textsubscript{x} and particulate matter. The feedback signals are based on estimates provided by a model-based observer. The control structure is extended by a cascaded controller for the center of combustion and the indicated mean effective pressure.

The advantages of this control structure, compared to conventional engine control, are demonstrated with various experiments. In particular, the control structure significantly reduces drift-based influences on the emissions and provides the possibility to implement varying emission strategies with just a minimal calibration effort. Furthermore, it allows the automatic calibration of the EGR and of the swirl valve, and it eliminates deviations of the characteristics of combustion.
Zusammenfassung


Mit dieser Arbeit wird eine umfassende Regelungsstruktur für die Partikel- und Stickoxidemissionen vor Abgasnachbehandlung präsentiert. Die Regelgrössen der Emissionen sind Schätzungen, die aus einem modellbasierten Beobachter gewonnen werden. Zusätzlich ist die Regelungsstruktur mit einem kaskadierten Regelkreis für den Verbrennungsschwerpunkt und für den indizierten Mitteldruck erweitert.

In experimentellen Untersuchungen zeigt die neue Regelung ihre gegenüber einer konventionellen Regelung vorteilhaften Eigenschaften. Durch die Regelungsstruktur werden insbesondere driftbedingte Abweichungen der Emissionen signifikant vermindert und verschiedene Emissionsstrategien lassen sich mit geringem Aufwand umsetzen. Des Weiteren werden die Abgasrückführrate und die Einlasskanalabschaltung durch die Regelung automatisch kalibriert, und Abweichungen der geregeltten Verbrennungseigenschaften werden eliminiert.
Résumé


Cette thèse présente un système de régulation des particules et des oxydes d’azote. Les variables régées sont estimées par un observateur d’état basé sur un modèle des émissions polluantes. En outre, le système de régulation est élargi par une régulation du centre de combustion et de la pression moyenne indicative.

Les avantages du système de régulation présenté sont démontrés par des expériences et en comparaison avec le système de régulation conventionnel. Ils se présentent sous forme d’une diminution significative des déviations d’émissions imputables à la dérive du système. La régulation proposée permet en outre d’implémente facilement des stratégies d’émissions différentes. En plus, la calibration de la recirculation des gaz d’échappement et du positionnement du papillon de turbulence s’effectue automatiquement et les déviations des propriétés de la combustion régées sont éliminées.
# Nomenclature

## Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>AFR</td>
<td>Air-to-fuel ratio</td>
</tr>
<tr>
<td>ATDC</td>
<td>After top dead center (crank angle)</td>
</tr>
<tr>
<td>AWBT</td>
<td>Anti reset-windup and bumpless transfer</td>
</tr>
<tr>
<td>BAFU</td>
<td>Swiss Federal Office for the Environment</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom dead center</td>
</tr>
<tr>
<td>BG</td>
<td>Burned gas</td>
</tr>
<tr>
<td>BTDC</td>
<td>Before top dead center (crank angle)</td>
</tr>
<tr>
<td>CA</td>
<td>Crank angle</td>
</tr>
<tr>
<td>CCA</td>
<td>Crank angle with respect to a specific cylinder</td>
</tr>
<tr>
<td>CI</td>
<td>Compression ignition</td>
</tr>
<tr>
<td>CoC</td>
<td>Center of combustion</td>
</tr>
<tr>
<td>ECA</td>
<td>Crank angle with respect to the engine</td>
</tr>
<tr>
<td>ECU</td>
<td>Electronic control unit</td>
</tr>
<tr>
<td>EGR</td>
<td>Exhaust gas recirculation</td>
</tr>
<tr>
<td>ETK</td>
<td>Emulator probe (Emulator Tastkopf)</td>
</tr>
<tr>
<td>FTP</td>
<td>Federal Test Procedure</td>
</tr>
<tr>
<td>FVV</td>
<td>Forschungsvereinigung Verbrennungskraftmaschinen e.V.</td>
</tr>
<tr>
<td>HFM</td>
<td>Hot-film mass flow meter</td>
</tr>
<tr>
<td>HRR</td>
<td>Heat release rate</td>
</tr>
<tr>
<td>IMEP</td>
<td>Indicated mean effective pressure (gross)</td>
</tr>
<tr>
<td>ISR</td>
<td>Interrupt service routine</td>
</tr>
<tr>
<td>MIMO</td>
<td>Multiple input multiple output</td>
</tr>
<tr>
<td>NEDC</td>
<td>New European driving cycle</td>
</tr>
<tr>
<td>OCA</td>
<td>On-line combustion analysis device</td>
</tr>
<tr>
<td>Symbol</td>
<td>Description</td>
</tr>
<tr>
<td>--------</td>
<td>-------------</td>
</tr>
<tr>
<td>OP</td>
<td>Operating point</td>
</tr>
<tr>
<td>PM</td>
<td>Particulate matter</td>
</tr>
<tr>
<td>pm</td>
<td>Phase margin</td>
</tr>
<tr>
<td>RGA</td>
<td>Relative gain array</td>
</tr>
<tr>
<td>RS</td>
<td>Robust stability</td>
</tr>
<tr>
<td>SCR</td>
<td>Selective catalytic reduction</td>
</tr>
<tr>
<td>SISO</td>
<td>Single input single output</td>
</tr>
<tr>
<td>SoC</td>
<td>Start of combustion</td>
</tr>
<tr>
<td>SOI</td>
<td>Start of injection</td>
</tr>
<tr>
<td>TDC</td>
<td>Top dead center</td>
</tr>
<tr>
<td>VGT</td>
<td>Variable geometry turbocharger</td>
</tr>
</tbody>
</table>

**Notation**

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$c$</td>
<td>Parameter [-], concentration [-]</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Specific heat capacity at constant pressure [J/kgK]</td>
</tr>
<tr>
<td>$c_v$</td>
<td>Specific heat capacity at constant volume [J/kgK]</td>
</tr>
<tr>
<td>$H_l$</td>
<td>Lower heating value of fuel [J/kg]</td>
</tr>
<tr>
<td>$k$</td>
<td>Parameter [-]</td>
</tr>
<tr>
<td>$m$</td>
<td>Mass [kg]</td>
</tr>
<tr>
<td>$m_{pm}$</td>
<td>Concentration of PM in the exhaust [mg/m$^3$]</td>
</tr>
<tr>
<td>$n$</td>
<td>Polytropic coefficient [-]</td>
</tr>
<tr>
<td>$n_e$</td>
<td>Engine speed [rpm]</td>
</tr>
<tr>
<td>$Q$</td>
<td>Heat energy [J]</td>
</tr>
<tr>
<td>$q_{inj}$</td>
<td>Injected fuel amount per cylinder [mm$^3$]</td>
</tr>
<tr>
<td>$R$</td>
<td>Specific gas constant [J/kgK]</td>
</tr>
<tr>
<td>$p$</td>
<td>Pressure [Pa]</td>
</tr>
<tr>
<td>$T$</td>
<td>Torque [Nm]</td>
</tr>
<tr>
<td>$u$</td>
<td>Input vector of a linear system [-]</td>
</tr>
<tr>
<td>$u_i$</td>
<td>Level of actuator $i$ [-]</td>
</tr>
<tr>
<td>$v$</td>
<td>Input vector of a system [-]</td>
</tr>
<tr>
<td>$V$</td>
<td>Volume [m$^3$]</td>
</tr>
<tr>
<td>$w$</td>
<td>Vector of grid points in some map [-]</td>
</tr>
<tr>
<td>$x$</td>
<td>State vector of a linear system [-]</td>
</tr>
<tr>
<td>$x_{bg}$</td>
<td>BG ratio in the intake manifold [-]</td>
</tr>
<tr>
<td>Symbol</td>
<td>Definition</td>
</tr>
<tr>
<td>--------</td>
<td>------------</td>
</tr>
<tr>
<td>$x_{bg,i}$</td>
<td>BG ratio in receiver $i$ [-]</td>
</tr>
<tr>
<td>$y$</td>
<td>Output vector of a linear system [-]</td>
</tr>
<tr>
<td>$z$</td>
<td>State vector of a system [-]</td>
</tr>
<tr>
<td>$z_{s,i}$</td>
<td>Observed/modeled signal of sensor $i$ [-]</td>
</tr>
<tr>
<td>$\Delta t$</td>
<td>Transport delay [s]</td>
</tr>
<tr>
<td>$\delta v$</td>
<td>Input vector to emission models [-]</td>
</tr>
<tr>
<td>$\zeta_i$</td>
<td>Observed/modeled raw emission species $i$ [-]</td>
</tr>
<tr>
<td>$\eta$</td>
<td>Efficiency [-]</td>
</tr>
<tr>
<td>$\vartheta$</td>
<td>Temperature [K]</td>
</tr>
<tr>
<td>$\theta$</td>
<td>Sensitivities of the emission models [-]</td>
</tr>
<tr>
<td>$\kappa$</td>
<td>Ratio of specific heat capacities [-]</td>
</tr>
<tr>
<td>$\lambda$</td>
<td>Air-to-fuel ratio [-], volumetric efficiency [-]</td>
</tr>
<tr>
<td>$\dot{v}$</td>
<td>White noise [-]</td>
</tr>
<tr>
<td>$\xi$</td>
<td>State for unmodelled emission influences [-]</td>
</tr>
<tr>
<td>$\sigma_{st}$</td>
<td>Stoichiometric AFR [-]</td>
</tr>
<tr>
<td>$\tau$</td>
<td>Time constant [s]</td>
</tr>
<tr>
<td>$\Phi$</td>
<td>Vector of interpolation weights [-], Inertia [kg m$^2$]</td>
</tr>
<tr>
<td>$\phi$</td>
<td>Crank angle [-], element of weighting vector $\Phi$ [-]</td>
</tr>
<tr>
<td>$\omega$</td>
<td>Rotational speed [rad/s]</td>
</tr>
</tbody>
</table>

**Subscripts**

<table>
<thead>
<tr>
<th>Subscript</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>b</td>
<td>Burned</td>
</tr>
<tr>
<td>bg</td>
<td>Burned gas</td>
</tr>
<tr>
<td>c</td>
<td>Cylinder, compressor, compression stroke, controller</td>
</tr>
<tr>
<td>coc</td>
<td>Center of combustion</td>
</tr>
<tr>
<td>d</td>
<td>Displaced (volume)</td>
</tr>
<tr>
<td>e</td>
<td>Engine, expansion stroke</td>
</tr>
<tr>
<td>eg</td>
<td>Exhaust gas</td>
</tr>
<tr>
<td>egr</td>
<td>EGR valve / flow</td>
</tr>
<tr>
<td>em</td>
<td>Exhaust manifold</td>
</tr>
<tr>
<td>evap</td>
<td>Evaporated</td>
</tr>
<tr>
<td>ff</td>
<td>Feedforward</td>
</tr>
<tr>
<td>im</td>
<td>Intake manifold</td>
</tr>
</tbody>
</table>
### Table of Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ic</td>
<td>Intercooler</td>
</tr>
<tr>
<td>inj</td>
<td>Injection</td>
</tr>
<tr>
<td>k</td>
<td>With respect to cycle #k</td>
</tr>
<tr>
<td>k+1</td>
<td>wrt. cycle #k+1 based on data from cycle #k</td>
</tr>
<tr>
<td>lg</td>
<td>Lag element</td>
</tr>
<tr>
<td>ld</td>
<td>Lead element</td>
</tr>
<tr>
<td>m</td>
<td>Main combustion or injection</td>
</tr>
<tr>
<td>n</td>
<td>Normalized</td>
</tr>
<tr>
<td>nox</td>
<td>Nitrogen oxides</td>
</tr>
<tr>
<td>nyq</td>
<td>Nyquist (frequency)</td>
</tr>
<tr>
<td>o</td>
<td>Observer, Kalman filter</td>
</tr>
<tr>
<td>ox</td>
<td>Oxygen</td>
</tr>
<tr>
<td>pi</td>
<td>Pilot injection</td>
</tr>
<tr>
<td>pim</td>
<td>With respect to p_{im}</td>
</tr>
<tr>
<td>pm</td>
<td>Particulate matter</td>
</tr>
<tr>
<td>s</td>
<td>Sensor</td>
</tr>
<tr>
<td>sv</td>
<td>Swirl valve</td>
</tr>
<tr>
<td>u</td>
<td>Uncertain (parameter)</td>
</tr>
<tr>
<td>ub</td>
<td>Unburned</td>
</tr>
<tr>
<td>t</td>
<td>Turbine</td>
</tr>
<tr>
<td>tc</td>
<td>Turbocharger</td>
</tr>
<tr>
<td>trig</td>
<td>Trigger pulse signal</td>
</tr>
<tr>
<td>vgt</td>
<td>VGT (actuator)</td>
</tr>
<tr>
<td>xbg</td>
<td>With respect to x_{bg,im}</td>
</tr>
<tr>
<td>φ</td>
<td>With respect to CA φ</td>
</tr>
<tr>
<td>ϕ</td>
<td>Fuel</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

1.1 Background and Motivation

In recent years, automotive compression ignition (CI) or diesel engines have experienced a significant technological development resulting in increased requirements on calibration and control [1]. This development has been driven by the severe tightening of the legislative limits set on the pollutant emissions in all relevant markets [2]. To reduce the emissions, additional components have been implemented in the air and fuel paths. Examples are cooled exhaust gas recirculation (EGR), the variable geometry turbocharger (VGT), swirl valves, and the common rail injection system. For the research community, the introduction of these systems has led to a number of challenging problems in control and combustion research. In particular, the control of the EGR [3, 4], and the conjoint control of the EGR and VGT [5–11] has resulted in numerous publications. Furthermore, investigation on the combustion and emission formation, particularly in connection with the high degree of freedom of a modern injection system, have led the way to an optimized operation of the diesel engine [12–18]. This development has been continued with the control of characteristics of the combustion [19–24]. However, so far, in production-type engines, the management of these systems has been performed without direct feedback from two main variables of interest, namely the amounts of nitrogen oxide (NOx) and particulate matter (PM) emitted. In order to remain within the limits set by legislation despite this lack of information, the calibration complexity has increased significantly. This drawback is even accentuated due to drift effects resulting from external influences in the long, medium, and short
timescale. Examples of such influences are the production spread and the aging of components, variations of the ambient conditions and fuel quality, or cylinder-individual EGR variations and transient deviations of the intake gas conditions.

The drawbacks with respect to the missing feedback of the engine-out emissions can be summarized as follows:

- The feasible design range within the legislative emission limits is reduced to compensate for emission variations resulting from drift influences such as aging, production spread, or fuel quality variations.

- A substantial number of compensation functions need to be calibrated in order to limit the influences of measurable drift sources such as variations in the ambient conditions.

- Engine-out emission information for optimized aftertreatment operation is missing.

With aftertreatment devices such as selective catalytic reduction (SCR) catalysts and diesel particle filters, a significant reduction of the tail-pipe emissions has been achieved. Nevertheless, due to the limited conversion efficiency of a NO\textsubscript{x} reduction device [25], the need for appropriate urea dosing, and of the additional fuel required for filter regeneration [26], well-balanced and known engine-out emissions of PM and NO\textsubscript{x} are still important.

The integration of feedback control for the engine-out emissions is a promising approach to relax these issues because it provides the possibility to control the engine-out emissions to a reference level, which can be determined based on the actual state of the entire propulsion system and further requirements.

1.2 State of the Art

1.2.1 Conventional Control of Diesel Engines

Modern low- and medium-duty compression ignition or diesel engines typically are controlled with a very limited number of feedback loops. Most importantly, feedback control is used for the air-path, i.e. for the boost
Table 1.1: Diesel engine actuators and their main purposes

<table>
<thead>
<tr>
<th>Actuator</th>
<th>Purpose and function</th>
</tr>
</thead>
<tbody>
<tr>
<td>intake throttle</td>
<td>increase of the pressure difference from exhaust to intake manifold for higher EGR capability</td>
</tr>
<tr>
<td>swirl valves</td>
<td>increased turbulence (swirl) within the cylinder</td>
</tr>
<tr>
<td>low-pressure fuel pump</td>
<td>provision of the fuel from the reservoir to the high-pressure pump</td>
</tr>
<tr>
<td>high-pressure fuel pump</td>
<td>provision of the high-pressure fuel for the common rail</td>
</tr>
<tr>
<td>fuel valve</td>
<td>actuator for control of the fuel pressure in the common rail</td>
</tr>
<tr>
<td>injectors</td>
<td>provision of the fuel for combustion with multiple injections per cycle</td>
</tr>
<tr>
<td>turbine actuator</td>
<td>(VGT or waste gate) control of the flow through the turbine to the requirements</td>
</tr>
<tr>
<td>EGR valve</td>
<td>control of the EGR flow</td>
</tr>
<tr>
<td>glow plugs</td>
<td>preheating of the cylinder charge for better ignition</td>
</tr>
</tbody>
</table>

pressure and the EGR ratio, with the turbine actuator\textsuperscript{1} and the EGR valve as principal manipulated variables. Further feedback loops are present for fuel pressure control, and in connection with the amount of injection into the individual cylinders. The latter is used to balance the energy released in each cylinder and to balance the total amount of injection with respect to the measured variables fresh air mass flow and exhaust air-to-fuel ratio $\lambda$. The information from further sensors is essentially used for on-board diagnosis and for the correction of map-based setpoints and reference values of the control loops and of further actuators, respectively. Furthermore, the information from sensors is used for the determination of the state of operation. A list of actuators that are found on a modern diesel engine are listed in Table 1.1.

\textsuperscript{1}VGT or waste gate

The correction of actuator setpoints and reference values is needed in order
to adapt the operation of the engine to external influences such that various requirements on the system are met. These requirements in particular include the appropriate amounts of PM and NO$_x$ emitted, in order to keep below the legislative limits on these emissions with a low consumption of fuel. Table 1.2 shows the development of the legislative emission limits for passenger cars in Europe. In connection with lacking emission information and the strict emission limits, the calibration of such correction functions is important and delicate.

Figure 1.1 shows a sketch of a conventional engine control structure with an emphasis on the emissions of NO$_x$ and PM. Feedback from the conditions of the exhaust gas is missing, except for the air-to-fuel ratio. This illustrates the conflict of limiting a disturbance (drift) influenced variable of major interest without feedback about the current level of the variable. The most relevant control loop regarding the emissions in connection to drift is the air-path. It is based on signals from the hot-film mass flow meter (HFM), the boost pressure sensor, sometimes a back pressure sensor and the air-to-fuel ratio sensor $\lambda$.

In diesel engine systems which are equipped with an SCR catalyst, pre-catalyst NO$_x$ sensors might be available. However, to the author’s knowledge, the use of their signal is limited to the operation of the SCR catalyst and is not included in a feedback loop with a higher bandwidth within the control structure of the engine.

Control loops for the combustion that are based on cylinder-pressure information are being addressed in current research for automotive diesel engines.
Table 1.2: EU emission standards for diesel engine passenger cars. (Limited to the NO\textsubscript{x} and PM masses emitted)

<table>
<thead>
<tr>
<th>Stage</th>
<th>Date</th>
<th>NO\textsubscript{x} [g/km]</th>
<th>PM [g/km]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Euro 3</td>
<td>2000</td>
<td>0.5</td>
<td>0.05</td>
</tr>
<tr>
<td>Euro 4</td>
<td>2005</td>
<td>0.25</td>
<td>0.025</td>
</tr>
<tr>
<td>Euro 5</td>
<td>2009</td>
<td>0.18</td>
<td>0.005</td>
</tr>
<tr>
<td>Euro 6</td>
<td>2014</td>
<td>0.08</td>
<td>0.005</td>
</tr>
</tbody>
</table>

1.2.2 Feedback Control of the Emissions

First steps in the field of emission control were presented in [27] with a model predictive control structure for the air path of heavy duty diesel engines. The controlled variables of the control structure can be switched to include the NO\textsubscript{x} signal from exhaust gas measurements.

An early approach which fully focused on the engine-out emissions was presented in [28], where a control structure to jointly control the NO\textsubscript{x} and the oxygen content of the exhaust\textsuperscript{2} was proposed. In [28], the plant with the controlled variables NO\textsubscript{x} and exhaust AFR was decoupled, and two single-input single-output (SISO) controllers were used, with the SOI and the EGR valve as manipulated variables, respectively. The work in [28] was based on the findings of the project Emissionsgeregelter Dieselmotor [29] that had been funded by the Forschungsvereinigung Verbrennungskraftmaschinen e.V. (FVV) and by the Swiss Federal Office for the Environment (BAFU). That project was the precursor for the present work.

An explicit model predictive approach for controlling the air path variables boost pressure and compressor air mass flow for a turbocharged diesel engine was proposed in [30]. Additionally, emissions feedback was integrated by introducing an upper bound level on the NO\textsubscript{x} emissions.

In [31], a control structure with superimposed controllers for the PM and the NO\textsubscript{x} was presented. The PM and NO\textsubscript{x} controllers were used to adapt the setpoint values of the controllers for the rail pressure, the boost pressure, the EGR, the center of heat release, and the engine-out torque. The results of that promising approach showed a good control performance on

\textsuperscript{2}The oxygen content of the exhaust gas is measured with the exhaust air-to-fuel (AFR) ratio sensor.
1.3 Objectives and Contributions

This work is based on the findings of the project *Russgeregelter Dieselmotor* published in [36]. The main objective is the development and integration of an appropriate control structure for the engine-out pollutant emissions of PM and NO$_x$ for the engine control architecture. Direct feedback control of the emissions from diesel engines is a relatively new but very promising approach to address the problem of remaining below the legislative limits on the emissions. Furthermore, an appropriate feedback controller for the emissions may help to reduce the complexity of calibration of a modern diesel engine. And it provides the potential for an optimized management of the entire engine system including the aftertreatment devices. This is illustrated in Figure 1.2. With feedback control of the emissions, the ellipsoid, signifying drift influences on the emissions, can be reduced.

the NO$_x$, while the PM emissions still tended to exceed their reference trajectory.

In [32], a model predictive approach was used for controlling the air and fuel path of a turbocharged heavy-duty diesel engine. In that study, the controlled variables included the indicated mean effective pressure (IMEP), the center of heat release, the maximum heat release rate, the exhaust gas NO$_x$ emissions, and the PM as controlled variables. The engine-out emissions were measured by a Siemens VDO NGK Smart NO$_x$ sensor and an opacimeter, respectively. The model used for controller design contained a set of linear models for different operating points, which were obtained by system identification.

A model-based approach was used in [33] for a feedforward actuator-trajectory optimization for reduced emissions. The authors proposed an offline optimization of the actuator trajectories over given operating-point (OP) trajectories to minimize the cost functional which contained the amounts of NO$_x$ and PM emitted. The approach was based on a model of the engine which included empirical emission models published in [34].

A further study with model predictive control of the engine-out emissions was presented in [35]. The authors proposed the control of the exhaust temperature, the NO$_x$, and the PM with the SOI, the fuel pressure in the common rail, and the ratio between main and pilot injection as manipulated variables.
Chapter 1  Introduction

Figure 1.2: Feedback control of the PM and NO\textsubscript{x} emissions reduces the influences of drift on the engine-out emissions. Furthermore, it provides a convenient possibility to adapt the engine calibration such that the system with aftertreatment optimally meets the legislative limits.

Furthermore, the controller provides a convenient way of calibrating the engine with respect to the emissions within the possibilities of the chosen manipulated variables. This is illustrated by the three star-markers which can be shifted within the PM-NO\textsubscript{x} trade-off. With an emission controller, a superimposed engine management structure can be developed in order to automatically adapt the engine calibration to the current state of the aftertreatment devices, such that the legislative limits on the emissions are met and the system performance is optimized conjointly.

The main contribution of this work is the feedback control structure for the emissions of PM and NO\textsubscript{x}, which has been published in [37].

For the measurement of particulate matter, production-type sensors are not available as yet. However, production-type PM sensors are expected to become available in the near future [38]. In this work, an AVL Micro Soot Sensor was used for the PM measurements. Due to the layout and installation of the sensor, a significant delay and a filtering effect are present on the measured signal. The same effects are present on the measured signal of the NO\textsubscript{x} emissions, where the delay is substantially smaller, however. Particularly for the measured PM signal, the delay is a highly limiting factor for an efficient feedback control. To overcome the issues in connection with these dynamic characteristics of the emission sensors, a
model-based observer for the current levels of engine-out NO\textsubscript{x} and PM has been developed. This is a contribution to enable high-bandwidth control of the emissions. The observer is based on the control oriented models of the emissions, which are discussed in Section 3.1 and which are based on the PM model presented in [39].

A further contribution of this work is the extension of the emission control structure with a cascaded combustion controller, which is discussed in Chapter 5 and which is published in [40].

A list of scientific publications which are a result of the research performed during the elaboration of this work is included in the CV at the end of this document.

1.4 Structure of the Thesis

The structure of this work is as follows: In Chapter 2, the test bench is described, and a short introduction to the analysis of the combustion characteristics is given. More detailed descriptions of the estimation of the combustion characteristics and of the corresponding embedded device are provided in Appendix C.

Chapter 3 presents a short overview of emission modeling and describes the control oriented emission model, which will be used as a basis for the emission observer. The design of this observer and the synthesis and tuning of the emission control structure are described in Chapter 4. This chapter also includes a robustness analysis of the controlled system.

The extension to cascaded control of the emissions and of the combustion is described in Chapter 5. The validation and a demonstration of the performance of the control structure with and without cascaded combustion control on certification cycles are shown in Chapter 6. The thesis closes with the conclusions and an outlook in Chapter 7.

Appendix A contains an overview of the mean value engine model, which has been used for simulation and which served as a basis for the synthesis of the air-path controller described in Appendix B. Part C of the Appendix contains descriptions of the hardware and interfaces of the combustion characteristics estimation device, its process structure and functionality, and the signal processing.
Chapter 2

Experimental Facility

All experiments reported in this work are based on the engine test bench which has been built up at the institute and which is described in a nutshell in this chapter. Furthermore, an overview to the combustion characteristics estimation device is given in this chapter, and in more detail in Appendix C.

2.1 Engine Test Bench

![Figure 2.1: Sketch of the engine with the most important components.](image)

Experiments have been carried-out on a production-type 6-cylinder 3-liter diesel engine OM642 provided by Daimler AG, which is illustrated in Figure 2.1. The calibration of the engine is intended for use with an oxidation
2.1. Engine Test Bench

Table 2.1: Characteristics of the test bench engine

<table>
<thead>
<tr>
<th>Type</th>
<th>OM 642, Euro 5, 2987 cm³, 6 cylinders</th>
</tr>
</thead>
<tbody>
<tr>
<td>Power</td>
<td>173 kW @ 3800 rpm</td>
</tr>
<tr>
<td>Torque</td>
<td>540 Nm @ [1500 ... 2500] rpm (limited to 400 Nm)</td>
</tr>
<tr>
<td>Features</td>
<td>cooled EGR, VGT, swirl valves, throttle, electrical VGT and EGR valve actuation</td>
</tr>
<tr>
<td>Injection</td>
<td>max. 1600 bar fuel pressure, 8-holes piezo injectors, up to 5 injections per cycle</td>
</tr>
</tbody>
</table>

catalyst, a particulate trap, and a SCR catalyst. Neither of which are installed on the test bench. The back pressure increase resulting from these devices is simulated by throttling the exhaust flow. The engine is equipped with a variable geometry turbocharger (VGT), cooled high-pressure exhaust gas recirculation (EGR), an intake throttle to increase the pressure difference over the EGR channel, and swirl valves\(^1\). The injection system operates with a fuel pressure of up to 1600 bar and is capable of providing up to 5 injection events per cycle. The relevant characteristics of the engine are summarized in Table 2.1. Due to brake and gearbox-substitute limitations on power and torque, limiting values of 400 Nm and 115 kW had to be respected.

The electric actuation of the EGR valve and of the VGT provide critical advantages compared to a pneumatically actuated correspondent. Further information about the engine is found in [41].

2.1.1 Test Bench Configuration

For bypassing and data acquisition, the engine control unit (ECU) is accessed with an ETAS ES910 rapid prototyping module via the emulator probe (ETK). The ES910 also handles the communication via CAN bus to additional devices such as the WAGO- and the CSM modules which

\(^1\)The swirl valve is a butterfly valve that continuously closes one of the two intake channels of each cylinder. Due to the resulting asymmetric flow into the cylinder the swirl, i.e. the turbulence within the cylinder is increased.
are connected to the sensors. The configuration of the test bench architecture with the data acquisition software INCA and the rapid prototyping environment INTECRIO is shown in Figure 2.2. Data acquisition from the ECU, as well as the adaptation of variables in the ECU is effectuated with the INCA software, which is running on the test bench control PC. For accessing variables of the rapid prototyping process, the experiment integration package (EIP) for INCA is required. The INTECRIO software compiles the rapid prototyping process and is used to configure the communication interface to the ECU and for the CAN bus. Additionally to the sensors that are included on the production type system, the engine has been equipped with several additional sensors which are listed in Table 2.2.

Table 2.2: Additional sensors on the engine

<table>
<thead>
<tr>
<th>Signal</th>
<th>Sensor &amp; Location</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>Thermocouple K-type, after each cylinder</td>
</tr>
<tr>
<td>Pressure</td>
<td>Sensortechnics BTE6002A4, after turbine</td>
</tr>
<tr>
<td>Air-fuel ratio</td>
<td>Bosch LSU4.9, after turbine</td>
</tr>
<tr>
<td>PM</td>
<td>AVL micro soot sensor, after turbine</td>
</tr>
<tr>
<td>$\text{NO}_x$</td>
<td>Continental Uninox24V, after turbine</td>
</tr>
</tbody>
</table>
2.2 Analysis of the Combustion

For the online estimation of combustion characteristics, an embedded device has been developed which is based on a STM32F107VC with a 72MHz Cortex-M3-ARM processor. The process running on the embedded device is triggered by counter-based interrupts from a $0.2^\circ$ crank angle (CA) pulse. Three interrupt routines handle the tasks of (1) pegging of the high-pressure sensor signal and data exchange with the ES910, (2) sampling of the cylinder-pressure in the relevant part of the cycle, and (3) estimation of combustion characteristics based on the sampled data. For online estimation and control of the combustion on a cycle-to-cycle basis, the interrupt routines have to be executed within the limited time until the next interrupt occurs. In this connection, the computationally demanding estimation of the combustion characteristics is the time-critical process which thus limits the operational area of the device to an upper engine speed. The estimation of the combustion characteristics are based on the pressure data from one single cylinder. An extension to cylinder-individual estimation and control is possible with additional devices. Detailed descriptions of the device and of the estimation techniques are provided in Appendix C.
Chapter 3

Modeling & System Analysis

For the analysis of the system characteristics and for the controller synthesis, empirical models of the engine-out PM and NO$_x$ emissions have been developed and adapted, respectively. These models are integrated into a mean-value model for the air-path and energy conversion, which is based on the principles described in [42, 43]. The modeling of the engine-out emissions is described below, while a detailed description of the mean-value model is provided in Appendix A. The modeling and system analysis is based on the publications [39] and [37].

3.1 Modeling of the Emissions and Sensors

3.1.1 Emission Modeling; An Overview

The appropriate modeling of the engine-out NO$_x$ and PM emissions of Diesel engines has been an issue in many publications. And depending on the requirements, models with various complexity have been developed. While the physical principles of the NO$_x$ formation are suitable to obtain a good prediction quality with limited model complexity, the formation principles of PM are much more difficult to be reproduced by models. Generally emission models can be classified as phenomenological models and (semi-) empirical models. Models exist in various modeling depths from detailed chemical modeling for CFD analysis to phenomenological mean-value modeling based on the conditions in the cylinder at intake valve close (IVC). Due to the relatively high computational burden of phenomenological models their use in real-time applications is limited to models with highly reduced complexity such as the virtual PM sensor developed for the project [36], which is described in [44]. In experiments,
the virtual sensor, which is a further development of the modeling approach of [45], has shown good estimation capability of the PM engine-out emissions with few parameters to be calibrated. A good example of a phenomenological NO\textsubscript{x} model is found in [46]. Empirical or semi-empirical models include the black- and gray-box approaches. In these modeling approaches, the physical principles of the emission formation are not included at all or only in a very general manner. As a consequence, these types of models need a relatively large number of parameters in order to be able to predict the emissions with sufficient precision. Additionally, the identified parameters are generally only valid for the engine that was used for identification. Furthermore, extrapolation is problematic or even impossible.

The non-phenomenological part of the model structure of the gray- and black-box models can be polynomial on the one hand or it is determined by artificial neural networks on the other hand. Examples for non-polynomial or neural network models of the PM and NO\textsubscript{x} emissions can be found in [34,47]. A polynomial modeling approach for the emissions of a diesel engine is presented in [48,49], where in the latter, the modeling is done for several overlapping regions of operation and linear interpolation between the regions. Models of the logarithmically scaled PM emissions estimated by quadratic equations of the inputs are presented in [50,51]. [52] provided an analysis of the advantages of different empirical modeling structures.

### 3.1.2 Modeling of the Engine-out PM and NO\textsubscript{x}

The model for the NO\textsubscript{x} emissions has been adapted from the one described by [53]. The PM model is an extension of the model presented in previous work [39]. The models provide estimates for the emissions based on the OP-dependent stationary emission levels and on the deviation of the engine state from the corresponding stationary state, due to transient effects, for example. They are intended for control purposes, and in particular as a basis for an observer which is described later. With the simple model structure chosen, the models are easy implementable and have a rather low footprint on the processes running on the ECU. However, the prediction capabilities of the models are limited due to their structure, in particular for the PM emissions. More precisely, the quality of the estimates is expected to decrease for increased deviation of the engine state from its stationary value. This is due to the increasing influence of higher-order
effects, which are not included in the model structure. Nevertheless, their applicability is justified with the simple structure and identifiability together with the good results obtained when controlling the emissions with feedback from an observer which is based on these models (cp. Chapter 6). The structure is identical for the PM and the NO\(_x\) models. It is shown in

![Block diagram of the control oriented PM model structure.](image)

**Figure 3.1:** Block diagram of the control oriented PM model structure. The inputs to the model are obtained from ECU-data.

The basic assumption of the modeling approach is the quasi-static dependence of the engine-out emissions on the conditions inside the cylinder and on the injection parameters. Furthermore, the engine state in stationary conditions is defined by the engine speed \(n_e\) and load, which in this work is defined as the amount of injection \(q_{\text{inj}}\). Therefore, for a first approximation of the emissions, a stationary map can be used. To take into account the influence of deviations in the engine inputs (for example in transient mode) the deviation of the emissions in function of the deviation of the engine state needs to be integrated as illustrated by the deviation calculation block in Figure 3.1. The engine-out emission level \(\zeta_i\) is fed into the sensor dynamics block to obtain an estimate of the corresponding sensor signal. The modeling of the sensor dynamics is required for the observer, which will be described in Section 4.1. The model input \(\delta v\) is the vector of deviations in the different model inputs, i.e. in the relevant engine variables. Based on the stationary emission base level \(\bar{\zeta}_i\) and its relative deviation \(\delta \zeta_i\), the engine-out emission level \(\zeta_i\) can be calculated, where the index \(i\) corresponds to PM or NO\(_x\).

\[
\zeta_i = \bar{\zeta}_i (n_e, q_{\text{inj}}) \left[1 + \delta \zeta_i (n_e, q_{\text{inj}}, \delta v_i)\right].
\] (3.1)
The complexity and the predictive accuracy of the model are determined by the correlation for the emission deviation \( \delta \zeta_i \) in function of the deviation of the engine from stationary conditions at the current operating point (OP) \( \delta v_i \).

\[
\delta \zeta_i = f_i (n_e, q_{inj}, \delta v_i)
\]

(3.2)

In this work, a linear approach is used, which is justified with the intended use of the model in an observer. Accordingly, the relative deviation of the emissions \( \delta \zeta_i \) is obtained with the vector of OP-dependent sensitivities \( \theta_i \) of the emissions on deviations in the model input vector \( \delta v_i \). The resulting model for the engine-out emissions can be rewritten as follows:

\[
\zeta_i = \bar{\zeta}_i (n_e, q_{inj}) \cdot [1 + \theta_i (n_e, q_{inj}) \cdot \delta v_i (n_e, q_{inj})].
\]

(3.3)

The vector of the model inputs \( \delta v_i \) is composed of the deviation of each input channel \( \delta v_{k,i} \) with respect to the corresponding OP-dependent steady-state or reference value \( \bar{v}_{k,i} \) and a normalization value \( v_{k,n} \) as follows:

\[
\delta v_{k,i} = \frac{v_{k,i} - \bar{v}_{k,i} (n_e, q_{inj})}{v_{k,n}}.
\]

(3.4)

For the present work, the vector of the model inputs has been adapted from [54] and [39] to take into account only the most relevant inputs. This is justified later in Section 4.1. The input channels for both the NO\(_x\) and the PM model are the burned gas (BG) ratio or rate in the intake \( x_{bg,im} \), the boost pressure \( p_{im} \), the SOI \( u_{soi} \), the swirl valve position \( u_{sv} \), and the deviation of the amount of injection \( \Delta q_{inj} \), i.e. \( v_i = [x_{bg} p_{im} u_{soi} u_{sv} \Delta q_{inj}] \). The BG rate in the intake manifold \( x_{bg} \) is defined as the mass fraction of burned gas to the total mass of gas in the intake manifold (cp. Eq. (A.5) in the Appendix). While it cannot be measured directly, it can be estimated on the basis of the measured fresh air mass flow \( \dot{m}_{air} \), the EGR mass flow \( \dot{m}_{egr} \), and the air-to-fuel ratio \( \lambda \) in the exhaust manifold, which is measured by an universal exhaust gas oxygen sensor. A model-based estimate for the EGR mass flow is provided by the ECU.

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1 Throughout the document, the notification \( x_{bg} \) is often used as a variable for the BG-rate in the intake manifold. The subscript \( im \) is omitted for space reasons.

2 As stated above, the amount of injection is used to define the engine load. It corresponds to a desired amount of injection which is requested in the ECU and in general is equal to the effectively injected amount. Nevertheless, under certain circumstances, it can be adapted. This is the case when controlling the IMEP as will be described in Chapter 5.
Sensor Dynamics

The dynamic characteristics of the sensor signals can be modeled as first-order lag element with a time delay resulting from the transportation of the gas to the sensor [39, 54, 55]. The signal $z_{s,i}$ of emission sensor $i$, thus can be modeled with the following differential equation, where $i$ corresponds to the emission species NO$_x$ or PM:

$$\frac{d}{dt} z_{s,i} (t) = -\frac{1}{\tau_{s,i}} \cdot [z_{s,i} (t) - \zeta_i (t - \Delta t_{s,i})]. \quad (3.5)$$

The variable $\zeta_i$ refers to the corresponding engine-out emission species according to (3.3) which reaches the sensor after a delay of $\Delta t_{s,i}$. The time constant $\tau_{s,i}$ of the first-order lag takes into account the response characteristics of the sensor, and the mixing of the gas on its way to the sensor. Measurements with stepwise changes in the SOI and in the swirl valve have been used for the identification of the sensor dynamics of the NO$_x$ and PM sensors, respectively. Due to the fact that a stepwise change in one of the actuators quasi-statically influences the engine-out emissions, all delay and filtering effects in the exhaust pipe and in the sensor can be identified with such measurements. For the NO$_x$ sensor, a time constant of $\tau_{s,nox} = 1.83$s and a transport delay of $\Delta t_{s,nox} = 0.3$s have been identified. The values for the PM sensor are $\tau_{s,pm} = 0.34$s and $\Delta t_{s,pm} = 1.25$s. Note that the dynamic characteristics of the Micro Soot Sensor are significantly influenced by the dilution ratio. For the experiments, in general a
3.1. Modeling of the Emissions and Sensors

Figure 3.3: Modeled and measured NO\textsubscript{x} (left) and PM (right) emissions together with the coefficient of determination of each emission species model.

dilution ratio of two has been used as the delay and the time constant are smaller for low dilution ratios.

3.1.3 Identification and Validation

The maps for the steady-state emissions $\bar{\zeta}_i \left( n_e, q_{\text{inj}} \right)$ and for the steady-state model input levels $\bar{v}_{k,i} \left( n_e, q_{\text{inj}} \right)$ have been obtained with stationary measurements. For the identification of the OP-dependent sensitivities $\theta_i$, stationary measurements with deviated model inputs at each OP have been used. The identification has been carried-out for all sensitivity parameters at once with the least-squares algorithm.

Figure 3.2 shows the steady-state levels of NO\textsubscript{x} (left) and PM (right) emissions within the engine operating range. For confidentially reasons, the absolute emission levels had to be normalized.

In Figure 3.3, the fit of the model for the NO\textsubscript{x} (left) and PM (right) emissions is shown together with the coefficient of determination $R^2$ of each model. The data shown contains all stationary basis and variation measurements used for identifying the engine-out emission models.

A validation measurement of the PM and NO\textsubscript{x} model for transient conditions is shown in Figure 3.4. The modeled NO\textsubscript{x} sensor signal shown in the top left plot matches satisfactorily with the measured signal (grey), except around 800 and 1100 seconds where some mismatch is visible. The speed and load trajectories of the engine are shown in the bottom plots in
Figure 3.4. Figure 3.5 shows the measured (grey) and modeled (dashed) sensor signals in more detail. Additionally, in Figure 3.5, the modeled engine-out emissions $\zeta_i$ are shown.

**Figure 3.4:** Modeled and measured (gray) sensor signal $z_{s,i}$ of the NO$_x$ (left) and PM (right) emissions on a part of the NEDC. The OP trajectory is shown in the bottom plots.

**Figure 3.5:** Measured emissions (gray) together with the modeled sensor signal $z_{s,i}$ and the modeled engine-out emissions $\zeta_i$ of NO$_x$ (left) and PM (right) on a part of the NEDC.
3.2 Control-oriented Analysis of the System

The appropriate choice of manipulated variables is crucial in order to obtain the desired effect on the emissions to be controlled, without compromising other requirements. Figure 3.6 shows a simplified block diagram with the processes relevant for the engine-out emissions in a conventional diesel engine. It illustrates the fact that the instantaneous emissions within the cylinder are essentially defined by the local temperature development and by the local availability of oxygen. The local temperature is a function of the combustion subsystem, which includes the ignition delay and the premixed and diffusion combustion. It is influenced mainly by the start of injection (SOI), the pressure and temperature of the gas, the local concentration of evaporated fuel $c_{\text{evap}}$, and the local concentration of oxygen $c_{\text{ox}}$. The local concentration of oxygen is mainly influenced by the dilution of the cylinder mixture with burned gas\(^3\) and by the turbulence-influenced transport phenomena of the gas. Note that the oxygen sink due to combustion is not explicitly illustrated in the block diagram. With the fuel-mixture preparation block, the processes of the distribution of the fuel droplets within the cylinder and their evaporation are represented. More detailed information about these processes and relationships can be found in [56], for example.

For the NO\(_x\) emissions, the local peak temperature is most relevant. Secondarily, the local oxygen concentration plays a role in the actual reaction

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\(^3\)The stoichiometric burned-gas part of the recirculated exhaust serves as inert gas.
process with the nitrogen. For the PM emissions, the principles of formation and oxidation are complicated. Details to these issues can be found in [57] or in [58], for example. The local concentration of oxygen $c_{ox}$ and a sufficiently high temperature for oxidizing the forming and growing particulates are of major importance. This means that rapid mixing, i.e., a high turbulence is required even after the combustion is completed. While for low levels of NO$_x$, low oxygen concentrations and low temperatures are required, the conditions for obtaining low levels of PM are just the opposite, which illustrates the PM/NO$_x$ trade-off.

Figure 3.7 shows the sensitivities of the emissions on variations of the influencing parameters which are relevant for control. In the top plots, the sensitivity for SOI variations is shown. The influence of the NO$_x$ on SOI variations is strong, while the PM is influenced only to a low extent. The second row of plots shows a small sensitivity of the NO$_x$ on boost pressure variations, while such variations, have a high impact on the PM. The sensitivity of the emissions on BG-rate variations is shown in the third row of plots. Clearly, the BG-rate has a tremendous impact on both the NO$_x$ and, with reversed sign, on the PM. In the bottom plots, the influence of the swirl valve on the emissions is shown. While its impact on the NO$_x$ is small, it has a significant influence on the PM, particularly at low speed and load, where high levels of PM are emitted. For example a deviation of the normalized swirl valve position by $\delta v_{sv,n} = 1$, which corresponds to closing the swirl valve by 10%, results in a PM decrease of approx. 30% at an operating point of 1500 rpm and 25 mm$^3$ injection amount.

In the following, the influence of each input on the emissions and its appropriateness for control are analyzed and discussed.

### 3.2.1 NO$_x$ Control

As described above, the NO$_x$ level in a cylinder over a cycle is essentially defined by the local\(^4\) temperature trajectory and by the local availability of oxygen, which are influenced by the intake conditions, the BG rate, the SOI and the turbulence. [17] has shown that the influence of the turbulence on the PM oxidation outweighs its influence on the amounts of NO$_x$ emitted. Those results have been verified with experiments on the test bench. Turbulence generators thus are not suitable for NO$_x$ control when

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\(^4\)Local refers to a limited region of the cylinder volume. The total emission level of one cylinder is the integral amount over the cylinder volume.
Figure 3.7: Maps of the OP-dependent sensitivities $\theta_{i,j}$ for the NO$_x$ model (left) and PM model (right) for normalized variations of the SOI (top), boost pressure ($2^{nd}$ row), BG-ratio ($3^{rd}$ row), and swirl valve (bottom). The normalization value of each input channel is specified above the left graphs.
the PM is also considered. The intake pressure $p_{im}$ also is not suitable for NO$_x$ control since it is not controllable at low speed and load regions with the given engine configuration. Furthermore, the influence of the boost pressure on the PM outweighs its NO$_x$ impact.

The BG rate and the SOI thus are the remaining candidates for controlling the NO$_x$ emissions. Additionally to the results shown in Figure 3.7, Figure 3.8 shows measurements of the emissions for positive and negative variations of the SOI and the BG rate at various operating points (OP).

As expected, the measurements approve the assumption that the injection timing and the BG rate have a monotonous influence on the NO$_x$. While the SOI influence on the PM is small and non-monotonic, the strong impact of the BG rate on the PM is illustrated in the right plot in Figure 3.8.

Controlling the NO$_x$ with the SOI therefore appears to be a promising approach. However, the fact that the SOI also influences the thermal efficiency, as well as the exhaust gas temperature and the combustion noise, has to be taken into account. Therefore, in the case of drift, the engine performance can be deteriorated if the NO$_x$ level is controlled with the SOI alone.

### 3.2.2 PM Control

Based on the considerations described in Section 3.2, the first-sight candidates for use as manipulated variables for control of the PM emissions are the BG rate, the boost pressure and turbulence-generators such as the
swirl valve, and the fuel injection. The fuel injection based turbulence is influenced by the pressure in the common rail and by the characteristics of the injection, i.e. the number of injections and the amount of fuel that is injected during each event. Splitting up the injection into multiple events is of interest due to the fact that the injection-based turbulence dissipates rapidly and that the fuel distribution within the cylinder is changed. Furthermore, pilot injections influence the portion of fuel that is burned in the PM-less premixed phase.

Many of the potential manipulated variables, however, are not reasonably applicable for PM control. This is the case for the boost pressure due to its poor controllability in the range of low speed and load with high PM emissions. The BG rate would provide a very powerful manipulated variable. However it is in conflict with the NO\textsubscript{x} requirement. The rail pressure generally is already maximized, and further increase could lead to wall impingement, resulting in a massive increase of the PM emissions and in oil dilution. Therefore, control of the PM with the rail pressure is inadvisable.

The remaining candidates are the swirl valve and the characteristics of the injection, which are analyzed in more detail.

**Injection Characteristics for PM Control**

The injection parameters available for PM control are the number, the amount, and the timing of pilot injections as well as the timing and amount of an early post injection. [59] and [23] have shown that turning-off pilot injections can result in a decrease in PM emissions. Experiments on the engine used for this work have confirmed this behavior. However, whenever the amounts of the pilot injections are varied, an appropriate controller is required for the combustion in order to guarantee combustion stability under varying external conditions such as the fuel quality. The pilot injection characteristics thus are better suited for use in a combustion control loop rather than in an emission control loop.

An investigation of the potential of an early post injection for PM control yielded results similar to those reported by [59]. However, the sensitivities of the PM emissions on the parameters of the post injection where found to be poorly suited for PM control. This issue is discussed in more detail in [37] and [60].
Figure 3.9: Influence of the swirl valve on the NO\textsubscript{x} and PM at various operating points (left). PM emissions in function of the swirl valve position \( u_{sv} \) (right)

**Swirl Valve for PM Control**

Figure 3.9 shows in the left plot the impact of the swirl valve position on the PM and NO\textsubscript{x} emissions for various closing ratios from 10% closed to 90% closed. Clearly the swirl valve has a massive impact on the PM emissions, particularly for closing ratios above 50%. This influence is present in the whole operating range. Furthermore, the sensitivity of the PM emissions on the swirl valve is monotonic for all OP, which is a desirable feature for control. In the right plot in Figure 3.9, the PM emissions corresponding to a number of swirl valve positions are shown for some exemplary operating points.

As a result, the swirl valve with its desirable sensitivity characteristics and its high PM impact clearly is the best choice as a manipulated variable for PM control. However, for OP with low PM and high NO\textsubscript{x} emissions, an increasing influence on the NO\textsubscript{x} emissions is observed. This influence on both the NO\textsubscript{x} and PM emissions is an additional motivation for integrating the swirl valve into an emission control loop because simply closing the swirl valve would lead to increased NO\textsubscript{x} emissions. Accordingly, for a good performance in the PM/NO\textsubscript{x} trade-off, an appropriately controlled swirl valve position is required.
Chapter 4

Emission Control Structure

In this chapter, the design of the observer for the engine-out emissions and the synthesis and analysis of the control structure are described in Sections 4.1 and 4.2, respectively. These Sections are based on [37]. The feasible operating range of the control structure is determined with a robustness analysis in Section 4.3. An experimental validation of the control structure is shown in Section 4.4.

4.1 Observer for the Engine-out Emissions

The sensors for the NO\textsubscript{x} and PM emissions have a relatively slow response time with respect to the process to be controlled. In order to increase the feasible bandwidth of the emission control loops, a model-based observer for the engine-out emissions of NO\textsubscript{x} and PM has been developed. The observer is based on control oriented models of the engine-out emissions and of the sensor dynamics as described in Section 3.1. The basis-model structure for the NO\textsubscript{x} and PM emissions which is described by Equation (3.3), is extended by a variable \(\xi_i\), which contains all influences that are not reproduced by the basis-model. This variable \(\xi_i\) is integrated into the model structure in form of an integrator which is driven by white noise \(\dot{\nu}_i\). Based on Equation (3.3), the observer-model thus can be written as follows:

\[
\frac{d}{dt} \xi_i = \dot{\nu}_i \\
\zeta_i = \bar{\zeta}_i (n_{e}, q_{\text{inj}}) \cdot \left[1 + \theta_i (n_{e}, q_{\text{inj}}) \cdot \delta v_i (n_{e}, q_{\text{inj}})\right] + \xi_i.
\] (4.1)

With the extension of the basis-model by the white-noise driven integrator, the observer becomes capable of estimating the emissions with zero
steady-state offset. Furthermore, unmodeled influences such as temperature variations, are taken into account implicitly.

The structurally identical models for the PM and NO$_x$ sensors and emissions described in Equations (3.5) and (4.1) can be rewritten in state-space form. An additional first-order lag is integrated to approximate the discrete behavior of the engine cycles as a continuous process. This is required to avoid direct feedthrough through the observer. Furthermore, the transport delay is moved from the sensor input to its output.

The system for emission species $i$ thus can be reformulated as follows:

$$\frac{d}{dt} x_i(t) = A_i x_i(t) + B u_i(t)$$

$$y_i(t) = C x_i(t - \Delta t_{s,i}),$$

(4.2)

where $y_i = z_{s,i}$ is the output of the system and $x_i = [z_{s,i} \zeta_i \xi_i]$ represents the state with $\zeta_i$ as the state variable for the continuous approximation of the engine-out emission species $i$.

With the time constant for the engine cycles $\tau_{cyc}$, the system matrices are obtained as follows:

$$A_i = \begin{bmatrix} -\frac{1}{\tau_{s,i}} & \frac{1}{\tau_{cyc}} & 0 \\ 0 & -\frac{1}{\tau_{cyc}} & \frac{1}{\tau_{cyc}} \\ 0 & 0 & 0 \end{bmatrix}, \quad B = \begin{bmatrix} 0 \\ \frac{1}{\tau_{cyc}} \\ 0 \end{bmatrix},$$

(4.3)

$$C = \begin{bmatrix} 1 & 0 & 0 \end{bmatrix}.$$

The time constant of the engine cycle (two rotations) is determined such that within one engine cycle the first order lag reaches 95% of its steady-state value, which corresponds to a third of the time per cycle.

$$\tau_{cyc} = \frac{2}{n_e \frac{1}{3}}$$

(4.4)

Based on the engine-out emission model summarized in (3.3), the input vector $u_i$ of the state-space system (4.2) is obtained as follows:

$$u_i = \begin{bmatrix} \bar{\zeta}_i [1 + \theta_i \delta v_i] \\ \dot{\nu}_i \end{bmatrix}$$

(4.5)

The OP dependency of $\bar{\zeta}_i$ and $\theta_i$ has been omitted for better readability.

The observer for the engine-out PM and NO$_x$ emissions is composed of two
parallel Kalman filters which are based on the system described in Eq. (4.2) for the respective emission species. Figure 4.1 shows the structure of the Kalman filter for emission species $i$. The estimated sensor signal $\hat{z}_{s,i}$ has to be delayed by the respective transport delay to yield the appropriate residual. Due to the model design with the state variable for unmodeled influences $\xi_i$, the estimate $\hat{\zeta}_i$ converges to the measured value $z_{s,i}$. The engine-out emission estimate $\hat{\zeta}_i$ is used for control. Due to the fact that $\hat{\zeta}_i$ does not include the sensor dynamics, it allows for a higher bandwidth of the controlled system.

The Kalman filters have been designed at a reference speed of 1500 rpm. The rate of convergence of the filters is obtained with the assumed covariances of the process and measurement noises $Q_i$ and $R_i$ [61]. For the process noise covariances $Q_i$, the identity matrix is used, while for the measurement noise covariances $R_i$, values of $R_{pm} = 3 \cdot 10^4$ and $R_{nox} = 5 \cdot 10^3$ have been chosen for PM and NO\textsubscript{x}, respectively. The observer for the engine-out emissions in the following designates the two Kalman filters for the NO\textsubscript{x} and PM emissions.

To improve the performance of the observer, the state-variables for unmodeled influences $\xi_i$ within the NO\textsubscript{x} and PM Kalman filters have been implemented in a learning feedforward control structure as described in Appendix D.

Figure 4.2 shows the Nyquist plots of the NO\textsubscript{x} (left) and PM (right) channels of the observer. The NO\textsubscript{x} channel has a high phase margin due to the relatively short time delay of the corresponding sensor. The PM sen-
In this section, the new controller for the emissions of NO\textsubscript{x} and PM is presented. Compared to conventional diesel engine controllers, the advantages of this controller for the engine-out emissions are that it provides an innovative way of relaxing the following issues in automotive diesel engine management:
4.2. Controller Design

The requirement of keeping within the emission limits set by legislation with minimal compromise on other requirements such as low levels of consumption and system cost.

2. The higher complexity of calibration in order to limit drift influences on the emissions, resulting from controlling merely representative variables instead of the signals of interest, i.e. the emissions.

Both issues originate from the lack of emission feedback, so that additional measures need to be taken to ensure compliance with the legislative limits and to minimize undetected drift. Measures that compromise other requirements could be an engine calibration with widened margins from the emission design point to the legislative limits, and a suboptimal configuration of the aftertreatment devices with respect to the engine potential, for example. The higher complexity of calibration is a result of not controlling the emissions directly. To compensate for external, measurable influences on the emissions due to variations of the ambient conditions, for instance, a number of additional functions need to be calibrated.

Taking into account these considerations and the related in Chapters 1 and 3, the following key attributes of the control structure have been determined:

- The observer-based engine-out emissions are used as controlled variables. The manipulated variables are the SOI and the position of
the swirl valve $u_{sv}$ for the values NO$_x$ and PM, respectively. These choices are motivated by the system analysis presented in Section 3.2.

- Based on the arguments in Section 3.2.1, the BG-rate reference of the air-path control loop is used as an additional manipulated variable for the NO$_x$ control loop to avoid any long-term deviation of the SOI and to automatically calibrate the BG-rate reference map.

The PM control loop consists of a PI controller for the estimated engine-out PM emissions with the swirl valve position as manipulated variable. The NO$_x$ controller also contains a proportional (P) and an integrator (I) element. Their respective outputs, however, are not fed into the plant via one single manipulated variable but via the SOI and via the reference value for the BG rate $r_{bg}$ of the air-path controller.

The reason for including the BG rate into the NO$_x$ loop is due to the fact that one of the main purposes for the recirculation of exhaust gas is the reduction of the NO$_x$ emissions \[3\]. Therefore, the exact amount of recirculated exhaust gas is of importance mainly with respect to the resulting NO$_x$ emissions. It makes thus sense to adapt the reference value of the BG rate in case the NO$_x$ emissions deviate from their reference value. However, due to the fact that the BG rate can be estimated on a significantly higher bandwidth than the NO$_x$ emissions, direct EGR valve actuation based on the NO$_x$ without a cascaded EGR controller is not recommended.

As shown in Section 3.2.1, the SOI provides an important influence on the NO$_x$ emissions. By controlling the NO$_x$ with the SOI, it is therefore possible to compensate for deviations of the BG rate in transients, for example. In accordance with these considerations, the P element of the NO$_x$ controller is connected to the SOI in order to provide fast NO$_x$ control.

To obtain a similar absolute NO$_x$ impact over the operating range with the P-part of the controller, the NO$_x$ error variable $e_{nox}$ for the P-part of the NO$_x$ controller is normalized with the operating point dependent steady-state level $\bar{\zeta}_{nox}(n_e,q_{inj})$. This is justified because the variability of the sensitivity $\theta_{nox,soi}$ from SOI to NO$_x$ is relatively small over the operating range.

The I element is used to adapt the BG rate reference $r_{bg}$ of the air path controller such that the BG rate, regarding the resulting NO$_x$ level, matches with the NO$_x$ reference for the current OP and conditions. To avoid any excessive adaptation of the BG rate reference in transients when the NO$_x$
deviation temporarily might be high, the input to the I element of the controller is limited by a saturation. To improve the performance of the controller and to provide automatic calibration, learning feedforward control [62], which corresponds to automatic OP-dependent calibration of the BG rate reference, has been implemented for the I element of the NO\textsubscript{x} controller. With the learning feedforward control implemented, the integrator state is stored in function of the OP. The details of implementation of this feature are described in Appendix D.

The validity of the control structure for the combined control of the PM and NO\textsubscript{x} emissions is limited to the engine operating range where the sensitivities of the PM and NO\textsubscript{x} emissions on the manipulated variables allow for combined control with the chosen manipulated variables. In Section 4.3, the operating range of the control structure is analyzed and discussed for the nominal and for the uncertain plant.

To avoid performance deterioration in case of actuator saturation, all integrators are implemented with anti reset-windup. In particular, anti reset-windup is used to recover the BG rate reference in case of saturation of the PM controller. A block diagram of the entire control loop except for the anti-windup and saturations is shown in Figure 4.4.
The permissibility of controlling the NO\textsubscript{x} and PM emissions in two SISO loops has been analyzed with the relative gain array (RGA), which represents the cross-coupling between the channels. Regarding the NO\textsubscript{x} loop, this analysis has been conducted for the SOI manipulated variable only. This is justified with the intention to adapt the BG rate reference in a slow manner. Therefore, the coupling of the BG rate to the PM is an issue on frequencies significantly lower than the bandwidth of the PM control loop. This will be shown in Section 4.2.

Figure 4.5 shows the first diagonal element of the RGA that has been obtained with the 2x2 nominal plant with inputs SOI and swirl valve, and outputs NO\textsubscript{x} and PM engine-out emissions. The value of the first diagonal element is close to one in the OP range considered. Since the cross-coupling within the plant for the considered input channels is small, the controller design with two SISO loops is justified. Note that the RGA-elements are frequency independent because the considered dynamics, i.e. the continuous approximation of the engine cycles are equal on both channels.

**Controller Tuning**

The following considerations have been made for tuning the control structure:

1. Transient PM emission peaks influence the cumulative PM emissions to a high extent [63]. Therefore fast PM control is desirable.
2. The cumulative NO\textsubscript{x} emissions are less influenced by transients. A lower bandwidth is thus acceptable on this controlled variable.

3. The bandwidth of the BG adaptation loop must be well below the bandwidth of the PM control loop due to its strong cross-coupling to the PM emissions.

The magnitudes of the loop gain $|L_i(j\omega)|$, the sensitivity $|S_i(j\omega)|$, and the complementary sensitivity $|T_i(j\omega)|$ of the PM and NO\textsubscript{x} loops, tuned according to the above considerations, are shown in the left and right plots of Figure 4.6, respectively. The right plot additionally shows the loop gain $|L_{bg}|$ for the BG-rate adaptation channel of the NO\textsubscript{x} controller, i.e. the I element with deactivated P element.

Clearly, the PM and NO\textsubscript{x} loop crossover frequencies, which are indicated by diamond markers, are separated by more than two octaves. Accordingly, the PM loop is significantly faster, which is illustrated by the higher bandwidth on the complementary sensitivity that is indicated by triangle markers. The disturbance rejection bandwidth\textsuperscript{1}, indicated by circle markers, is similarly high for both loops, which is due to the identical first-order lag approximation of the engine process on both channels.

\textsuperscript{1}These frequencies only account for disturbances which are reproduced by models used in the observer.
When considering the NO$_x$ loop with the deactivated P element, i.e. control with the BG-rate adaptation only, the crossover frequency on $L_{bg}$ is further reduced. The crossover frequency on this channel is significantly lower than the bandwidth of the PM control loop and than the bandwidth of common EGR controllers [4]. Therefore, the two loops are also decoupled when taking into consideration the BG rate adaptation channel from the NO$_x$ controller.

### 4.3 OP Range of the Controller

#### 4.3.1 Nominal Operating Range

The control structure for the combined control of the PM and NO$_x$ emissions is intended for operating regions with a sufficiently high sensitivity of the PM on the swirl valve. Such conditions are found in the lower speed and lower load range of the given engine. Under conditions of a very low PM level and high NO$_x$ emissions which result from a very low BG-rate, for example, the appropriateness of PM feedback control is arguable. This is because the sensitivity of the PM on the swirl valve is low. In such regions, NO$_x$ control clearly is of highest priority while the PM can be expected to remain in a narrow band without feedback control.

Figure 4.7 indicates the nominal limit of operating range of the control structure for the given engine and calibration. It has been obtained with the sensitivities of the emissions on the manipulated variables. The levels in Figure 4.7 correspond to the normalized DC gain from the swirl valve on the PM emissions with the NO$_x$ control loop closed. They show the sensitivity of the PM on the swirl valve plus the sensitivity of the PM on the BG-rate adaptation which is required to compensate for the NO$_x$ influence of the swirl valve. As highlighted with a thick line in Figure 4.7, for high speeds and higher loads, the DC-gain changes sign. This sign change is thus a result of cross-couplings which are present in the plant on low frequencies due to slow BG-rate reference adaptation of the NO$_x$ control loop, in combination with a relatively low sensitivity of the PM on the swirl valve. In the region with the reversed sign, the control structure cannot be used.
4.3. OP Range of the Controller

Figure 4.7: Normalized DC gain of the plant from the swirl valve to the PM emissions with a closed NOx control loop. The sign change of the DC gain is highlighted with a thick line.

4.3.2 Robust Operating Range

As stated in Section 3.1 the emission models used in this work have a limited extrapolation capability. Thus for increasing deviation of the engine from its stationary state, the quality of the emission estimates is expected to decrease. Such effects, can be expressed as an uncertainty of the model parameters. To consider such effects in the determination of the feasible operating range of the control structure, a robustness analysis of the control loop with uncertain parameters has been carried out.

Figure 4.8 shows a block diagram of the uncertain control system which has been used for the robustness analysis. The uncertain plant $\tilde{P}(s)$ represents the engine and sensors and is composed of the uncertain emission model and the uncertain sensor and process time constants. The observer $K(s)$ is composed of the nominal plant $P(s)$ which is extended by the filter gain input. Its vector of estimated engine-out emissions $\hat{\zeta}$ is used as controlled variable. The estimated vector of sensor signals $\hat{z}$ together with the vector of emissions $z$, which are measured on the uncertain plant $\tilde{P}(s)$, yields the estimation error $\hat{e}$.

The robustness analysis has been performed within a grid of OP in the relevant operating range of the engine. The range of uncertainty around the nominal value of the relevant emission model parameters $\theta_{i,k}$ has been
determined as follows:

\[
\theta_{u,i,k}(n_e, q_{inj}) = \in [\tilde{\theta}_{\min,i,k}, \tilde{\theta}_{\max,i,k}] \tag{4.7}
\]

The limits of the uncertainty range are obtained according to the following equation. The bracket indicating the OP-dependency of the parameters \(\theta_{i,k}(n_e, q_{inj})\) is omitted for space reasons.

\[
\begin{align*}
\tilde{\theta}_{\min,i,k} &= \min \{\theta_{i,k} - \Delta \theta_{i,k}, 0.9 \cdot \theta_{i,k}, 1.1 \cdot \theta_{i,k}\} \\
\tilde{\theta}_{\max,i,k} &= \max \{\theta_{i,k} + \Delta \theta_{i,k}, 0.9 \cdot \theta_{i,k}, 1.1 \cdot \theta_{i,k}\} 
\end{align*} \tag{4.8}
\]

The absolute deviation \(\Delta \theta_{i,k}\) is determined with respect to the mean value of the corresponding parameter \(\theta_{i,k}\) over all grid points \(n, l\) in the operating range.

\[
\Delta \theta_{i,k} = 0.1 \frac{\sum_{n=1}^{N} \sum_{l=1}^{L} \theta_{i,k}[n,l]}{N \cdot L} \tag{4.9}
\]

Additionally, uncertainty has been added on the time constants of the plant \(\tau_{\text{nox}}, \tau_{\text{pm}}\), and \(\tau_{\text{cyc}}\). For these parameters, an uncertainty range of \(\pm 10\%\) around the respective nominal value has been chosen.

Figure 4.9 highlights the region of operation of the control structure where robust stability (RS) holds. The levels show the lower bound of the stability margin of the closed-loop system. In some regions with high load or high speed, robust stability does not hold. This is mainly due to the local values of the emission model parameters \(\theta_{i,k}\). In the regions where robust stability does not hold, relatively small variations in the emission model parameters may result in a sign-change of the DC-gain as described in Section 4.3.1.
4.4 Validation of the Emission Controller

Figures 4.10 and 4.11 show the performance of the controller for steps in the PM reference value and for a BG deviation, respectively. The advantage of the observer-based controlled variables is clearly visible in Figure 4.10. The estimated engine-out PM emissions $\hat{\zeta}_{pm}$ shown in the top right plot are set to the new reference value immediately, while the measured value reaches the new setpoint later due to the sensor dynamics.

Figure 4.11 shows a stepwise BG rate change (bottom left) at 2000 rpm, 170 Nm. A deviation of the BG rate could appear in transients or, on a longer timescale, due to aging, for example. In a conventionally controlled diesel engine these cases could lead to a significant deviation of the emissions, as illustrated by the modeled NO$_x$ emissions with deactivated emission control (top left). With the proposed control architecture, the emission deviation resulting from this BG rate deviation is minimized as shown in the measured NO$_x$ emissions (top left). Additionally, the SOI deviation for NO$_x$ control (bottom right) is eliminated by the automatic recovery of the BG rate reference (bottom left) to a value where the NO$_x$ emissions are met without SOI control action. In case of drift, a deviation of the BG rate might not be apparent on the estimated engine-out emissions $\hat{\zeta}_{nox}$. Thus, the compensation in such a case would be limited by the observer bandwidth, which however is significantly faster than the rate of change of most conceivable drift sources.
Figure 4.10: Measured emissions, reference trajectories, and observed engine-out emissions for stepwise changes in the PM reference (top plots). Trajectories of the SOI and the swirl valve position (bottom plots).

Figure 4.11: Measured emissions, reference trajectories, and emission trajectories modeled without NO\textsubscript{x} control on the SOI (top plots). Basis BG rate reference $r_{bg,\text{base}}$, BG rate $x_{bg}$, and BG reference adapted by the NO\textsubscript{x} controller (bottom left). SOI control action $\delta u_{soi}$ (bottom right).
Figure 4.12: Cumulative emissions with (gray) and without (black) emission control for various drift-afflicted experiments. The drift impact is illustrated by the minimum-surface ellipsoids which contain all points for active and deactivated emission control, respectively. The results for active emission control are highlighted in the detail plot.

Figure 4.12 shows the drift compensation capabilities of the emission control structure. Each marker corresponds to the normalized cumulative emissions of one measurement in a series of measurements that have been carried-out with either active (grey) or deactivated (black) emission controller. The results have been obtained on a part of the NEDC cycle. For each measurement of the series, the air-path controlled variables boost pressure $p_{im}$ and BG rate $x_{bg}$ have been changed by multiplication with a drift factor to obtain a drift-deviation of the engine. Clearly, the emission control structure significantly reduces the drift influence on the emissions. This is also illustrated by the ellipsoids with minimum surface, which contain all points of each measurement series.

Figures 4.13 to 4.16 show the performance of the control structure for load ramps with varying duration, and compare them to identical load ramps with conventional ECU control. The load ramps have been tested at an engine speed of 1300 rpm. Figure 4.13 shows a load ramp with 1.5 seconds duration. During the ramp, the swirl valve gets saturated and a slight PM
overshoot is visible in the estimated emissions as well as in the measured (delayed) PM trajectory in the top left plot. Also the estimated NO\(_x\) emissions, shown in the top right plot show a certain overshoot with respect to the reference value trajectory, which is limited by control action on the SOI (second-row right plot). The measured NO\(_x\) emissions include the filtering effect of the sensor. During the ramp, the air path variables boost pressure and BG rate, which are shown in the third-row plots, follow their respective reference trajectories quite well with only minor deviations in the BG rate. Note that the boost pressure is not feedback controlled at the low load OP, which explains the steady-state offset. The bottom plots show the engine speed and brake torque trajectories, respectively.

Figure 4.14 shows the same load ramp as shown in Figure 4.13 in comparison to conventional ECU control. The ECU performance is shown in gray. In comparison to conventional control, the emission controller significantly reduces the peak in PM emissions during the ramp, which is due to appropriate control of the swirl valve. This is also illustrated with the normalized, cumulative PM emissions in the lower left plot. In the top right plot it is visible that the measured NO\(_x\) emission trajectory for active emission control is slightly higher, compared to conventional ECU control. However this corresponds to the NO\(_x\) reference trajectory (cp. Fig 4.13) and can be assigned to the slightly decreased BG rate and the more closed swirl valve during and shortly after the ramp.
Figure 4.13: Load ramp with 1.5s duration. The graphs show the measured, estimated, and reference emissions (top), swirl valve and SOI (2nd row), boost pressure and BG rate (3rd row), and OP trajectory (bottom).
Figure 4.14: Comparison of active emission control with conventional control (ECU) for the load ramp as shown in Figure 4.13. The bottom left plot shows the normalized, cumulative PM emissions.
Figures 4.15 and 4.16 show a load ramp of five seconds duration. Again, the control performance of the emission controller on the PM and NO\textsubscript{x} during the ramp is good as shown in Figure 4.15. The overshoot in both the PM and the NO\textsubscript{x} emissions, visible in the top plots, again is due to swirl valve saturation and due to the proportional control action on the SOI (second row plots). The air-path controlled variables (third row) also are controlled well. The bottom right plot shows the torque trajectory, with the ramp clearly visible. The advantage of the emission control structure over the conventionally controlled engine is clearly illustrated in Figure 4.16. Although the duration of the ramp is fairly long and the dynamics of the ramp are thus very moderate, the ECU controller is unable to limit the PM peak because of the map-based swirl valve actuation, which is shown in the top left and bottom left plots. For such a slow ramp, it would not be a problem to drastically cut the PM emissions as shown with the results obtained by the emission control structure. However, because the ECU controller does not have emission feedback, actuation of the swirl valve is not adapted to the current intake conditions or emissions. As a consequence of the slightly lower boost pressure and the slightly higher BG rate in case of ECU control (third row plots) in combination with the OP dependent swirl valve actuation, a massive PM peak appears. The NO\textsubscript{x} emissions are slightly higher during the ramp for the emission controlled trajectory, in comparison to the ECU controlled trajectory. Nevertheless, the advantage of emission feedback is made evident with the unnecessarily high PM peak in the conventionally controlled case. As shown in the bottom left plot in Figure 4.16, the emission controller is able to cut the cumulative PM emissions by approximately fifty percent with only slightly increased NO\textsubscript{x} emissions. The brake torque shown in the bottom right plot is similar for both cases.
Figure 4.15: Load ramp with 5s duration. The graphs show the measured, estimated, and reference emissions (top), swirl valve and SOI (2nd row), boost pressure and BG rate (3rd row), and OP trajectory (bottom).
Figure 4.16: Comparison of active emission control with conventional control (ECU) for the load ramp as shown in Figure 4.15. The bottom left plot shows the normalized, cumulative PM emissions.
Chapter 5

Combustion Control

In this chapter, a cascaded controller for the combustion characteristics\textsuperscript{1}, with the emission controller from Chapter 4 in the outer loop, is presented. The modeling of the plant and the synthesis of the controller are described in Section 5.1. The combustion controller is validated with experiments in Section 5.2. The chapter is based on the publication [40].

Introduction and Objectives

Besides the properties of the gas within the cylinder, the characteristics of the combustion have a major influence on the thermodynamic efficiency, the pollutant emissions, and the noise of a diesel engine. The fuel injection system in nowadays production type diesel engines is actuated basically in a feedforward manner, i.e. without feedback about the characteristics of the combustion. As a consequence, external influences on the combustion remain uncontrolled. Examples for external influences are varying ambient conditions, deviations of the trapped gas composition and properties due to transients and cylinder-individual EGR variation, and varying fuel properties. Furthermore, with a controller for the emissions as described in Chapter 4, slight deviations of the IMEP, have to be expected due to SOI actuation.

To limit the above mentioned detrimental effects on the engine performance, the emission control structure from Chapter 4 is extended with a cascaded controller for the indicated mean effective pressure (IMEP) and for the center of combustion (CoC).

Feedback control of combustion characteristics has been an issue in vari-

\textsuperscript{1}The term combustion characteristics denotes features of the combustion such as center of combustion, ignition delay, and generally cylinder-pressure related variables.
ous publications, e.g. [22, 24, 64, 65]. The goal in this work is to develop an appropriate multivariable controller for the inherently discrete plant which incorporates significant measurement noise, and to integrate it into the emission control structure.

In contrast to common CoC-control, in this work the center of combustion of the fuel injected during the main injection, $\phi_{\text{coc,m}}$, is controlled. This choice of controlled variable has the advantage that the possibility to optimize the pilot injection parameters independently of the CoC-controller remains unimpaired. Hence, an additional control loop for characteristics of the pilot injection such as proposed in [23] could be implemented without significant reconsideration of the combustion control structure.

5.1 Modeling and Controller Design

In context of nowadays possibilities of fuel-injection actuation\(^2\), the characteristics of the combustion are an inherently discrete process. It is furthermore assumed, that the influence of the characteristics of combustion in some cycle $k$ on the subsequent cycle $k + 1$ is negligible. Because the characteristics of combustion $z_k$ of some cycle $k$ cannot be estimated before the end of the combustion, they are available for control for the next cycle $k + 1$ at the earliest. The characteristics of combustion thus can be written as follows, where $\mathbf{v}$ represents the configuration of the input.

$$z_{k+1} = f(\mathbf{v}_k), \quad (5.1)$$

With the sensitivity of the characteristics of combustion $z$ with respect to the input $\mathbf{v}$, the function $f(\mathbf{v})$ is replaced with the following affine approximation.

$$f(\mathbf{v}) \approx B(\mathbf{v} - \mathbf{v}_0) + f(\mathbf{v}_0) \quad (5.2)$$

Based on some stationary operating point (OP) and the corresponding vector of the characteristics of combustion $z_0$ and input $\mathbf{v}_0$, the linear plant is obtained with the deviation of the characteristics of combustion

\(^2\)The determination of fuel-injection parameters such as the SOI and the duration of injection is due at some instant in the cycle and after that instant it is assumed that no degree of freedom is left to influence the development of the combustion process.
\[ x = z - z_0 \] and the deviation in the vector of inputs \( u = v - v_0 \) as follows.

\[
\begin{align*}
x_{k+1} &= 0x_k + Bu_k \\
y_k &= Cx_k
\end{align*}
\] (5.3)

In accordance with the controlled and manipulated variables, the state-, input-, and output vectors are determined as \( x = [\Delta \phi_{\text{coc,m}} \Delta \text{IMEP}] \), \( u = [\Delta u_{\text{soi}} \Delta q_{\text{inj,m}}] \), and \( y = [\Delta \phi_{\text{coc,m}} \Delta \text{IMEP}] \), respectively. Other influencing variables are considered as disturbances. The matrix \( C = I \) is equal to identity.

Figure 5.1 shows the performance of the plant model for variations in the input vector compared to the measured values. Clearly the model fits the estimated (measured) trajectories well.

### 5.1.1 Combustion Controller Synthesis

For the controller synthesis, all frequencies are scaled with respect to the varying\(^3\) sample time of the inherently discrete process to be controlled. This is justified with the fact that the controller is synchronized to the variable sample time of the process. The following considerations have been made with respect to the desired properties of the combustion controller:

- A reasonably high bandwidth and zero steady-state offset are required for reference-tracking.

\(^3\)The sample time of the process is the duration of an engine cycle, which varies with the engine speed.
To avoid amplification of measurement noise, peaking of the sensitivity at higher frequencies has to be limited.

A low order of the controller is desirable.

The controller is synthesized with the $H_\infty$ method. To obtain the desired properties, the plant is extended with the following frequency weighting function for both elements of the error variable $e$, i.e. for the sensitivity $S$.

The frequency weighting function is composed of a lead- and a lag element in series and is defined in continuous time.

$$W_e(s)^{-1} = 10^{-3} \frac{T_{ld} \cdot s + 1}{T_{ld} \alpha_{ld} \cdot s + 1} \frac{T_{lg} \cdot s + 1}{T_{lg} \alpha_{lg} \cdot s + 1}$$ (5.4)

The parameters of the lead and lag elements are provided in Table 5.1.

<table>
<thead>
<tr>
<th>Table 5.1: Parameters of the lead and lag element</th>
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<tbody>
<tr>
<td>Lead element</td>
</tr>
<tr>
<td>$\alpha_{ld}^{-1}$</td>
</tr>
<tr>
<td>$T_{ld}^{-1}$</td>
</tr>
<tr>
<td>Lag element</td>
</tr>
<tr>
<td>$\alpha_{lg}$</td>
</tr>
<tr>
<td>$T_{lg}^{-1}$</td>
</tr>
<tr>
<td>Nyquist frequency</td>
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<td>$f_{nyq}$</td>
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</table>

The lead element defines the bandwidth of the closed loop and ensures elimination of steady-state offset. The lag element enforces the inevitable\(^4\) peak in $S$ to be reduced for higher frequencies, in order to avoid high-frequency noise amplification. Weighting of the controlled variable $y$ and of the manipulated variable $u$ is omitted to obtain a low order of the controller and due to the facts that the resulting closed loop shows no peaking in the complementary sensitivity and that no further explicit limitations on the manipulated variable are required.

For controller synthesis, the weighting function $W_e$ is discretized. Controller synthesis is carried-out by solving the discrete-time $H_\infty$ problem. The resulting singular value trajectories of the loop gain $L$, the complementary sensitivity $T$ and the sensitivity $S$ are shown in Figure 5.2, together

\(^4\)see [66]
with the inverse of the weighting function. Note that the minimum and maximum magnitudes of the respective singular value trajectories shown are very close to each other.

To obtain an immediate response of the plant in the case of variations in the reference, a feedforward path has been implemented additionally. The feedforward path is obtained as follows with the regular matrix $\mathbf{B}$ from Eq. (5.3).

$$u_{\text{ff}} = \mathbf{B}^{-1}\delta r \tag{5.5}$$

Figure 5.3 shows the structure of the combustion controller. In accordance with the implemented feedforward path and the characteristic of the discrete plant, the error variable $e_k$ is calculated with the delayed reference $r_{k-1}$.

The reference $r_k$ is composed of the reference deviation $\delta r_k$, and of the OP-dependent base reference $\bar{r}_k (n_e, q_{\text{inj}})$. A reference deviation $\delta r_k$ at cycle $k$ has a direct effect on the manipulated variable $\delta u_k$ via the feedforward path $\delta u_{\text{ff},k}$. The plant input $\mathbf{u}_k$ is composed of the manipulated variable $\delta u_k$ and of the OP-dependent base actuator settings $\bar{u}_k (n_e, q_{\text{inj}})$. Thus, for perfect model match and appropriately set base values $\bar{r}_k$ and $\bar{u}_k$, the current cycle output $y_{k+1|k}$ which is obtained in the subsequent cycle $k+1$, corresponds to the current reference $r_k$ and no further control action is required.
5.1.2 Integration into the Emission Controller

For the integration of the cascaded combustion controller into the emission control structure described in Chapter 4, the manipulated variable \( \delta u_{soi} \) of the emission controller is switched to the CoC reference value deviation \( \delta r_{coc,m} \). The IMEP reference variable is obtained from interpolation on an OP-dependent reference map and is not influenced by the emission controller.

Figure 5.4 shows the block diagram of the cascaded control loops. The level of detail of the outer loop emission controller has been simplified for space reasons. The processes running on the OCA are triggered by the CA pulse signal \( u_{trig} \) and yield estimates for the CoC and the IMEP, based on the sampled in-cylinder pressure \( p_c \). The air path controller with controlled variables BG rate \( x_{bg} \) and boost pressure \( p_{im} \) operates in parallel and is not a part of the proposed control structure. The block engine, sensors & observers incorporates the combustion and emission formation principles as well as the dynamics of the air path, sensors, and the observers for the PM and NO\(_x\) emissions.

5.2 Combustion Controller Validation

The performance of the proposed combustion controller is demonstrated with CoC and IMEP steps at 1500 rpm and 100 Nm, as shown in Figures 5.5 and 5.6, respectively. The controlled variables \( \phi_{coc,m} \) and IMEP are shown in the top left and right plots, respectively, together with their reference trajectories. The response of the control loop on the reference
Chapter 5  Combustion Control

Figure 5.4: Block diagram of the cascaded control loop including the engine with sensors and observers, the emission control structure, the OCA, and the cascaded combustion controller.

Figure 5.5: The top plots show the $\phi_{\text{coc},\text{m}}$ (left) and IMEP (right) and their corresponding reference values for steps in the $\phi_{\text{coc},\text{m}}$ reference at 1500 rpm, 100 Nm. The bottom plots show the SOI and the injected fuel amount, respectively.
5.2. Combustion Controller Validation

Figure 5.6: The top plots show the $\phi_{coc,m}$ (left) and IMEP (right) and their corresponding reference values for steps in the IMEP reference at 1500 rpm, 100 Nm. The bottom plots show the SOI and the injected fuel amount, respectively.

steps is without delay, which is due to the implemented feedforward path. Clearly, the proposed combustion controller is able to compensate for the coupled influences, as can be seen with the combined actuation on both manipulated variables, combined with the good reference tracking in both controlled variables. Furthermore, when comparing the CoC and SOI trajectories it can be seen that the measurement noise is not amplified, which is a requirement of the combustion controller.

The influence of variations of the cylinder-charge composition at an OP of 1300 rpm, 60 Nm is shown in Figure 5.7. At approximately four seconds, the EGR rate is varied, resulting in the AFR trajectory $\lambda$ shown in the top right plot. The combustion controller has to deviate the SOI by approx. 3°CA in order to compensate for the influence on the CoC (left plots). From the nearly constant amount of injection (bottom right), it can be concluded that the IMEP is hardly influenced by the EGR variation.

Figure 5.8 shows the influence of a step in the NO$_x$-reference at 1400 rpm. At the instant of the NO$_x$-reference step (bottom left plot), the outer loop emission controller reacts with a $\phi_{coc,m}$ reference step for the combustion controller.
In case of a deactivated combustion controller, the $\phi_{\text{coc,m}}$ reference step is transformed into a SOI step with identical amplitude. In the top left plot, it is clearly visible that the reference step, due to the feedforward path, has an immediate effect on the plant, i.e. the estimated CoC at the next update already shows the step. In case of a deactivated combustion controller, the SOI control action is also coupled to the IMEP, which shows a stepwise deviation from its original value in the top right plot. With the active combustion controller, this IMEP deviation is compensated by the controller by adapting the injected amount of fuel, as shown in the middle left plot. The IMEP deviation for a deactivated combustion controller is also clearly visible in the (smoothened) brake torque, which is shown in the middle right plot. Due to the coupling of the IMEP to the SOI, the brake torque drops as a result of the control action of the emissions controller on the SOI. In contrast, with the active combustion controller, this reduction in brake torque is compensated. Note that the trajectories of the measured NO$_x$ emissions in the bottom left plot hardly show a reaction due to the slow sensor dynamics. The PM emissions are shown in the bottom right plot.
5.2. Combustion Controller Validation

Figure 5.8: Influence of a NO\textsubscript{x} reference step at 1400 rpm with active and deactivateated combustion controller.
Chapter 6

Certification Cycle Experiments

In this chapter, the performance of the emission control structure is demonstrated with experiments on certification cycles. The experiments have been carried out either with the emission control structure alone, or with the additional cascaded combustion control structure.

For the experiments, the engine governor is actuated such that the brake torque follows the reference load trajectory of the cycle. The engine speed is controlled synchronously by the brake in order to follow the corresponding speed trajectory. For the NEDC, the reference trajectories are based on measurement data of an NEDC test conducted with a vehicle equipped with an engine of the same type as the one that has been used for this work. The FTP-72 cycle reference trajectories have been obtained from a simulation with a vehicle model.

The starting point for the emission reference maps in this work is based on the steady-state emission data at each node of the OP map and have been obtained with the conventionally calibrated and controlled engine. On the vehicle, aftertreatment devices are included that were not available in this engine test rig. Therefore, the results shown cannot be brought into direct context with legislative limits that the entire vehicle must comply with.

6.1 New European Driving Cycle

The NEDC experiments have been conducted without the cascaded combustion control structure. Figure 6.1 shows a section of the urban part of the cycle. Plots one and two show the measured emissions of NO$_x$ and PM with their respective reference trajectories, and the observer output $\hat{\zeta}_i$. In plots three and four, the trajectories of the manipulated variables SOI
and swirl valve are shown together with the corresponding OP-dependent basis-value trajectories. The bottom pair of plots shows the trajectories of the engine speed and load, respectively.

Clearly, the observer-based PM emission trajectory $\hat{\zeta}_{pm}$ follows the reference trajectory, except when the swirl valve is in saturation. Also for the NO$_x$ value, the observed emissions $\hat{\zeta}_{nox}$ are set to the reference value by the SOI and with the appropriately auto-calibrated BG rate reference map. For the NO$_x$ emissions it has to be kept in mind that the actuation of the SOI is based on a P controller. Accordingly, some offset to the reference has to be expected in engine transients. The measured emissions (gray lines) satisfactorily correspond to the reference values when considering the slow dynamics of the NO$_x$ sensor and the significant time delay of the PM sensor. The SOI trajectory in stationary phases is identical to the OP-dependent basis-value trajectory $u_{soi,op}$. The swirl valve, however, is actuated such that the PM emissions meet the reference, and is not required to stay close to the OP-dependent basis-value.

The overland part of the NEDC is shown in Figure 6.2. Again the measured emissions with their respective reference trajectories, and the observer output are shown in plots one and two. Plots three and four show the trajectories of the manipulated variables SOI and swirl valve are shown together with the OP-dependent basis-value, and the bottom pair of plots shows the trajectories of the engine speed and load, respectively.

For this higher load part of the cycle as well, the emission control performance is good.
Figure 6.1: Section of the urban part of the NEDC. The observed engine-out emissions, and the delayed, filtered, measured emissions follow their respective reference trajectories. The trajectories of the manipulated variables SOI and swirl valve as well as the OP are shown in the lowest four plots.
Figure 6.2: Overland part of the NEDC. The measured emissions are set to the reference value except in case of actuator saturation.
Figure 6.3: The proposed controller compared to the results obtained with standard ECU control. In the two bottom plots, the normalized cumulative emissions are shown.

The entire potential and performance of the proposed control structure are shown in Figures 6.3 and 6.4, where the results of experiments on the entire NEDC are shown.

Figure 6.3 shows the emission mass flows with emission control in comparison to the conventionally controlled engine. To obtain the results shown in Figure 6.3, the reference values of the NO\textsubscript{x} and PM simply have been changed by multiplication with a factor, i.e. the whole reference map has been scaled. For the PM emissions, this scaling factor corresponded to a PM reduction of 62\%, while for the NO\textsubscript{x} reference an increase of 33\% had to be allowed for feasibility. In the lower two plots, the normalized cu-
6.1. New European Driving Cycle

Cumulative emissions are shown. The normalization was obtained by scaling with the final value of the cumulative emissions that has been obtained with the conventionally controlled engine. Clearly, the cumulative PM level has been reduced significantly by the emission controller with this simple adaptation of the reference emission levels.

It has to be noted that after changing the emission references, the critical parts of the cycle had to be driven a few times to allow auto-calibration of the BG reference to take place.

Figure 6.4 shows the cumulative emissions over the entire NEDC for various emission references. For these experiments as well, the calibration work consisted solely of some simple adaptation of the reference maps for the PM and NO\(_x\) emissions. Clearly, for all measurements the deviation between the cumulated measured emissions and the corresponding cumulated reference values are small, which means that the control performance is good also over a wide range of emission references. Additionally, Figure 6.4 shows that with effortless calibration work, analogously to the procedure described above, quite distinct emission strategies are feasible with the proposed control architecture.
6.2 Federal Test Procedure 72

The experiments on the FTP-72 have been conducted with the emission control structure with both activated and deactivated cascaded combustion control. Figures 6.5 and 6.6 illustrate the benefits and performance of the cascaded structure with combined control of the emissions and of the combustion. Figure 6.5 shows the performance of the cascaded controller on the first 120 seconds of the FTP-72 cycle. In the first and second row plots, the trajectories of the NO$_x$ and PM are shown. The dashed lines correspond to the PM and NO$_x$ reference, respectively. The controlled variables, i.e. the estimated engine-out emissions $\hat{\zeta}_i$, are drawn in black, while the measured emissions which contain the delays and filtering effects of the sensors, are drawn in gray. The emissions of PM and NO$_x$ follow their respective reference value well, except at approx. 45s, where a PM overshoot is visible. This overshoot is due to a saturation of the swirl valve, which can be concluded from the overshooting trajectory of estimated PM emissions. Plots three and four show the trajectories of the CoC and IMEP, respectively, together with their reference trajectories. Clearly, the combustion controller shows a good performance on this transient OP trajectory. The engine speed and load trajectories are shown in the bottom plots. Note that at low load, the combustion controller is deactivated as can be seen in the fifth-row plot with the normalized manipulated variables of the combustion controller. The reason to deactivate the controller in such cases is due to the potentially low quality estimates of IMEP and CoC at low load. As a consequence, in such cases, the controlled variables of the combustion controller deviate from their respective reference.

A comparison of conventional control with the cascaded controller on an extract of the FTP-72 cycle is provided in Figure 6.6. The first two plots show the emissions of NO$_x$ and PM together with the reference trajectories. In the third row plot, the BG rate trajectories are shown. They are not directly comparable. However, the peak in the BG rate that appears in the conventionally controlled engine leads to a significant deviation of the CoC towards late, as can be seen in the fourth row plot. The late combustion results in a diminished IMEP (fifth row) and brake torque (bottom right). With the cascaded combustion controller, such deviations can be avoided and the controlled variables remain close to their reference trajectories.
Figure 6.5: Part of the FTP-72 cycle with the cascaded emissions and combustion controller. The plots show the estimated, measured and reference NOx and PM emissions, the CoC, the IMEP, the normalized manipulated variables $\delta u_{i,n}$ of the combustion controller, and the speed and load trajectories (from top to bottom).
Figure 6.6: Comparison of the cascaded controller with a conventional controller on an extract of the FTP-72 cycle. The plots show the NO$_x$ and PM emissions, the BG rate $x_{bg}$, the CoC, the IMEP, and the engine speed and load (from top to bottom).
Figure 6.7 shows the first 320 seconds of the FTP-72 cycle, which include the highest load parts of the cycle. The results shown have been obtained with deactivated combustion control. Evidently, the emissions follow their respective reference trajectory, except at approx. 220 seconds, where a PM overshoot is visible. This overshoot results from a poor PM estimation of the emissions observer, as can be concluded from the not fully closed swirl valve. Note that the swirl valve mostly is more open than the OP-dependent base-map value $u_{sv,op}$. This fact illustrates the advantage of the emission controller compared to conventional engine control. With the emission controlled engine, the desired emission levels are met, while the calibration complexity is reduced. For map-based actuation of the swirl valve, the engine would emit less PM, and therefore the NO$_x$ emissions would be higher due to the more closed swirl valve. Furthermore, without the proposed emission control structure, the swirl valve position map needs to be calibrated precisely.

Figure 6.8 shows a comparison of the cascaded emission and combustion control structure with the emission controller without combustion control on the high-load part of the FTP-72. The two first plots show the measured emissions of NO$_x$ and PM together with the reference trajectory. Clearly, in both cases the control performance on these two controlled variables is good, except around 220 seconds, where some overshoot is present in the PM emissions due to under-estimation of the engine-out PM emissions. Furthermore, the cascaded controller shows PM overshoot at some load peaks, which is a result of the increased amount of injection required to meet the IMEP in combination with saturation of the swirl valve actuator. The CoC and the IMEP are shown in plots three and four respectively. Also here the control performance is good in general. Plot five shows the activation state of the combustion controller for cascaded control. As stated above, at low load the combustion controller is deactivated due to the possibly poor estimation quality of the combustion characteristics. This results in some offset of the combustion characteristics at the corresponding time intervals.
Figure 6.7: Section of the FTP-72. The plots show the emissions of NO\textsubscript{x} and PM, emission controller manipulated variables SOI and \( u_{sv} \), and the engine speed and brake torque (from top to bottom).
Figure 6.8: Comparison of the emission controller with and without cascaded control of the combustion. The plots show the NO\textsubscript{x} and PM emissions, the CoC, the IMEP, the combustion controller activation (only relevant for cascaded control), and the engine speed and load (from top to bottom).
Chapter 7

Conclusions and Outlook

7.1 Conclusions

In this work it has been shown that the integration of feedback from the engine-out emissions of PM and NO\textsubscript{x} into an adequate control structure addresses problems that are present in automotive diesel engine applications. In particular, the emission control structure provides the following improvements:

- The feasible design range within the legislative emission limits can be extended with the emission control structure. This is due to the fact that emission variations resulting from drift and transients are reduced with the emission controller.

This has been demonstrated with the compliance of the engine-out PM and NO\textsubscript{x} emissions with their respective reference values shown in Chapter 6 and Section 4.4, and in particular with Figure 6.4. With an extended design range on the emissions, a potential for improvement on other requirements such as the fuel consumption is provided.

- The complexity of calibration is reduced because with the emission control structure, the OP-dependent calibration of the EGR reference and swirl valve position is provided automatically. Furthermore, the calibration of compensation functions to account for emission influences resulting from ambient condition variations, for instance, can be omitted.

The benefit on both issues is a result of the advantage of directly controlling the variables of interest, i.e. the pollutant emissions, instead of representative variables effecting the emissions themselves. Therefore, drift
influences such as production spread, aging and environmental condition variations as well as transient influences on the emissions can be reduced. With a reduced spread of engine-out emissions and the potential of adapting the emission setpoints during operation, a potential for improving the coordination between the engine and the configuration and operation of aftertreatment devices is offered. This potential of improvement is clearly illustrated with the distinct engine-out emission strategies shown in Figure 6.4. The emission control structure thus provides a potential for optimizing the cost of the aftertreatment which is related to the costs of the devices and of their operation due to consumption of urea for SCR on the one hand and of fuel for filter regeneration on the other hand.

It has been shown in this work that for control of the emissions with the sensors available, observers for the engine-out emissions are inevitable for a sufficiently high bandwidth of the control loop. With the observer presented, which is based on rather simple control-oriented models of the pollutant emissions of NO\textsubscript{x} and PM, good results have been obtained in various experiments.

The extension of the emission controller with the cascaded combustion controller provides feedback control at the source of emission generation, that is at the combustion process. The following issues are addressed by the integration of the cascaded combustion controller into the emission control structure:

- Influences of air-path deviations and of fuel quality and ambient condition variations onto the IMEP and \( \phi_{\text{coc,m}} \) are eliminated.

- Deviations of the IMEP as a consequence of manipulating the parameters of the injection, e.g. by the emission controller, are reduced with the proposed combustion controller.

- With the designated controlled variables, the cascaded combustion controller serves as a basis for an extension to additional controlled variables.

The last point is due to the fact that the CoC of the main injection is controlled, instead of the CoC of the combustion in general, i.e. of all combustion events in global. Thus when changing parameters of the pilot injections their effect on the controlled (main) CoC is due to an actual deviation of the main combustion itself, instead of a mixed influence due
to the combustion of all injection events. Furthermore, the structure of the combustion controller is well adapted to the inherently discrete process of combustion.

As illustrated by the results, the cascaded control architecture for the emissions of PM and NO\textsubscript{x}, and for the combustion characteristics is a promising approach for controlling both the engine-out emissions and the characteristics of the combustion at a high bandwidth. This combined approach thus combines the aforementioned advantages of both controllers.

### 7.2 Outlook

Further steps in the field of engine-out emission control for diesel engines are the optimized determination of reference values for the respective emission species. The engine-out emission reference values for the NO\textsubscript{x} and PM emissions are important regarding the optimized operation of the entire propulsion system in a diesel engine automotive application. They have to be determined under consideration of requirements on the entire system such as the configuration and current efficiency of aftertreatment devices, and depending on the legislative emission limits that have to be respected for the application considered. A first basic example for the optimized determination of engine-out emission reference values have been proposed in [28] with an algorithm to adapt parameters of a calibration in order to meet emission limits while optimizing the fuel consumption. A further step towards cost-optimal determination of the engine-out emissions under consideration of aftertreatment efficiency has been shown in [67].

Furthermore, the extension of the combustion control structure could provide the potential of a further optimization of the engine performance regarding consumption and emissions. The integration of injection parameters such as the pilot injection into a feedback control loop could help to address issues in connection with the combustion stability, the consumption, and the pollutant and noise emissions. The potential of such efforts with respect to the emissions has been illustrated in [23]. Additionally, the integration of further injection parameters into appropriate control loops would further reduce the complexity of calibration on the many degrees of freedom of a modern injection system for diesel engines.
Appendix A

Mean Value Engine Model

In the following, the components of the mean-value air-path and energy conversion model are described. The mean-value model is composed of the following submodels: Intake and exhaust manifolds, compressor and VGT, turbocharger rotor inertia, mass flows through the EGR valve and engine, and energy conversion efficiency. The structures of the submodels generally are adapted or adopted from [43].

A.1 Submodels

Manifolds

The manifolds are modeled as isothermal receivers for air and BG. With the fuel flow into the cylinders \( \dot{m}_\varphi \) and the BG ratio as defined in (A.5), the mass balances for the receivers are obtained as shown in equations (A.1) - (A.4) [43].

\[
\frac{d}{dt} m_{\text{air,im}} = \dot{m}_c + (1 - x_{\text{bg,em}}) \dot{m}_{\text{egr}} - (1 - x_{\text{bg,im}}) \dot{m}_e \tag{A.1}
\]

\[
\frac{d}{dt} m_{\text{bg,im}} = x_{\text{bg,em}} \dot{m}_{\text{egr}} - x_{\text{bg,im}} \dot{m}_e \tag{A.2}
\]

\[
\frac{d}{dt} m_{\text{air,em}} = (1 - x_{\text{bg,im}}) \dot{m}_e - \dot{m}_\varphi \sigma_{st} - (1 - x_{\text{bg,em}}) (\dot{m}_{\text{egr}} + \dot{m}_t) \tag{A.3}
\]

\[
\frac{d}{dt} m_{\text{bg,em}} = x_{\text{bg,im}} \dot{m}_e + \dot{m}_\varphi (\sigma_{st} + 1) - x_{\text{bg,em}} (\dot{m}_{\text{egr}} + \dot{m}_t) \tag{A.4}
\]

The BG ratio in the receivers is defined as follows:

\[
x_i = \frac{m_{\text{bg,i}}}{m_{\text{bg,i}} + m_{\text{air,i}}} \tag{A.5}
\]
The exhaust manifold BG rate is related to the measured air-to-fuel ratio \( \lambda \) as follows with the stoichiometric air-to-fuel ratio \( \sigma_{st} \):

\[
x_{bg,em} = \frac{1 + \sigma_{st}}{1 + \lambda \cdot \sigma_{st}}
\]  

(A.6)

The pressure in the receivers is obtained with the perfect-gas assumption and the volume of the respective receivers \( V_i \):

\[
p_i = \frac{m_i R \vartheta_i}{V_i}
\]

(A.7)

The balance for the internal energy within the manifolds is defined as follows:

\[
\frac{d}{dt} U = \dot{H}_{in} - \dot{H}_{out} + \dot{Q}_{in}
\]

where \( \dot{H} \) are the enthalpy flows, \( \dot{Q}_{in} \) is the heat flux which is assumed to be zero. Equation A.8 can be reformulated to obtain the differential equations for the temperatures of the gas within the manifolds:

\[
\begin{align*}
\frac{d}{dt} \vartheta_{im} &= \frac{1}{m_{im} c_v} \left( (\dot{m}_c \vartheta_{ic} + \dot{m}_{egr} \vartheta_{egr}) c_p - \dot{m}_e c_p \vartheta_{im} - \frac{d}{dt} m_{im} c_v \vartheta_{im} \right) \\
\frac{d}{dt} \vartheta_{em} &= \frac{1}{m_{em} c_v} \left( (\dot{m}_e + \dot{m}_\varphi) c_p \vartheta_{eg} - (\dot{m}_c \vartheta_{em} + \dot{m}_{egr} \vartheta_{em}) c_p - \frac{d}{dt} m_{em} c_v \vartheta_{em} \right)
\end{align*}
\]

(A.9) (A.10)

The specific heat capacities for constant volume and pressure are assumed to be fixed and indifferent for burned gas and air.

**Turbocharger**

For the compressor mass flow, the model from [68] is adopted. The mass flow is obtained empirically from the intake manifold pressure \( p_{im} \) and the compressor speed \( \omega_{tc} \):

\[
\dot{m}_c = f(p_{im}, \omega_{tc})
\]

(A.11)

The compressor torque is obtained as:

\[
T_c = \dot{m}_c c_p \vartheta_{us} \left[ \Pi_c^{\frac{k-1}{k}} - 1 \right] \frac{1}{\eta_c \omega_{tc}}
\]

(A.12)
where $\Pi_c$ is the pressure ratio over the compressor and $\vartheta_{us}$ is the upstream temperature, $\kappa$ and $c_p$ represent the ratio of specific heat capacities and the specific heat capacity at constant pressure, respectively.

The VGT mass flow and torque models are adapted from [43]. The parameters $c_t$ and $k_t$ are third-order polynomials in $u_{vgt}$

$$\dot{m}_t = \frac{p_{em}}{\sqrt{\vartheta_{em}}} c_t \sqrt{1 - \Pi_t^{k_t}} \tag{A.13}$$

where $\Pi_t$ is the pressure ratio over the VGT. With the turbine efficiency $\eta_t (p_{em}, \omega_{tc}, u_{vgt})$, the turbine torque is obtained as follows:

$$T_t = \dot{m}_t c_p \vartheta_{em} \left[ 1 - \Pi_t^{1-\kappa} \right] \frac{\eta_t}{\omega_{tc}} \tag{A.14}$$

With the turbocharger inertia $\Theta_{tc}$, the differential equation for $\omega_{tc}$ is given as follows:

$$\frac{d}{dt} \omega_{tc} = \frac{1}{\Theta_{tc}} (T_t - T_c) \tag{A.15}$$

### Compressible Flow through Orifices

The mass flows through the orifices, EGR valve and throttle, are modeled with the Bernoulli equation for an isothermal orifice with the flow area $A_{eff}$, which is a function of the position $u_i$ of the corresponding valve actuator.

$$\dot{m}_i = A_{eff} \frac{p_{em}}{\sqrt{R \vartheta_{em}}} \Psi \left( \frac{p_{im}}{p_{em}} \right) \tag{A.16}$$

With critical pressure ratio $\Pi_{cr}$, the flow function $\Psi (\Pi)$ is defined as follows:

$$\Psi (\Pi) = \begin{cases} \Pi^{1/\kappa} \sqrt{\frac{2}{\kappa-1}} \left( 1 - \Pi^{\frac{\kappa-1}{\kappa}} \right) & 1 > \Pi \geq \Pi_{cr} \\ \sqrt{\kappa} \left( \frac{2}{\kappa+1} \right)^{\frac{\kappa+1}{\kappa-1}} & \Pi < \Pi_{cr} \end{cases} \tag{A.17}$$

### Engine Mass Flow

The engine mass flow $\dot{m}_e$ is obtained with the volumetric efficiency coefficients $\lambda_{\omega}$ and $\lambda_{lp}$ [43]:

$$\dot{m}_e = \lambda_{\omega} \lambda_{lp} \frac{\omega_e V_d}{4\pi} \frac{p_{im}}{R \vartheta_{im}} \tag{A.18}$$
The volumetric efficiency coefficients $\lambda_l(\omega_e)$ and $\lambda_{lp}$ take into account the influences from the engine speed $\omega_e$ and from the residual gas $\lambda_{lp}$:

$$
\lambda_{lp} = \frac{\epsilon - \left( \frac{p_{em}}{p_{im}} \right)^{\frac{1}{\kappa}}}{\epsilon - 1} \tag{A.19}
$$

**Temperatures**

The temperatures of the intake gas after the intercooler $\vartheta_{ic}$ and of the EGR after the EGR-cooler $\vartheta_{egr}$ are interpolated from OP dependent maps.

The temperature of the exhaust gas leaving the cylinder $\vartheta_{ev}$ is modeled as follows:

$$
\vartheta_{ev} = \frac{\dot{m}_e c_p \vartheta_{im} + k_{\vartheta} (\dot{m}_{\varphi} H_l - T_e \omega_e)}{\dot{m}_e + \dot{m}_{\varphi}} \tag{A.20}
$$

The part of fuel energy leaving the engine via the exhaust $k_{\vartheta}$ is approximated in function of the amount of injection $m_{\varphi,cyl}$, the engine speed $\omega_e$, and a constant factor $k_{eg}$ as follows:

$$
k_{\vartheta} = k_{eg} + k_{\omega} (\omega_e - \omega_{e,ref}) + k_1 (m_{\varphi,cyl} - m_{\varphi,cyl,ref}) \tag{A.21}
$$
Appendix B

Air-Path Control

The air-path controller described here is based on the publication [69]. The use of an own air-path controller is motivated with the repeatability of experiments in context with the highly nested controller and reference-value structures on the ECU. Furthermore, control of the air-path is a challenging issue in control of Diesel engines [9, 11, 70, 71], in particular in connection with the emissions. This is due to the fact that deviations of the intake gas conditions are a major source of emissions, in particular with respect to the PM emissions [63]. Thus a good controller for the air-path is a requirement for good performance on the emissions.

B.1 Reduced Engine Model

For controller design, a reduced model has been deduced from the mean-value model described in Appendix A. It covers only the most important dynamics of the process in order to obtain a low order of the controller. The reduced model is limited to the submodels: Intake and exhaust manifolds, compressor and VGT, turbocharger rotor inertia, and mass flows through the EGR valve and engine.

The model’s state $\mathbf{z}$ is composed of five variables, namely the mass of air and of burned gas (BG) in the intake manifold and in the exhaust manifold, and the turbocharger kinetic energy represented by its speed $\omega_{tc}$.

$$\mathbf{z} = [m_{\text{air,im}} \ m_{\text{bg,im}} \ m_{\text{air,em}} \ m_{\text{bg,em}} \ \omega_{tc}]^T$$  \hspace{1cm} (B.1)

The vector $\mathbf{v}$ of the model inputs is composed of the positions of the EGR valve $u_{egr}$ and VGT actuator $u_{vgt}$.

$$\mathbf{v} = [u_{egr} \ u_{vgt}]^T$$  \hspace{1cm} (B.2)
The fuel mass flow $\dot{m}_\varphi$ and the engine speed $\omega_e$ are not included in the vector of model inputs $\mathbf{v}$. The influence of variations in those two variables is thus considered as a disturbance on the plant.

The model output vector $\mathbf{w}$ contains the intake manifold BG ratio $x_{bg}$ and pressure $p_{im}$.

$$\mathbf{w} = [x_{bg} \ p_{im}]^T \quad (B.3)$$

The resulting nonlinear model can be written as:

$$\begin{align*}
\dot{\mathbf{z}} &= \mathbf{f}(\mathbf{z}, \mathbf{v}) \\
\mathbf{w} &= \mathbf{g}(\mathbf{z})
\end{align*} \quad (B.4)$$

The linearized and normalized analogue of (B.4) around an operating point can be written as:

$$\begin{align*}
\dot{\mathbf{x}} &= \mathbf{A}\mathbf{x} + \mathbf{B}\mathbf{u} \\
\mathbf{y} &= \mathbf{C}\mathbf{x}
\end{align*} \quad (B.5)$$

where the variables $\mathbf{x}$, $\mathbf{y}$, $\mathbf{u}$ are the normalized deviations from the steady-state values at the OP $\{\mathbf{z}_{\text{op}}, \ \mathbf{v}_{\text{op}}, \ \mathbf{w}_{\text{op}}\}$.

For feedback, the intake BG-rate $x_{bg}$ is estimated as follows from the measured fresh air mass flow, the measured exhaust BG-rate $x_{bg,em}$ (cp.
Equation (A.6)) and the estimated EGR mass flow.

\[
x_{bg} = \frac{\dot{m}_{egr} x_{bg,cm}}{\dot{m}_{egr} + \dot{m}_{hfm}} \tag{B.6}
\]

Figure B.1 shows a validation example of the nonlinear model (B.4) for the intake and exhaust manifold pressures and the exhaust air-fuel ratio \( \lambda \), respectively.

## B.2 Control Structure

The air-path control structure is composed of two controllers. The first controller is a multivariable gain scheduled controller for the boost pressure and the intake BG-ratio, the other one is a PI controller for the intake BG-ratio. The control authority is switched between the two controllers, depending on the current OP of the engine. The PI controller is active in low speed and load regions, where the controllability of the boost pressure is poor due to the low energy content in the exhaust gas. The VGT actuator, in case of an activated PI controller, is controlled in a feedforward manner.

The controlled variables are the measured intake pressure \( p_{im} \) and the estimated intake BG-ratio \( x_{bg} \) (cp. Equation (B.6)). The manipulated variables are the EGR and VGT actuator positions \( u_{egr} \) and \( u_{vgt} \). Bumpless transfer is implemented on both controllers of the structure, in order to avoid actuator steps due to controller switching.

In the following, the synthesis of the multivariable controller is described, while the PI controller is not further discussed.

### Controller Synthesis

The multivariable controller is designed as an optimal linear output feedback controller derived with the Linear-Quadratic-Gaussian method with Loop-Transfer-Recovery (LQG/LTR) according to [61,72,73]. Robustness issues of the controlled system are discussed in Section B.2. The controller synthesis is based on the plant (B.5).

In order to eliminate steady-state offset, the plant output is extended with
proportional-integral (PI) elements (state $\nu$) for the controller synthesis.

$$\begin{bmatrix} \dot{x} \\ \dot{\nu} \end{bmatrix} = \begin{bmatrix} A & 0 \\ B_1C & 0 \end{bmatrix} \begin{bmatrix} x \\ \nu \end{bmatrix} + \begin{bmatrix} B \\ 0 \end{bmatrix} u$$  \hspace{1cm} (B.7)$$

$$y = \begin{bmatrix} C & I \end{bmatrix} \begin{bmatrix} \dot{x} \\ \nu \end{bmatrix}$$

where $B_1$ is a diagonal matrix defining the time constants of the PI elements. The values of these time constants need to be chosen in accordance with the plant characteristics and the specifications of the control system. The weighting matrices of the regulator ($Q$ and $R$) and of the observer ($Q_o$ and $R_o$) are obtained as follows:

$$Q_o = BB^T$$  \hspace{1cm} (B.8)$$

$$R_o = \mu \cdot I$$  \hspace{1cm} (B.9)$$

$$Q = \begin{bmatrix} C & I \end{bmatrix}^T \begin{bmatrix} C & I \end{bmatrix}$$  \hspace{1cm} (B.10)$$

$$R = \rho \cdot I$$  \hspace{1cm} (B.11)$$

where the parameters $\mu$ and $\rho$ are used to tune the observer and regulator of the controller, respectively. The adequate tuning of $\mu$ and $\rho$ is crucial for defining the robustness and bandwidth of the controlled system.

The controller gain $\tilde{K}$ is obtained as the solution to the LQR problem with the extended plant (B.7) and contains the state feedback gains $K$ and $K_I$ on the observed state of the plant $\hat{x}$ and on the integrator state $\nu$, respectively:

$$\tilde{K} = \begin{bmatrix} K & K_I \end{bmatrix}$$  \hspace{1cm} (B.12)$$

The observer gain matrix $H$ is obtained as the solution to the dual problem. To obtain a good performance throughout the region of operation, the air-path controller is synthesized at the four vertex operating points within the speed and load intervals $[n_{e,low}, n_{e,high}]$, $[q_{inj,low}, q_{inj,high}]$. During operation, the matrices of the controller system are obtained by bilinear interpolation on the four vertex controllers $k$ designed.

$$M_c = \sum_{k=1}^{4} \phi_k M_{c,k}$$  \hspace{1cm} (B.13)$$
where $M_{c,k}$ represents the matrices $A_{c,k}$, $B_{c,k}$, $C_{c,k}$, or $D_{c,k}$ of the controller system on vertex $k$. The weighting parameter $\phi_k$ is obtained with the vertex OP $k \in \{1, \ldots, 4\}$.

$$\phi_k = \left(1 - \frac{|n_e - n_e,k|}{n_{e,\text{high}} - n_{e,\text{low}}} \right) \left(1 - \frac{|q_{\text{inj}} - q_{\text{inj},k}|}{q_{\text{inj,high}} - q_{\text{inj,low}}} \right)$$

Figure B.2: Singular value trajectories of the loop gain $L$, the sensitivity $S$ and the complementary sensitivity $T$ of the controlled system at the vertices. The diamond indicates the frequency of the non-minimum phase zero of the plant (if present).
Robustness and Anti Reset-Windup

Robustness of a controller derived with the LQG/LTR method is not guaranteed for non-minimum phase plants [61]. However, if the bandwidth of the controlled system is sufficiently small with respect to the non-minimum phase zero with the lowest frequency, the robustness specifications of the controlled system can be met [72]. Figure B.2 shows the loop gain $L(s)$ and the sensitivities $S(s)$, $T(s)$ of the controlled system at each vertex OP. Clearly, the crossover frequency is more than one octave lower than the frequency of the non-minimum phase zero (indicated by a diamond, if present). This is a compromise between robustness [72] and a high bandwidth of the closed loop.

An analysis of the robustness for the gain scheduled system according to [73, 74] was out of the scope of the project.

Due to the characteristics of the system (no poles on the imaginary axis), integrators are required to attain the reference values without steady-state offset. As a result, anti reset-windup and bumpless-transfer (AWBT) is indispensable to avoid significant overshoot or even instability in case of actuator saturation or switching from a different control mode. Furthermore, for the same cases, poor estimates of the observer state $\hat{x}$ have to be avoided. The latter issue can be solved by feeding back the actual manipulated variable after saturation $u_{\text{sat}}$, as shown in Fig. B.3 and as proposed in [75].

The integrator state $\nu$ of the controller is adapted for AWBT with a coord-
dinate transformation with the integrator state feedback $K_I$ as follows:

$$\tilde{\nu} = K_I \nu$$  \hspace{1cm} (B.15)

As a result of this transformation, each state variable $\tilde{\nu}_i$ of the integrator influences just one manipulated variable and therefore can be halted in case of saturation of the corresponding actuator.

$$u_I = -K_I \nu = -K_I K_I^{-1} \tilde{\nu}$$  \hspace{1cm} (B.16)

$$\dot{\tilde{\nu}} = -K_I B_I e$$  \hspace{1cm} (B.17)

Figure B.3 shows the structure of the multivariable controller with AWBT and including the saturation. Each matrix of the controller block is obtained from interpolation on the four vertex controllers according to Equation (B.13). The PI controller for controlling the BG-ratio at low speed and load is not shown in the Figure.

Figure B.4 shows the control performance of the air path controller during a section of the FTP-72. The multivariable controller is active when the controller switch in the fourth row plot is equal to 1. Otherwise, the VGT is feedforward controlled, and the BG-ratio is controlled via the PI-controller. Note that the reference value for the intake pressure is not feasible when the multivariable controller is not active.
Figure B.4: Performance of the air path controller in a section of the FTP-72. The multivariable controller is active when the controller switch in the fourth row plot is on.
Appendix C

Combustion Characteristics Estimation

In this chapter, the processes running on the device for online analysis of the combustion characteristics, and the hardware of the device (OCA) itself are described. The description is based on the work of Michael Merz, Lukas Wunderli, and [76].

C.1 Hardware and Interfaces

Microcontroller Device

<table>
<thead>
<tr>
<th>Feature</th>
<th>Characteristic</th>
</tr>
</thead>
<tbody>
<tr>
<td>Name</td>
<td>STM32F107VC ComStick</td>
</tr>
<tr>
<td>Processor</td>
<td>Cortex-M3-ARM</td>
</tr>
<tr>
<td>Clock</td>
<td>72 MHz</td>
</tr>
<tr>
<td>Memory</td>
<td>256 kByte Flash</td>
</tr>
<tr>
<td></td>
<td>64 kByte SRAM</td>
</tr>
<tr>
<td>I/O</td>
<td>2 CAN-modules</td>
</tr>
<tr>
<td></td>
<td>2 ADC-modules (12 bit)</td>
</tr>
</tbody>
</table>

For the characteristics of the ADC, see [77, p. 73 ff.].

Input Channels

Cylinder Pressure: The characteristics of the Kistler in-cylinder pressure sensor and sampling are provided in the following table.
<table>
<thead>
<tr>
<th>Type</th>
<th>Signal range</th>
<th>Amp. range</th>
<th>Samp. range</th>
</tr>
</thead>
<tbody>
<tr>
<td>Kistler 6041AQ03S31</td>
<td>0...250 bar</td>
<td>10 V</td>
<td>2047 (12 bit)</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>[−10...10] V</td>
</tr>
</tbody>
</table>

**Triggering and Referencing:** The characteristics of the crank angle trigger and of the reference pulse are provided in the following table. The reference pulse is triggered at 0 °ECA. In accordance with the interrupt functionality described in Section C.2, a pressure sensor on cylinder three has to be used for the combustion analysis.

<table>
<thead>
<tr>
<th>Pulse</th>
<th>Amplitude</th>
<th>Periodicity</th>
<th>Pulse width</th>
</tr>
</thead>
<tbody>
<tr>
<td>Reference</td>
<td>5 V</td>
<td>720 °CA</td>
<td>0.05 °CA</td>
</tr>
<tr>
<td>Position</td>
<td>5 V</td>
<td>0.2 °CA</td>
<td>0.1 °CA</td>
</tr>
</tbody>
</table>

**CAN Bus Interface**

The following CAN-frames are implemented for communication with the ES910:

<table>
<thead>
<tr>
<th>Rx-Frame</th>
<th>ID: 0x0202, 8 bytes of data</th>
</tr>
</thead>
<tbody>
<tr>
<td>Byte 0</td>
<td>High-byte 16-bit SOI [#samples w.r.t. −40 °ATDC]</td>
</tr>
<tr>
<td>Byte 1</td>
<td>Low-byte SOI.</td>
</tr>
<tr>
<td>Byte 2</td>
<td>High-byte 16-bit $p_{im}$ [(bara/5) · 2047 · (100/50)]</td>
</tr>
<tr>
<td>Byte 3</td>
<td>Low-byte $p_{im}$.</td>
</tr>
<tr>
<td>Byte 4</td>
<td>Empty</td>
</tr>
<tr>
<td>Byte 5</td>
<td>Empty</td>
</tr>
<tr>
<td>Byte 6</td>
<td>Empty</td>
</tr>
<tr>
<td>Byte 7</td>
<td>Empty</td>
</tr>
<tr>
<td>Tx-Frame</td>
<td>ID: 0x0201, 8 bytes of data</td>
</tr>
<tr>
<td>----------</td>
<td>-----------------------------</td>
</tr>
<tr>
<td>Byte 0</td>
<td>8-bit $\phi_{coc,m}$ [#samples w.r.t. $-10 , ^\circ$ATDC].</td>
</tr>
<tr>
<td>Byte 1</td>
<td>1-bit update-oscillator (bit-pos: 0). 7-bit $\tau_{ign}$ [#samples] (bit-pos: [1 . . . 7]).</td>
</tr>
<tr>
<td>Byte 2</td>
<td>8-bit $p_{cyl,max}$ [bara].</td>
</tr>
<tr>
<td>Byte 3</td>
<td>8-bit $\phi_{coc,pi}$ [#samples w.r.t. $-15 , ^\circ$ATDC].</td>
</tr>
<tr>
<td>Byte 4</td>
<td>8-bit $Q_{pi}/Q$ [%] fraction of energy in the pilot injection.</td>
</tr>
<tr>
<td>Byte 5</td>
<td>Low-byte 12-bit $dQ/d\phi_{max}$.</td>
</tr>
<tr>
<td>Byte 6</td>
<td>4 highest bits of $dQ/d\phi_{max}$ (bit-pos: [0 . . . 3]). 4 lowest bits of 12-bit IMEP (bit-pos: [4 . . . 7]).</td>
</tr>
<tr>
<td>Byte 7</td>
<td>8 highest bits of IMEP.</td>
</tr>
</tbody>
</table>

**Interface to Engine Cycle**

The configuration of the engine is given in the following table:

<table>
<thead>
<tr>
<th># Cylinder</th>
<th>Firing order</th>
<th>Firing CA</th>
<th>Position</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1</td>
<td>0, 720 °ECA</td>
<td>Front right</td>
</tr>
<tr>
<td>2</td>
<td>3</td>
<td>240 °ECA</td>
<td>Middle right</td>
</tr>
<tr>
<td>3</td>
<td>5</td>
<td>480 °ECA</td>
<td>Back right</td>
</tr>
<tr>
<td>4</td>
<td>2</td>
<td>120 °ECA</td>
<td>Front left</td>
</tr>
<tr>
<td>5</td>
<td>4</td>
<td>360 °ECA</td>
<td>Middle left</td>
</tr>
<tr>
<td>6</td>
<td>6</td>
<td>600 °ECA</td>
<td>Back left</td>
</tr>
</tbody>
</table>

**C.2 Process Functionality**

**Program Structure**

The program is composed of the following C-files.
<table>
<thead>
<tr>
<th>Program</th>
<th>Functionality</th>
</tr>
</thead>
<tbody>
<tr>
<td>main.c</td>
<td>Initialization.</td>
</tr>
<tr>
<td></td>
<td>Handling of serial data request (invocation of the corresponding method).</td>
</tr>
<tr>
<td></td>
<td>handling of incoming CAN messages (invocation of CAN-receive method).</td>
</tr>
<tr>
<td>combustion.c</td>
<td>Methods for the combustion analysis invoked by the ISR.</td>
</tr>
<tr>
<td>communication.c</td>
<td>Methods for configuration and execution of communication tasks and interfaces (serial and CAN).</td>
</tr>
<tr>
<td>configuration.c</td>
<td>Methods for device configuration as provided by the manufacturer.</td>
</tr>
<tr>
<td>hw_config.c</td>
<td>Basic device configuration and initialization methods as provided by the manufacturer.</td>
</tr>
<tr>
<td>stm32f10x_it.c</td>
<td>Main interrupt service routines as provided by the manufacturer.</td>
</tr>
<tr>
<td></td>
<td>(invocation of the methods for the combustion analysis)</td>
</tr>
</tbody>
</table>

**Interrupt Service Routines**

The interrupts are triggered by a counter which counts the position pulses. The functionality of the interrupt service routines (ISR) to be executed is defined by the current value of the counter, which is compared to the triggering variable of each ISR.

**Sensor-Pegging ISR**

The sensor-pegging ISR is required to reference the high-pressure signal to the pressure in the cylinder. It is carried-out close to BDC of the intake stroke, where the pressure within the cylinder is assumed to be equal to the intake manifold pressure and thus known. Furthermore, during the sensor-pegging interrupt routine, the transmission via CAN of the estimated combustion characteristics of the previous cycle is invoked.
### C.2. Process Functionality

<table>
<thead>
<tr>
<th>Counter value</th>
<th>1475</th>
</tr>
</thead>
<tbody>
<tr>
<td>CA from engine TDC</td>
<td>295 °ECA</td>
</tr>
<tr>
<td>CA cylinder</td>
<td>175 °CCA</td>
</tr>
<tr>
<td><strong>Functionality</strong></td>
<td>DMA-configuration. ADC-configuration. High pressure signal sampling (20 samples). DMA-configuration for analysis-sampling. ADC-configuration for analysis-sampling. Sensor pegging with ECU intake pressure. CAN send.</td>
</tr>
</tbody>
</table>

---

**Sampling ISR**

The cylinder pressure is sampled from 40 °BTDC to 60 °ATDC. It is assumed that the combustion of all fueling events of the cycle completely occur within the sampled interval.

<table>
<thead>
<tr>
<th>Counter value</th>
<th>2200</th>
</tr>
</thead>
<tbody>
<tr>
<td>CA from engine TDC</td>
<td>440 °ECA</td>
</tr>
<tr>
<td>CA cylinder</td>
<td>320 °CCA (40 °before combustion TDC).</td>
</tr>
<tr>
<td><strong>Functionality</strong></td>
<td>High pressure signal sampling (500 samples).</td>
</tr>
</tbody>
</table>

---

**Signal-Processing ISR**

The signal-processing ISR triggers the estimation of the combustion characteristics with the data obtained during the precedent sampling interrupt.
### C.3 Signal Processing

This section describes the algorithms used for the scaling of the pressure and for the estimation of the characteristics of combustion [40].

#### Scaling of the Cylinder Pressure

Computation time is a crucial limiting factor for the online analysis of the combustion characteristics with the hardware chosen. Therefore, all variables and pressure signals have been scaled such that, as far as possible, all calculations can be performed with int32 data type while limiting numerical errors due to phenomena such as cancellation or round-off.

#### Heat Release Rate Estimation

The heat release rate (HRR) is the basis for the estimation of the center of combustion (CoC) and of the start of combustion (SoC). The model
C.3. Signal Processing

for the HRR is based on the first law of thermodynamics and a constant ratio of specific heats $\kappa$ as described in [78]. With the sampled pressure trace and a central second-order finite difference scheme for the derivative $d p_c / d \phi$, the HRR can be approximated as follows.

\[
\frac{d}{d \phi} \frac{d Q}{d \phi} = \frac{\kappa p_{c,i}}{\kappa - 1} \left[ \frac{d}{d \phi} V_i \right] + \frac{V_i}{\kappa - 1} \frac{p_{c,i+1} - p_{c,i-1}}{2 \Delta \phi} \tag{C.1}
\]

Where $V_i$ and $p_{c,i}$ represent the in-cylinder volume and pressure at sample $i$, respectively. Those variables in Eq. (C.1), which are not influenced by the process, i.e. the derivative of the cylinder volume with respect to the CA and the ratio of specific heats, are computed and scaled off-line.

SoC Estimation and Filtering

The estimation of the start of combustion (SoC) of the main injection is one of the key issues. Several methods for SoC detection have been proposed in the literature, where the contributions of [79] and [80] provide a good overview. The method used here is similar to the one proposed by [81], where the SoC is assumed to coincide with the maximum of the second derivative of the HRR.

In contrast to the approach in [81], in this work it is not assumed that the maximum of the second derivative of the HRR appears at the same location as the maximum of the third derivative of the pressure trace. Although [81] showed that the maximum of the third derivative of the pressure provides good SoC estimates, the maximum of the second derivative of the HRR in this work needs to be computed and thus is available anyway. Due to the fact that numerical derivation is sensitive to measurement noise, which is present in the HRR, filtering of the HRR is crucial in order to obtain precise SoC estimates. To avoid phase lag, the HRR is filtered forward and backward with a fourth-order Chebychev filter with 0.05 dB ripple and a normalized passband edge frequency of 0.06. The filtering is also critical with respect to computational burden and in connection with numerical precision and avoidance of overflow. To limit the computational burden, the filtered HRR $d Q_f / d \phi$ is thus calculated only in a reduced range where the SoC is assumed to occur. The forward filtering is obtained with the
HRR from $i_{start}$ to $i_{end}$ as follows.

$$\left[ \frac{d}{d\phi} Q_{ff} \right]_i = \frac{b^{T}_{\text{cheb}} \left[ \frac{d}{d\phi} Q \right]_{[i, i-1, i-2, i-3, i-4]} a^{T}_{\text{cheb}} \left[ \frac{d}{d\phi} Q_{ff} \right]_{[i-1, i-2, i-3, i-4]}}{a^{T}_{\text{cheb}} \left[ \frac{d}{d\phi} Q_{ff} \right]_{[i-1, i-2, i-3, i-4]}}$$ (C.2)

Subsequently, the backward filtering is carried-out with the forward-filtered HRR from $i_{end}$ to $i_{start}$.

$$\left[ \frac{d}{d\phi} Q_f \right]_i = \frac{b^{T}_{\text{cheb}} \left[ \frac{d}{d\phi} Q_{ff} \right]_{[i, i+1, i+2, i+3, i+4]} a^{T}_{\text{cheb}} \left[ \frac{d}{d\phi} Q_f \right]_{[i+1, i+2, i+3, i+4]}}{a^{T}_{\text{cheb}} \left[ \frac{d}{d\phi} Q_f \right]_{[i+1, i+2, i+3, i+4]}}$$ (C.3)

Where $a_{\text{cheb}}$ and $b_{\text{cheb}}$ are the vectors of filter coefficients.

With the central second-order finite difference scheme for second derivatives, the SoC of the main injection is found at the CA corresponding to the index $i_{soc}$ that is found with the following operation.

$$\arg \max_{i} \left( \frac{\left[ \frac{d}{d\phi} Q_f \right]_{i+1} - 2 \left[ \frac{d}{d\phi} Q_f \right]_i + \left[ \frac{d}{d\phi} Q_f \right]_{i-1}}{\Delta \phi^2} \right)$$ (C.4)

The search range is limited from SOI to the maximum HRR.

**CoC Estimation**

The $\phi_{coc,m}$ is defined as the CA at which 50% of the available chemical energy of the fuel from the main injection has been released. Under the assumption that the fuel of pilot injections has been oxidised completely at the index of start of combustion of the main injection $i_{soc}$, the $\phi_{coc,m}$ is found as the CA corresponding to the highest index $i_{coc,m}$ which fullfills the following inequlitity.

$$\frac{\sum_{j=i_{soc}}^{i_{coc,m}} \left[ \frac{d}{d\phi} Q \right]_j}{\sum_{j=i_{soc}}^{n} \left[ \frac{d}{d\phi} Q \right]_j} \leq 0.5$$ (C.5)
IMEP Estimation

The gross IMEP is defined as the specific piston work over the compression and expansion strokes [56].

\[
\text{IMEP} = \frac{1}{V_d} \int_{BDC_c}^{BDC_e} p_c \, dV \quad (C.6)
\]

With the compression and expansion work \( \Delta W_i \), and based on Eq. (C.6), the IMEP can thus be estimated with the explicit Euler integration scheme as follows.

\[
\text{IMEP} \approx \frac{1}{V_d} \left( \Delta W_c + \sum_j p_{c,j} \Delta V_j + \Delta W_e \right) \quad (C.7)
\]

Due to the fact that no combustion is assumed to occur outside of the sampled interval, the non-sampled parts of the compression and expansion work \( \Delta W \) are obtained with a polytropic process over the interval \([\phi_1, \phi_2]\).
with reference cylinder pressure $p_{c,r}$ and reference volume $V_r$ at index $r$.

$$\Delta W = \int_{\phi_1}^{\phi_2} p_{c,r} \left( \frac{V_r}{V} \right)^n \, dV$$  \hspace{1cm} (C.8)

The reference CA index $r$ corresponds to the beginning or the end of the sampled interval, respectively. Under the assumption of a constant polytropic coefficient $n$, Equation (C.8) can be simplified.

$$\Delta W = \frac{p_{c,r} V_r^n}{1 - n} \left( V_{\phi,2}^{1-n} - V_{\phi,1}^{1-n} \right)$$  \hspace{1cm} (C.9)

Figure C.1 shows an example for the traces of the cylinder pressure (top), the normalized HRR and the normalized second derivative of the filtered HRR (middle), and the cumulative HRR (bottom) with the events of the SOI of the main injection, the SoC and the center of the main combustion. The peak of the second derivative of the filtered HRR clearly indicates the SoC, and the noisy HRR illustrates the importance of appropriate filtering. In the bottom plot the released energy from pilot injections is clearly visible.
Appendix D

Learning Feedforward Control

The on-line adaptation of look-up tables, so-called learning feedforward control, is a simple yet powerful technique for auto-calibration and identification in connection with varying operating conditions, where the varying operating conditions are not represented by phenomenological modeling. The learning feedforward control implementation that has been used in context of this work is described in the following. It is based on the findings in [62].

Given a matrix of grid-points (nodes) of some look-up table or map \( M \in \mathbb{R}^{n,m} \). The two vectors \( v_1 \in \mathbb{R}^{n,1} \), \( v_2 \in \mathbb{R}^{m,1} \) define the operating conditions, which correspond to each grid-point in \( M \). In this work, the OP is defined by the engine speed \( n_e \) and by the amount of injection corresponding to the load \( q_{\text{inj}} \). The vector \( w \in \mathbb{R}^{n \cdot m,1} \) is a reshaped version of the map \( M \). The output \( y \) of the look-up table is obtained with the vector of interpolation weights \( \Phi (n_e, q_{\text{inj}}) \in \mathbb{R}^{n \cdot m,1} \) for the current OP.

\[
y = w^T \cdot \Phi (n_e, q_{\text{inj}}) \quad (D.1)
\]

In this work, bilinear interpolation has been used. Thus the elements \( \phi_k \) of \( \Phi \) are unequal to zero solely on the four closest nodes around the current OP. Furthermore, the following equation holds.

\[
\sum_{k=1}^{n \cdot m} \phi_k \equiv 1, \quad \phi_k \in [0, 1] \quad \forall k \quad (D.2)
\]

The on-line modification or adaptation of the vector \( w \) is achieved with the same vector of interpolation weights \( \Phi \) and some operation such as recursive least squares [62]. For the purposes of this work, integration of
the variable \( e \) has been used for adaptation of the vector \( \mathbf{w} \).

\[
\frac{d}{dt} \mathbf{w} = \Phi (n_e, q_{inj}) \cdot e
\]

where the variable \( e \) corresponds to the input of the integrator which has been substituted by the learning feedforward control implementation. In case of transport delay on the signals which are used to obtain the variable \( e \), the variables \( n_e \) and \( q_{inj} \), defining the current OP, and thus the interpolation vector \( \Phi \) for the on-line modification, have to be delayed accordingly.
List of Tables

1.1 Diesel engine actuators and their main purposes 3
1.2 EU emission standards for diesel engine passenger cars. (Limited to the NO$_x$ and PM masses emitted) 5
2.1 Characteristics of the test bench engine 10
2.2 Additional sensors on the engine 11
5.1 Parameters of the lead and lag element 50

List of Figures

1.1 Conventional engine control structure without emission feedback. The drift influence on the emissions remains uncontrolled. 4
1.2 Feedback control of the PM and NO$_x$ emissions reduces the influences of drift on the engine-out emissions. Furthermore, it provides a convenient possibility to adapt the engine calibration such that the system with aftertreatment optimally meets the legislative limits. 7
2.1 Sketch of the engine with the most important components. 9
2.2 Architecture of the engine control devices with the rapid prototyping device ES910, the production type ECU and additional devices. 11
3.1 Block diagram of the control oriented PM model structure. The inputs to the model are obtained from ECU-data. 15
3.2 Normalized steady-state NOₓ (left) and PM (right) emissions. 17
3.3 Modeled and measured NOₓ (left) and PM (right) emissions together with the coefficient of determination of each emission species model. .......................... 18
3.4 Modeled and measured (gray) sensor signal \( z_{s,i} \) of the NOₓ (left) and PM (right) emissions on a part of the NEDC. The OP trajectory is shown in the bottom plots. ................. 19
3.5 Measured emissions (gray) together with the modeled sensor signal \( z_{s,i} \) and the modeled engine-out emissions \( \zeta_i \) of NOₓ (left) and PM (right) on a part of the NEDC. .................. 19
3.6 Block diagram of the basic processes relevant for the engine-out NOₓ and PM emissions. Inputs: intake pressure and temperature, BG rate, rail pressure, SOI, injection amount, and swirl valve position. ................................. 20
3.7 Maps of the OP-dependent sensitivities \( \theta_{i,j} \) for the NOₓ model (left) and PM model (right) for normalized variations of the SOI (top), boost pressure (2nd row), BG-ratio (3rd row), and swirl valve (bottom). The normalization value of each input channel is specified above the left graphs. ... 22
3.8 Influence of \( \pm 3^\circ \) SOI and \( \pm 3\% \) BG-rate variations on the PM and NOₓ at various operating points ................. 23
3.9 Influence of the swirl valve on the NOₓ and PM at various operating points (left). PM emissions in function of the swirl valve position \( u_{sv} \) (right) ....................... 25

4.1 Block diagram of the Kalman filter with the estimated engine-out emission species \( \hat{\zeta}_i \) that is observed with the measured signal \( z_{s,i} \) and the output of the basic engine-out emission model \( u_{i,(1)} \). ................................. 28
4.2 Nyquist plots of the Kalman filters for the NOₓ (left) and PM (right) emissions. The phase margin (pm) is indicated in the graphs. ................................. 29
4.3 Compared to the measured NOₓ and PM emissions, the estimated engine-out emissions \( \hat{\zeta}_i \) evidently provide predictive information. The estimated sensor signal \( \hat{z}_{s,i} \) is also shown. 30
4.4 Block diagram of the control structure including the engine, the controllers, the sensors, and the observer for the engine-out emissions. The proposed control structure consists of the PI controller for the PM, and of the NO\textsubscript{X} controller with the P element on the SOI and the learning feedforward BG-rate adaptation \(\delta r_{bg}\) as an I element. The air-path controller is not part of the proposed control structure. 32

4.5 Value of the first diagonal element of the relative gain array. The level close to one justifies the use of SISO loops. 33

4.6 Magnitudes of the loop gain \(L\), the sensitivity \(S\), and the complementary sensitivity \(T\) of the PM and NO\textsubscript{X} loops. In the right plot, the loop gain \(L_{bg}\) is shown, which is obtained by considering solely the BG-rate adaptation channel within the NO\textsubscript{X} controller. 34

4.7 Normalized DC gain of the plant from the swirl valve to the PM emissions with a closed NO\textsubscript{X} control loop. The sign change of the DC gain is highlighted with a thick line. 36

4.8 Block diagram of the uncertain control loop including the controller \(C(s)\), the uncertain plant \(\tilde{P}(s)\) and the observer \(K(s)\), which is based on the nominal plant \(P(s)\). 37

4.9 Lower bound on the stability margin of the uncertain plant. 38

4.10 Measured emissions, reference trajectories, and observed engine-out emissions for stepwise changes in the PM reference (top plots). Trajectories of the SOI and the swirl valve position (bottom plots). 39

4.11 Measured emissions, reference trajectories, and emission trajectories modeled without NO\textsubscript{X} control on the SOI (top plots). Basis BG rate reference \(r_{bg,base}\), BG rate \(x_{bg}\), and BG reference adapted by the NO\textsubscript{X} controller (bottom left). SOI control action \(\delta u_{soi}\) (bottom right). 39

4.12 Cumulative emissions with (gray) and without (black) emission control for various drift-afflicted experiments. The drift impact is illustrated by the minimum-surface ellipsoids which contain all points for active and deactivated emission control, respectively. The results for active emission control are highlighted in the detail plot. 40
4.13 Load ramp with 1.5s duration. The graphs show the measured, estimated, and reference emissions (top), swirl valve and SOI (2\textsuperscript{nd} row), boost pressure and BG rate (3\textsuperscript{rd} row), and OP trajectory (bottom). ........................................... 42

4.14 Comparison of active emission control with conventional control (ECU) for the load ramp as shown in Figure 4.13. The bottom left plot shows the normalized, cumulative PM emissions. ........................................... 43

4.15 Load ramp with 5s duration. The graphs show the measured, estimated, and reference emissions (top), swirl valve and SOI (2\textsuperscript{nd} row), boost pressure and BG rate (3\textsuperscript{rd} row), and OP trajectory (bottom). ........................................... 45

4.16 Comparison of active emission control with conventional control (ECU) for the load ramp as shown in Figure 4.15. The bottom left plot shows the normalized, cumulative PM emissions. ........................................... 46

5.1 Modeled and estimated $\phi_{\text{coc,m}}$ (left) and IMEP (middle) for variations in the SOI $\Delta u_{\text{soi}}$ and amount of main injection $\Delta q_{\text{inj,m}}$ (right). ........................................... 49

5.2 Singular values of the loop gain $L$, the sensitivity $S$, the complementary sensitivity $T$, and the inverse weighting function $W_e^{-1}$. ........................................... 51

5.3 Block diagram of the combustion controller structure with the feedforward path. ........................................... 52

5.4 Block diagram of the cascaded control loop including the engine with sensors and observers, the emission control structure, the OCA, and the cascaded combustion controller. ... 53

5.5 The top plots show the $\phi_{\text{coc,m}}$ (left) and IMEP (right) and their corresponding reference values for steps in the $\phi_{\text{coc,m}}$ reference at 1500 rpm, 100 Nm. The bottom plots show the SOI and the injected fuel amount, respectively. .............. 53

5.6 The top plots show the $\phi_{\text{coc,m}}$ (left) and IMEP (right) and their corresponding reference values for steps in the IMEP reference at 1500 rpm, 100 Nm. The bottom plots show the SOI and the injected fuel amount, respectively. .............. 54
<table>
<thead>
<tr>
<th>Figure</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.7</td>
<td>Influence of an EGR variation at 1300 rpm and 60 Nm with active combustion control. The top right plot shows the measured air-to-fuel ratio $\lambda$. Significant control action on the SOI (bottom left) is required to compensate the influence on the $\phi_{\text{coc,m}}$ (top left).</td>
<td>55</td>
</tr>
<tr>
<td>5.8</td>
<td>Influence of a NO$_x$ reference step at 1400 rpm with active and deactivateated combustion controller.</td>
<td>56</td>
</tr>
<tr>
<td>6.1</td>
<td>Section of the urban part of the NEDC. The observed engine-out emissions, and the delayed, filtered, measured emissions follow their respective reference trajectories. The trajectories of the manipulated variables SOI and swirl valve as well as the OP are shown in the lowest four plots.</td>
<td>59</td>
</tr>
<tr>
<td>6.2</td>
<td>Overland part of the NEDC. The measured emissions are set to the reference value except in case of actuator saturation.</td>
<td>60</td>
</tr>
<tr>
<td>6.3</td>
<td>The proposed controller compared to the results obtained with standard ECU control. In the two bottom plots, the normalized cumulative emissions are shown.</td>
<td>61</td>
</tr>
<tr>
<td>6.4</td>
<td>The cumulative measured and reference engine-out emissions over the NEDC.</td>
<td>61</td>
</tr>
<tr>
<td>6.5</td>
<td>Part of the FTP-72 cycle with the cascaded emissions and combustion controller. The plots show the estimated, measured and reference NO$<em>x$ and PM emissions, the CoC, the IMEP, the normalized manipulated variables $\delta u</em>{i,n}$ of the combustion controller, and the speed and load trajectories (from top to bottom).</td>
<td>64</td>
</tr>
<tr>
<td>6.6</td>
<td>Comparison of the cascaded controller with a conventional controller on an extract of the FTP-72 cycle. The plots show the NO$<em>x$ and PM emissions, the BG rate $x</em>{\text{bg}}$, the CoC, the IMEP, and the engine speed and load (from top to bottom).</td>
<td>65</td>
</tr>
<tr>
<td>6.7</td>
<td>Section of the FTP-72. The plots show the emissions of NO$<em>x$ and PM, emission controller manipulated variables SOI and $u</em>{\text{sv}}$, and the engine speed and brake torque (from top to bottom).</td>
<td>67</td>
</tr>
</tbody>
</table>
6.8 Comparison of the emission controller with and without cascaded control of the combustion. The plots show the NO\textsubscript{x} and PM emissions, the CoC, the IMEP, the combustion controller activation (only relevant for cascaded control), and the engine speed and load (from top to bottom). 68

B.1 Modeled and measured (dotted) signals for load steps around 2800 rpm and 180 Nm. The left plot shows the pressures in the exhaust (gray) and intake manifolds, respectively. The right plot shows the exhaust air-to-fuel ratio. 77

B.2 Singular value trajectories of the loop gain $L$, the sensitivity $S$ and the complementary sensitivity $T$ of the controlled system at the vertices. The diamond indicates the frequency of the non-minimum phase zero of the plant (if present). 80

B.3 The output feedback controller with integrator and AWBT. $I$ is the identity matrix and $k_{aw}$ is a diagonal matrix to avoid windup on each channel of the saturated manipulated variable $u_{sat}$. 81

B.4 Performance of the air path controller in a section of the FTP-72. The multivariable controller is active when the controller switch in the fourth row plot is on. 83

C.1 Cylinder pressure (top), normalized HRR (unfiltered) and normalized second-order derivative of the filtered HRR (middle), cumulative HRR (bottom). The markers indicate the locations of the corresponding events. 92
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