DIESEL ENGINE AIR-PATH MANAGEMENT
Commercial vehicle applications

BÁRDOS ÁDÁM

Doctoral supervisor: Dr. Németh Huba

Kandó Kálmán Doctoral School
Department of Automotive Technologies
Faculty of Transportation Engineering and Vehicle Engineering
Budapest University of Technology and Economics
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To my loving Family.
This thesis deals with the air-path management of commercial vehicle diesel engines. Developers of modern diesel engines have to face severe restrictions in conventional air pollutants such as nitrogen oxides and particulate matter as well as in carbon dioxide as a greenhouse gas. Parallel, especially in the commercial vehicle market, the need for economical operation and reliability is dominating. Among several methods (DPF, SCR, improved injection, etc.) which are used to handle air pollutant emission, the precise control of the cylinder charge composition seems to be a cost-effective solution. The installation of flap valves with a fast and accurate position control into the engine air-path offers several possible new and extended functionalities which are investigated in this work. These include cylinder charge composition control, exhaust gas thermal management, brake blending, and automated manual transmission support.

The first part of this thesis summarizes the challenges which should be solved by modern diesel engine developers and the motivation of this thesis. After that, possible new applications of flap valves installed in the engine air-path are worked out. The optimal realization of these targeted possible new functions gives the aim of the following research. The intake manifold oxygen concentration is defined as a novel performance output of the controller. Finally, in the framework of a thorough literature review, the state of the art of diesel engine control and mechanism of the nitrogen oxide and particulate matter formation is investigated.

There are numerous possible location for flap valves in the engine intake and exhaust system (e.g., upstream the turbine, downstream the turbine, etc.) which could be used similarly for the realization of the targeted functions. However, flap valve operation at different locations of the air-path system shows a different effect on the engine operation measures, e.g., on fuel consumption, emission. To be able to choose an optimal solution the detailed model of the investigated engine is built and validated in the GT-Suite environment. Using the simulation model for a comparison an optimal air-path flap valve actuator setup is derived.

In the third part of this thesis, the backpressure control function is worked out. As a first step, the control aims and requirements are defined. Thereafter, a first-engineering-principle-based, mean-value, control-oriented, nonlinear model of the engine and the actuator is described and validated with test bench measurements. The design and performance comparison of four model-based controller structures are presented: an LQ servo, an LQ servo with a model-inversion-
based feedforward, an LPV model based H-infinity and a sliding mode. The controller performance and the compliance of requirements are evaluated in three different test cycles on a medium-duty diesel engine, simulating brake blending, thermomanagement, and EGR support operations.

Similarly to the above structure in Part 4 the cylinder charge air controller design process with high-pressure EGR and exhaust throttling is described. A first-engineering-principle-based, mean-value, nonlinear model of the engine air-path system with HP-EGR valve and throttling downstream the turbine is introduced and validated with test bench measurements. Finally, an LQ servo controller is synthesized which performance is demonstrated with engine dyno measurements.

The fifth part of this thesis summarizes the contribution of the work. The main findings can be found distilled in four thesis points.

ÖSSZEFoglalás


A disszertáció első része összefoglalja a modern dízelmotorok fejlesztői által megoldandó problémákat és a munka motivációit. Ezután a haszonjármű dízelmotor töltötcere rendszerébe épített szelepek lehetséges új funkciói kerülnek kidolgozásra. Vizsgálataik és indoklás után az EGR-ráta helyett a szívótartályi oxigén koncentráció került kijelölésre, mint szabályozandó jellemző. Ezen funkciók optimális megvalósítása adja a következőben ismertetendő fejlesztési folyamat célját. Végül egy részletes irodalom kutatás keretén belül a jelenlegi dízelmotor szabályozási megoldások és károsanyag emisszió keletkezésének mechanizmusára kerül ismertetésre.

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Szelepek a feltöltött dízelmotor töltetcsere rendszerének számos pontján elhelyezhetőek, például a turbina előtt vagy után stb. A különböző elhelyezkedésű beavatkozók hasonlóképpen alkalmazhatók lehetnek a megcélzott lehetséges új funkciók megvalósítására. Habár a motor töltetcsere rendszerében különböző helyen elhelyezett szelepek működtetése különböző hatással lehetnek motorikus paraméterekre például a fogyasztásra, emisszióra stb. Az optimális elhelyezés meghatározása végezt a vizsgált motor részletes modellje került felépítésére GT-Suite környezetben. A simuláció segítségével készült összehasonlítás segítségével egy optimális szelep aktuátor elhelyezés került meghatározásra a motor töltetcsere rendszerében.

A disszertáció harmadik részében az ellennyomás szabályozó funkció kerül bemutatásra. Az első lépésben a szabályozási célok és követelmények kerülnek meghatározásra. Ezután a motor és az aktuátor egy fizikai elvenek alapuló, nemlineáris, szabályozástervezési célú, középérték modellje kerül felírásra és validációra motorfékpadi méresekkel. Ezt követően négy modellalapú szabályozási struktúra (egy LQ servo, egy modell inverzió alapuló előrecsatolással kiegészített LQ servo, egy LPV modell alapú H-végtelen és egy csúszómód szabályozó) megtervezésére és összehasonlítására kerül sor. A szabályozási teljesítmények és a követelményeknek való megfelelés három különböző tesztciklusban került kiértékelésre egy közepes nagyságú haszonjármű motoron. Az első ciklus az üzemi fék kiegészítését, a második a kipufogógáz hőmérséklet menedzsmentet, a harmadik pedig a kipufogógáz visszavezetés támogatásának vizsgálatát célozza.

Az előző rész struktúrájához hasonlóan a negyedik részben egy hengertöltet szabályozó megtervezésére kerül sor magasnyomású EGR szelep és kipufogó oldali, turbina után beépített szelep beavatkozók segítségével. A motor töltetcsere rendszerének magasnyomású, pozíciósabályozott EGR szeleppel és turbina utáni fokozatmentesen állítható pillangószeleppel kiegészített, fizikai elvenek alapuló, nemlineáris, szabályozástervezési célú, középérték modellje kerül felírásra és validációra motorfékpadi mérésekkel. Végezetű egy LQ servo szabályozó került megtervezésre, melynek teljesítménye motorfékpadi mérésekkel került demonstrálásra.

A mű ötödik részében az új tudományos eredmények kerülnek összegzésre négy tézisben.
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NOMENCLATURE

\( \alpha \) Exhaust throttle angular position \( [\text{°}] \)  
\( \alpha_0 \) Fully closed exhaust throttle flap angular position \( [\text{°}] \)  
\( \dot{x} \) The augmented state vector  
\( m_{\text{valve}} \) Measured mass flow rate through the valve \([\text{kg/s}]\)  
\( \eta_t \) Turbocharger efficiency \([-]\)  
\( \eta_{\text{vol}} \) Engine volumetric efficiency \([-]\)  
\( \eta_{\text{vol},\Delta p} \) Engine volumetric efficiency depending on pressure ratio \([-]\)  
\( \eta_{\text{vol},n_e} \) Engine volumetric efficiency depending on engine speed \([-]\)  
\( \hat{u} \) Equivalent control  
\( \kappa \) Adiabatic exponent of air \([-]\)  
\( \lambda \) Air-fuel ratio \([-]\)  
\( \lambda_{cr} \) Sliding surface slope  
\( Q \) Connection rod ratio \([-]\)  
\( R \) The state weighting matrix  
\( x_p \) The control input weighting matrix  
\( x_{\text{ref}} \) The controlled state(s) actual value  
\( \omega_e \) The reference state(s)  
\( \omega_e \) Engine angular speed \([-]\)  
\( \bar{M}_{\text{air}} \) Air average molar mass \([\text{g/mol}]\)
<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>$\rho_{im}$</td>
<td>Intake manifold density $\left[ \frac{kg}{m^3} \right]$</td>
</tr>
<tr>
<td>$\rho_{is}$</td>
<td>Isentropic density of the flowing media $\left[ \frac{kg}{m^3} \right]$</td>
</tr>
<tr>
<td>$\sigma_{air}$</td>
<td>Fresh air mass flow rate $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_c$</td>
<td>Compressor mass flow rate $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_{egr,air}$</td>
<td>Air mass flow from the recirculated gas[kg]</td>
</tr>
<tr>
<td>$\sigma_{egr,eg}$</td>
<td>Exhaust gas mass flow from the recirculated gas[kg]</td>
</tr>
<tr>
<td>$\sigma_{egr}$</td>
<td>Exhaust gas recirculation mass flow rate $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_{ei,air}$</td>
<td>Engine intake fresh air mass flow [kg]</td>
</tr>
<tr>
<td>$\sigma_{ei,eg}$</td>
<td>Engine intake exhaust gas mass flow [kg]</td>
</tr>
<tr>
<td>$\sigma_{ei}$</td>
<td>Mass flow rate into the cylinders $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_{engine}$</td>
<td>Inlet mass flow rate of the engine $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_{eo}$</td>
<td>Engine outflowing mass flow rate $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_{et}$</td>
<td>Mass flow rate through the exhaust throttle $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_{fuel}$</td>
<td>Fuel mass flow rate $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_{in}$</td>
<td>Inflowing massflow $[kg/s]$</td>
</tr>
<tr>
<td>$\sigma_{out}$</td>
<td>Outflowing massflow $[kg/s]$</td>
</tr>
<tr>
<td>$\sigma_{t,red}$</td>
<td>Reduced turbine mass flow $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\sigma_t$</td>
<td>Turbine mass flow rate $\left[ \frac{kg}{s} \right]$</td>
</tr>
<tr>
<td>$\varphi$</td>
<td>The instantaneous crank angle $[deg]$</td>
</tr>
<tr>
<td>$\varphi_{et,act}$</td>
<td>Exhaust throttle flap actual position [-]</td>
</tr>
<tr>
<td>$\varphi_{et,dem}$</td>
<td>Exhaust throttle flap position demand [-]</td>
</tr>
<tr>
<td>$\tilde{x}$</td>
<td>The tracking error</td>
</tr>
<tr>
<td>$A_{eff,v}$</td>
<td>Effective cross sectional area of the valve $[m/s^2]$</td>
</tr>
<tr>
<td>$A_{egr}$</td>
<td>EGR valve geometrical flow area $[m^2]$</td>
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<tr>
<td>$A_{et,0}$</td>
<td>Wide open exhaust throttle area for the linear area model $[m^2]$</td>
</tr>
<tr>
<td>$A_e$</td>
<td>Exhaust throttle geometrical flow area $[m^2]$</td>
</tr>
<tr>
<td>$A_{leak}$</td>
<td>Fully closed exhaust throttle flap leakage area $[m^2]$</td>
</tr>
<tr>
<td>$A_p$</td>
<td>Piston area $[m^2]$</td>
</tr>
<tr>
<td>$A_{R,0}$</td>
<td>Reference cross sectional area of the valve $[m^2]$</td>
</tr>
<tr>
<td>$c$</td>
<td>Constant [-]</td>
</tr>
<tr>
<td>$c_{d,egr}$</td>
<td>EGR valve discharge coefficient [-]</td>
</tr>
<tr>
<td>$c_{d,et}$</td>
<td>Exhaust throttle discharge coefficient [-]</td>
</tr>
<tr>
<td>$c_{D,v}$</td>
<td>Valve discharge coefficient [-]</td>
</tr>
<tr>
<td>$c_p$</td>
<td>Air specific heat at constant pressure $\left[ \frac{J}{kg\cdot K} \right]$</td>
</tr>
<tr>
<td>$c_t$</td>
<td>Turbine mass flow model parameter [-]</td>
</tr>
<tr>
<td>$D$</td>
<td>Bore diameter of the cylinder $[m]$</td>
</tr>
<tr>
<td>$D_{th}$</td>
<td>Exhaust throttle diameter $[m]$</td>
</tr>
<tr>
<td>$EFR$</td>
<td>Engine firing rate $[Hz]$</td>
</tr>
<tr>
<td>$H_l$</td>
<td>Diesel lower heating value $[J/kg]$</td>
</tr>
<tr>
<td>$i$</td>
<td>Number of revolutions per cycle [-]</td>
</tr>
<tr>
<td>$J$</td>
<td>The state weighting matrix</td>
</tr>
<tr>
<td>$k$</td>
<td>Sliding mode gain</td>
</tr>
</tbody>
</table>
\( K_{eo} \) Engine outflowing enthalpy ratio [-]
\( K_{lo} \) Stoichiometric air to fuel ratio [-]
\( K_t \) Turbine temperature drop constant [-]
\( k_t \) Turbine mass flow model parameter [-]
\( l \) Connection rod length [m]
\( m \) Mass [kg]
\( m_{em,\text{air}} \) Air mass in the exhaust manifold [kg]
\( m_{em,\text{eg}} \) Exhaust gas mass in the exhaust manifold [kg]
\( m_{em} \) Total gas mass in the exhaust manifold [kg]
\( M_e \) Engine brake torque [Nm]
\( m_{im,\text{a}} \) Air mass in the intake manifold [kg]
\( m_{im,\text{eg}} \) Exhaust gas mass in the intake manifold [kg]
\( m_{im} \) Total gas mass in the intake manifold [kg]
\( M_{O_2} \) Oxygen molar mass [g/mol]
\( n_e \) Engine speed [1/s]
\( p \) Pressure [Pa]
\( p_{amb} \) Ambient pressure [Pa]
\( p_{c,\text{in}} \) Compressor inlet pressure [Pa]
\( p_{cr} \) Critical pressure [Pa]
\( p_{em,\text{sim},\text{woEGR}} \) Simulated exhaust manifold pressure in the non-EGR case [Pa]
\( p_{em} \) Exhaust manifold pressure [Pa]
\( p_e \) Engine brake power [W]
\( p_{fric} \) Friction mean pressure [Pa]
\( p_{f ric} \) Friction mean pressure [Pa]
\( p_{im,\text{meas},\text{woEGR}} \) Measured intake manifold pressure in the non-EGR case [Pa]
\( p_{im,\text{meas},\text{wEGR}} \) Measured intake manifold pressure with the use of EGR [Pa]
\( p_{im,\text{sim},\text{wEGR}} \) Simulated measured intake manifold pressure with the use of EGR [Pa]
\( p_{im,\text{sim},\text{woEGR}} \) Simulated measured intake manifold pressure in the non-EGR case [Pa]
\( p_{im} \) Intake manifold pressure [Pa]
\( p_i \) Indicated mean pressure [Pa]
\( p_{to} \) Turbine outlet pressure [Pa]
\( q \) Sigmoid slope
\( Q_b \) The heat released by the burned fuel [J]
\( Q_w \) Wall heat losses [J]
\( R \) Specific gas constant \( \left[ \frac{1}{kgK} \right] \)
\( r \) Crank radius [m]
\( R_{air} \) Specific gas constant of air \( \left[ \frac{1}{kgK} \right] \)
\( s \) The sliding surface
s_p Stroke remaining to top dead centre (TDC) at a given crank angle [m]

T Temperature [K]

t Time [s]

T_{amb} Ambient temperature [K]

T_{c,in} Compressor inlet temperature [K]

T_{cycle} Validation cycle length [s]

T_{em} Exhaust manifold temperature [K]

T_{et,out} Exhaust throttle outlet temperature [K]

T_{et} Exhaust throttle actutor time constant [s]

T_{im} Intake manifold temperature [K]

T_{to} Turbine outlet temperature [K]

V Volume [m³]

V_{cyl} Instantaneous cylinder displacement volume above the piston at a given crank angle and is the compression ratio [-]

V_c Compression chamber volume [m³]

V_d Cylinder swept volume [m³]

V_{em} Exhaust manifold volume [m³]

V_{im} Intake manifold volume [m³]

v_{is} Isentropic flow velocity [m/s²]

V_{o} Volume between turbine and exhaust throttle [m³]

w_{em,air} Exhaust manifold air mass fraction [-]

w_{em,air} Exhaust manifold air mass fraction [-]

w_{eo,air} Engine outflowing gas air mass fraction [-]

w_{im,a} Intake manifold air mass fraction [-]

w_{O_2,air} Air oxygen mass fraction [-]

x_{egr} EGR rate [-]

x_{O_2,air} Oxygen volume fraction of fresh air [-]

x_{O_{2,em}} Exhaust manifold oxygen volume fraction [-]

x_{O_{2,co}} Engine outlet gas oxygen volume fraction [-]

x_{O_{2,im}} Intake manifold oxygen volume fraction [-]

x_{O_{2,lo}} Oxygen volume fractions in the exhaust manifold [-]

z Number of engine cylinders [-]

ACRONYMS

AMT Automated Manual Transmission

EGR Exhaust Gas Recirculation

HP-EGR High-Pressure Exhaust Gas Recirculation

xx
LP-EGR  Low-Pressure Exhaust Gas Recirculation
PM  Particulate Matter
DPF  Diesel Particulate Filter
VGT  Variable Geometry Turbine
VNT  Variable Nozzle Turbine
HC  Hydrocarbon
SI  Spark-ignited
CI  Compression-ignited
EDC  Electronic Diesel Controller
SCR  Selective Catalytic Reduction
ICE  Internal Combustion Engine
EBS  Electronic Brake Control System
ECU  Electronic Control Unit
DOE  Design of Experiments
IMEP  Indicated Mean Effective Pressure
IMEP360  Gross indicated Mean Effective Pressure
FMEP  Friction mean effective pressure
PMEP  Pumping mean effective pressure
RMS  Root-mean-square
BSFC  Brake Specific Fuel Consumption
XBR  External Brake Request Interface
BMEP  Brake mean effective pressure
LTC  Low Temperature Combustion
XTC  Exhaust Throttle Controller
PSD  Power Spectral Density
CARE  Control Algebraic Ricatti Equation
LPV  Linear Parameter-varying
UEGO  Universal Exhaust Gas Oxygen
ESC  European Stationery Cycle
MPC  Model Predictive Control
MIMO  Multiple Input Multiple Output
SISO  Single Input Single Output
WHTC  World Harmonized Transient Cycle
HCCI  Homogeneous Charge Compression Ignition
LQG  Linear Quadratic Gaussian Control
OEM  Original Equipment Manufacturer
CAN  Controller Area Network
Part I

INTRODUCTION, MOTIVATION AND RESEARCH AIMS
The key component of a road vehicle is the propulsion system which creates propulsive force from an energy source. Since its invention more than hundred years ago diesel engines became the dominant propulsion system in road vehicles between petrol engines at the early of the nineteenth century. Rudolf Diesel’s invention could initially be used only for stationary applications. It was to be decades before the diesel engine finally “hit the road”, after the solution of difficulties concerning fuel injection. At these early stages, the main driver of the development was motivated by the need for increasing brake and specific power. This led to the increasing of the number of cylinders and especially by diesel engines to the turbocharging. [6]

The air pollution problem became apparent in the 1940s in Los Angeles. It was demonstrated that the smog results mainly from the reaction of the nitrogen oxides and hydrocarbon compounds. It revealed that the internal combustion exhaust gases of the automobiles are the primary sources of the high levels of the $NO_x$ and $HC$ in urban areas. Diesel engines are a significant source of small soot or smoke particles as well as $HC$ and $NO_x$. Therefore, emission standards were introduced first in California in the 1960s followed by regulations worldwide [31]. Since then, legislative emission limitation requirements dominated the process of the diesel engine development.

During the 1970s the crude petroleum price rose rapidly. Internal Combustion Engine (ICE) development efforts turned to increasing the efficiency which needs in many cases contradictory modifications to emission control. At the end of the 1990s the focus of interest turned to the vehicle $CO_2$ emission. Not only for the economical operation and dwindling oil reserves but it became clear that with high probability the increasing level of $CO_2$ and other greenhouse gases in the atmosphere are warming the globe. At the end of the 90s first voluntary agreements were made between the European Commission and the automotive industry then California emerged the first mandatory regulation in 2004. It was followed by similar restrictions worldwide. In 2006 Japan introduced the first heavy-duty vehicle fuel economy standards in the world. [38] These steadily tightening $CO_2$ restrictions parallel with the similarly more and more rigorous emission restrictions gave the fundamental challenges for nowadays diesel engine development and the motivation of this work.

This thesis aims to focus primarily on commercial vehicle diesel engine applications. Of course, the main findings could be applicable on different types of diesel engines, but the engine setup, actuators,
and functions used as a starting point are originated from commercial vehicles. The main difference between the commercial vehicle and the passenger car market that customers are mainly motivated by the achievable economic profit. Therefore, the low vehicle and maintenance cost and reliability are extremely important factors in the competition for the market share of vehicle manufacturers and their suppliers. Moreover, the application of expensive parts and technologies, if they can provide lower operation costs, can be economically feasible due to the high mileage. The legal framework of emission legislation and CO₂ limitations require in most cases contrary interventions into the engine operation (e.g., decreasing NOₓ emission with reduced compression ratio but increasing fuel consumption). Moreover, regulatory requirements are often undesirable from consumer’s point of view (e.g., higher operation and maintenance cost due to aftertreatment systems). The acceptance of the legal criterion while providing more benefits for the users gives an excellent challenge for the researcher and developer moreover a drive of the present research.

The most challenging for diesel engine developers are especially the reduction of PM and nitrogen oxide contents of the exhaust gases. There are two possibilities to achieve this: exhaust gas aftertreatment and reduction of raw emissions. The use of a Diesel Particulate Filter (DPF) to reduce the PM emission of a diesel engine results in higher fuel consumption caused by the generated backpressure and the filter regeneration. Selective Catalytic Reduction (SCR) catalysts are expensive and reach their nominal efficiency only in a limited exhaust gas temperature range. The required urea and its reservoir further increase the costs and reduce the space available for the fuel tanks. It is clearly seen that aftertreatment systems help to agree with legal emission limits but significantly raise the prices and in most cases the fuel consumption. For this reason, the research of the current thesis aims to reduce aftertreatment system size. The other possibility, namely the raw emission reduction can be achieved by controlling the cylinder charge gas quantity and quality called diesel engine air-path management. There are numerous ways to solve it, e.g., Variable Geometry Turbine (VGT) turbochargers, Exhaust Gas Recirculation (EGR), etc. Throttle valves at different locations in the intake and exhaust system and their synergic application involve the possibility of a cost-effective solution. Moreover, with a suitable control, they could also provide additional commercial vehicle specific functions (such as endurance braking with brake blending, Automated Manual Transmission (AMT) support, etc., see detailed in Chapter 2). Due to the above beneficial possible new functionalities of throttle valves in a commercial vehicle diesel engine air-path, this thesis aims to provide an optimal engine air-path setup with throttle valves and new control methods to take advantage of the above mentioned possible new functionalities of the throttle valve in the engine gas-path.
THE AIM OF THIS WORK - ACHIEVABLE NEW ENGINE FUNCTIONS

Commercial vehicle diesel engines which are in the focus of this research need to fulfill several requirements traced from different sources, e.g., legislation, economical operation, safety, etc. Engine air-path management can be an effective tool for the fulfillment of these requirements. For example, engine raw emission is affected strongly by the amount and composition of the cylinder charge in which combustion takes place. Flow area control at different locations of the intake and exhaust system offers a fast intervention so can be used to reduce emission peaks in sudden load changes. It is more and more important in current highly transient emission measurement cycles. Moreover, electromechanical throttle valves are considerably cheaper than the installation and operation of an aftertreatment system. This way, air-path management offers an economical solution. Furthermore, air-path flow control can provide several additional functions by the ability to affect precisely and quickly the engine load, braking power or deceleration speed. It makes the installation of the throttle valve economically more beneficial. This section aims to summarize possible new functions that could be achieved by the appropriate control of the engine air-path. The realization and demonstration of these functions give the aim of this work.

2.1 BACKPRESSURE CONTROL

2.1.1 Brake blending

Decades ago numerous dangerous driving situations and sometimes accidents occurred with commercial vehicles due to brake fade. The brake fade caused by the overheating of friction surfaces of wheel brakes after long-term application, especially while descending downhill with high load, which leads to a reduction of brake power. Since wheel brakes of commercial vehicles are not designed for such continuous operation, new regulations were introduced. Since 1991 buses over a total weight of 5.5 t and trucks over a total weight of 9 t have to be equipped with an additional brake system, which operates independently of the service brake system. It has to be able to keep the vehicle at a fully loaded condition below 30 km/h on a 7 % slope for a distance of at least 6 kilometers [20]. On commercial vehicle diesel engines, a widely used and cost-effective endurance brake is the exhaust brake. These brake retarders are typically designed as
butterfly valves and utilized for generating backpressure for the engine exhaust. The brake power can be increased by creating higher backpressure in the exhaust manifold, but its value is defined by the valvetrain design \([33]\). Consequently, the exhaust manifold pressure has to be limited to avoid valvetrain failure. The overpressure limit is engine specific, so it is an exhaust brake design parameter. Therefore several exhaust throttle designs are required. For example: by means of a pressure regulator valve which can bypass the restrictor valve or flap axle with an offset etc. see, e.g., \([71]\) and \([72]\). To fulfill the requirements as endurance brakes, exhaust throttles on modern commercial vehicle diesel engines have only two states: fully opened and fully closed.

The engine braking power can be calculated based on the following formula:

\[
P_e = M_e \cdot \omega_e = i \cdot p_e \cdot V_d \cdot n_e
\]

Hence the braking torque can be calculated as:

\[
M_e = \frac{i \cdot p_e \cdot V_d \cdot n_e}{2 \cdot \Pi \cdot n_e} = \frac{1}{\Pi \cdot i} \cdot p_e \cdot V_d = c \cdot p_e
\]

As a consequence, the engine braking torque can be derived from the Brake mean effective pressure (BMEP) multiplied by a constant. The BMEP can be distributed into the following to parts:

\[
p_e = p_{fri} + p_i
\]

It means that the negative engine power during endurance braking is proportional with the Friction mean effective pressure (FMEP) and Indicated Mean Effective Pressure (IMEP). The FMEP consist of the driving of auxiliary equipment (water pump, oil pump, etc.), the friction of the piston and cylinder wall, bearing, etc. FMEP is aimed to minimize by engine designers and fundamentally cannot be affected by the engine air-path control. The indicated mean effective pressure consist of the IMEP\(_{360}\) which is approximately equal to zero (because there is no fuel injection into the cylinders during engine braking operation) and the PMEP. PMEP can be affected mainly by the pressure difference between the intake and exhaust strokes. Intake pressure levels can be decreased by throttle valves, but the potential increase in the PMEP is limited. Hence, the control of the throttle flaps in the engine intake is not a proper strategy to realizing the targeted brake blending function. Nevertheless, the exhaust pressure level can be increased more significantly as described above for auxiliary braking applications.

The generation of intermediate pressure levels and tracking pressure demands with a high response would allow the use of exhaust throttles in the following possible new application fields suggested
by this thesis. With appropriate control of the engine exhaust manifold pressure, the engine brake power could be adjusted arbitrarily. In this way, the backpressure controlled exhaust brake operation could substitute a service brake at moderate brake torque demands. This is called brake blending. So the exhaust throttle could be integrated into the service brake operation and can increase the lifetime of brake pads.

Electronic Brake Control System (EBS) units control the vehicle brake system on the basis of the demanded deceleration request. These request can be rooted from the driver through the brake pedal or by other sources (such as the longitudinal controller of automated driving functions, etc.) through an External Brake Request Interface (XBR). After arbitration based on the priorities of requests from different sources, the EBS distributes the deceleration demand between the different braking actuators such as wheel brakes, engine brakes, retarders, etc. Based on the deceleration demand a brake power (torque) demand is calculated and sent out to the actuators by the EBS. As a conclusion the chose of the exhaust backpressure as a controlled variable for realizing the aimed brake blending function is the suitable solution. The dynamic of the pressure buildup in the engine exhaust manifold is also an important question because the braking torque buildup at the vehicle wheels should be as fast as possible. This depends on the actuator dynamics (how quickly the flap can reach the target position from the fully opened position) and on the volume of the piping between the exhaust valve and exhaust throttle valve. There are two possible placements can be considered for exhaust throttles (see the investigation in Chapter 6): upstream the turbine and downstream the turbine. The flap valve installed upstream of the turbine provides the aimed minimum volume. However, gas temperature and pressure are much higher in the exhaust manifold than downstream the turbine. It can be challenging for the material choice and lifetime of the flap valve, heat can also be dangerous for the actuator (extra cooling and material need), and high-pressure levels would need complicated sealings. In the commercial vehicle market the cost, lifetime and reliability are vital issues. Therefore, the downstream the turbine location was chosen as a suitable arrangement for backpressure generation.

2.1.2 AMT support

Another benefit of an enhanced engine deceleration through the above described exhaust backpressure control could be utilized during the upshifting of gears with a commercial vehicle AMT gearbox. Before the clutch engagement, the engine speed needs to be synchronized to the gearbox input shaft speed. The time demand for this synchronization process, namely the engine deceleration, could be decreased with
suitable backpressure generation in the exhaust manifold. This results in quicker vehicle acceleration. A similar application is described in [68] with an engine compression release brake.

During the upshifting process with an AMT transmission, its logic aims to decelerate the gearbox input shaft speed after the driveline engagement to be able to synchronize the next gear. It usually can be done by cutting off the fuel from the engine. In this case, the engine friction torque and the torque results from the PMEP decrease the engine speed (see detailed explanation above). The PMEP, however, is small because the intake and exhaust manifold pressure levels are approximately equal to the ambient pressure. With the actuation of the exhaust throttle the exhaust backpressure and as a result the engine PMEP can be increased significantly, and it can be converted to faster engine crankshaft deceleration. The maximum backpressure needs to be limited of course, and a quick pressure buildup is necessary due to the limited time available. Fig. 2.1 shows an upshifting process measured with a heavy-duty truck equipped with an AMT. In the first subplot, the engine speed and actual engine percent torque (which is proportional to the fueling rate) can be seen. The second subplot depicts that after selecting the next gear, the driveline will be engaged and after it the gearbox Electronic Control Unit (ECU) cuts off the fuel to synchronize the next gear. After it, the next gear was actuated, and the driveline was disengaged. The third subplot shows the pressure in the exhaust manifold which is approximately 1 bar due to there was no additional throttling in the exhaust system. The whole engine deceleration process needed 1.12 second in this case. In Fig. 2.2 the same upshifting operation is depicted with the same truck but while the driveline was engaged exhaust backpressure was generated with the exhaust throttle. The time demand of the engine deceleration reduced to 0.62 second which means 45% reduction of the time demand. It can be seen that the AMT upshifting process can be enhanced effectively by backpressure generation and this possible new function can also be rooted back to backpressure control. Upstream the turbine throttle installation would be more beneficial (due to the faster dynamics potential) even for this application but due to the adisadvantages as mentioned above the downstream of the turbine location seems to be a suitable compromise.
The two ways which can be utilized in order to produce more environmental friendly engines are exhaust gas aftertreatment (i.e., SCR, DPF, etc.) and the restriction of pollutant formation during the combustion process (i.e., exhaust gas recirculation). To meet new requirements in the future, the use of both possible methods will probably be unavoidable [73].

To maintain high efficiency in engine exhaust aftertreatment systems, it is desirable to avoid cooling by cold exhaust gases in low-load engine conditions and also rapid heating. For details see [53]. In this
case, a controlled exhaust backpressure could be used for exhaust gas thermomanagement.

The exhaust gas temperature depends mainly on the air-to-fuel ratio which is proportional to the engine load. Therefore, the exhaust temperature can be controlled with the engine load. However, at engine idling where the low temperature of the exhaust gases causes the low efficiency of the aftertreatment system a load torque cannot be applied on the engine crankshaft. In these cases, the artificial increase of the PMEP can be a solution which can be done by exhaust throttling. Additionally, the exhaust throttling reduces the volumetric efficiency of the engine (see Eq. 8.8) consequently the air-to-fuel ratio. As a conclusion, the exhaust pressure control can be an appropriate way to the exhaust thermomanagement.

Finally, we can summarize that the possible new functions of the brake blending, AMT support, and exhaust gas thermomanagement can be optimally realized as exhaust manifold pressure control with a flap actuator downstream to the turbine. The complete controller design and demonstration process for these novel functions are described in Part iii.

2.2 CYLINDER CHARGE CONTROL

Current and next generation emission standards (e.g., Euro 6 and US EPA 13) include significant limitations. The most challenging for engine developers are especially the reduction of PM and nitrogen oxide contents of the exhaust gases. Basically, there are two possibilities to achieve this: exhaust gas aftertreatment and a decrease in raw emissions. The use of a DPF to reduce the PM emission of a diesel engine results in higher fuel consumption caused by the generated backpressure and the filter regeneration. SCR catalysts are expensive and reach their nominal efficiency only in a limited exhaust gas temperature range. The required urea and its reservoir further increase the costs and reduce the space available for the fuel tanks. Due to the above disadvantages, diesel engine development today aims to reduce aftertreatment system size [43].

Exhaust gas recirculation is an effective tool in the limitation of the nitrogen oxides and in combustion control see, e.g., Low Temperature Combustion (LTC). However, to significantly reduce $NO_x$ emission and especially for the realization of advanced combustion processes the amount of the recirculated exhaust gases need to be adjusted in a wide range. In High-Pressure Exhaust Gas Recirculation (HP-EGR) systems the back-flowing exhaust gas mass flow rate is driven by the pressure difference between the exhaust and the intake manifold. The resulting conditions depend mainly on the turbocharger and its cooperation with the engine so the dilutent gas mass flow is limited. However, with the application of throttling in the engine air-
path, the pressure drop on the EGR duct could be adjusted, and EGR rate could be increased. Additional nitrogen-oxide emission reduction and combustion method improvement would also be possible. An exhaust backpressure controller with a variable geometry turbine for turbocharged Spark-ignited (SI) engines was presented in [23] for similar applications.

Figure 2.3 depicts the measured pressure differences on the test engine with closed EGR valve. As it reveals in large part of the engine operation map (at low speed and high load region), the difference of the average pressure levels is negative.

![Figure 2.3: Exhaust and intake manifold pressure differences in the operation map of the investigated engine](image)

Fig. 2.4 shows the measured EGR rates in the same engine. As it can be seen the EGR mass flow in those regions of the engine map where the pressure difference is negative is very limited and not enough for significant reduction of the NO\textsubscript{x} emission. However, emission test cycles use most frequently this part of the engine map. Please note that the small amount of the EGR backflow can be evolved due to the extensive pressure oscillation both in the intake and exhaust manifold so despite the fact that the difference of the average pressure is negative for short periods positive pressure ratios can be evolved thanks to the pressure oscillation. Moreover, in the EGR duct, a check valve was installed to avoid charge air flow to the exhaust side.

To be able to reach arbitrarily high EGR mass flow rates the increasing of the exhaust manifold pressure or lowering of the intake manifold pressure gives the opportunity. It can be realized by throttling at four different locations of the intake and exhaust system: upstream and downstream the compressor and upstream and downstream the turbine. However, the effect of throttling at different locations are not equal to the engine performance measures and emission such as BSFC, PM emission, etc.
Figure 2.4: EGR rates in the operation map of the investigated engine

To objectively evaluate and compare the different engine layouts simulation a study was carried out in Chapter 6.

2.2.1 Intake manifold oxygen concentration as controlled variable

To handle \( NO_x \) and PM formation during the combustion process the precise adjustment of the cylinder charge composition is an effective tool and can be achieved by exhaust gas recirculation. Current EGR systems concentrate mostly on controlling EGR rates thereby ignoring the quality of the recirculated exhaust gases [36]. EGR replace or add to the fresh air amount and the recirculated \( CO_2 \) and water vapor will increase the heat capacity of the cylinder charge. Thanks to the dilution effect of the EGR the oxygen concentration in the cylinder charge also will decrease. The ignition in a non-premixed flame occurs where the local equivalence ratio is stoichiometric or slightly above it. The fuel spray must penetrate more and occupy larger volume to achieve ignition conditions in a diluted (by EGR) cylinder charge compared with fresh air alone. Due to the larger heat capacity (larger volume) of the flame region, the local temperature will be lower. More detailed explanation and investigation of the EGR effect can be found in [45–48].

Moreover, with higher EGR rates the realization of different types of LTC will be possible. For details see [2] and [64]. Also, synergies of advanced combustion systems by reducing low load \( NO_x \) are revealed in [39].

The EGR rate can be calculated based on the following formula:

\[
x_{egr} = \frac{\sigma_{egr}}{\sigma_{air} + \sigma_{egr}} = \frac{x_{O_2,im} - x_{O_2,air}}{x_{O_2,lo} - x_{O_2,air}}
\]  \ (2.4)
As it was revealed in [45–48] the effect of the EGR is based on the dilution effect of the exhaust gas components (water and carbon dioxide) on the combustion process. However, a high EGR rate not necessarily means a high amount of water and carbon-dioxide content in the cylinder charge because the exhaust gas of the diesel engines usually contains large quantities of fresh air due to the lean mixture combustion, especially at part loads. It can be seen clearly from the definition of the air-fuel ratio below:

\[
\lambda = \frac{\sigma_{\text{air}}}{K_{L0} \cdot \sigma_{\text{fuel}}} = \frac{1}{1 + \frac{x_{O_2,\text{im}}}{x_{O_2,\text{air}}}} \quad (2.5)
\]

As it reveals at high lambda values, the exhaust gas contains mostly fresh air, and consequently, its oxygen concentration converges to those of the fresh air. The following formula gives the connection between the EGR rate, lambda value and intake manifold oxygen volume fraction based on [30]:

\[
x_{O_2,\text{im}} = \frac{M_{\text{air}}}{M_{O_2}} \cdot \frac{\lambda \cdot K_{L0} + 1 - x_{\text{egr}} - K_{L0} \cdot x_{\text{egr}}}{1 + \lambda \cdot K_{L0}} \quad (2.6)
\]

It can be derived that the intake manifold oxygen concentration gives an objective measure of how diluted is the cylinder charge namely how much is the concentration of the water damp and carbon dioxide in it. Fig. 2.5 shows that a given EGR rate the intake manifold oxygen concentration values and consequently the dilutent water and carbon-dioxide concentrations can alter in a wide range in function of the lambda values.

![Figure 2.5: Intake manifold oxygen concentration in function of the air-fuel ratio](image-url)

A strong correlation can be observed between lambda, intake manifold oxygen concentration and nitrogen-oxide emission for which an
example is depicted in Fig. 2.6. The specific values are highly depending on the engine design (e.g., combustion chamber shape, etc.), fuel injection and combustion process but the quality-related characteristic is identical for every diesel engine. Hence, a for a given engine a simple approximation model can be made based on the measurement data similarly to [30] and based on the model, and with an intake manifold oxygen concentration controller the indirect control of the \( NO_x \) emission is possible. Advantages of this approach were demonstrated over the traditional EGR rate based control methods in [11] and [10] with an oxygen concentration observer and closed-loop control.

![Figure 2.6: Nitrogen oxide concentration in function of the air-fuel ratio and intake manifold oxygen concentration [10]](image)

Considering the evolution of the development, requirements and legislative environment of the commercial vehicle diesel engines which was introduced above this thesis aims to design, implement and test air-path control systems for commercial vehicle diesel engines. The present research targets the design of such control solutions that are able to utilize the possible new applications achievable with flap valves installed in the engine air-path listed in this Chapter while using the individual flap valve actuators in the air-path. For this reason, first of all, a detailed model of the test engine need to be developed and validated to be able to analyse the effect of the flap valves at different locations of the air-path and to identify possible synergies and the optimal engine setup. Based on the results simplified, control-oriented models can be designed in state-space form as a basis for the controller synthesis. By using the models, a complete engine air-path control solution had to be developed which can realize the possible new engine functions achievable by the air-path management. Moreover, this work aims to test the designed new features on an engine dynamometer to present their performance.
3.1 DIESEL ENGINE CONTROL

Control systems are inherently essential parts of diesel engines for idle speed control and maximum speed limitation. Of course, the working of these controllers were based on mechanical principles, e.g., flyweight in a Bosch RQ governor. However, in the last decade vehicles tend to incorporate an even greater extent of their part value in electric and electronic components, and this trend seems to continue steadily. Numerous devices help to decrease fuel consumption and emission, increase safety, driveability and comfort. With the increasingly powerful and of course more complex electronic systems, there will be the opportunity to replace mechanical functions with them. In this context the diesel engine control system, that is in the focus of this work, is a part of the whole truck control structure among others like EBS, traction or steering control. Electronic control systems on engines were introduced on a larger scale on SI engines for the control of three-way catalytic converters. Compression-ignited (CI) engines were less advanced in the utilization of electronic control systems. However, it is changed, and they help substantially improve the performance of the diesel engine. This trend started with the advent of the electronically controlled injection systems and continued in nowadays diesel emission control systems which are in the focus of this work.

The torque output of a diesel engine is controlled by changing the air/fuel ratio in the combustion chamber. Therefore, the three-way catalytic converter system which is successfully used in SI engines is not possible to use. Moreover, in a modern diesel engine, the turbocharger introduce an additional feedback path considerably complicating the dynamic behavior of the entire engine system. The primary objective for electronic Diesel-engine control-systems is to provide the required engine torque while minimizing fuel consumption and complying with exhaust-gas emissions and noise level regulations. As an intervention, there are three main paths which have to be considered: fuel, air and recirculated exhaust gas. The mixture formation is dominantly done by unit pump or common-rail systems that ensure the flexibility of the injection with several manipulated variables (e.g., rail pressure, start of injection, etc.). The air-path is dominated by the turbocharger which is controlled typically by wastegate or Variable Nozzle Turbine (VNT) actuators. The application of VNT turbines in the commercial vehicle sector was delayed due to its cost
and reliability issues. There are three main methods to control the amount of recirculated exhaust gases: internal EGR with a variable valve system, high and low-pressure external recirculation system. In case of an external EGR, the recirculated rate can be controlled by the opening area of the EGR valve. Moreover, throttle valves at different placements in the engine intake and exhaust system offer additional possible control inputs which are of particular interest in this thesis. Since almost all control input in the diesel engine control system has a contradictory effect on control targets (high efficiency, low NO<sub>x</sub> and PM) finding an optimum is a challenge for the control engineer. About the diesel engine control problem the reader can see a detailed overview in [27].

Typically, an ECU includes standard microcontroller hardware (process interfaces, RAM/ROM, CPU, etc.) and at least one additional piece of equipment, which is often designated as a time processing unit (TPU). It synchronizes the time-based signals of the microcontroller with the cycle based operation of the engine. The Electronic Diesel Controller (EDC) receives the targeted torque signal from the driver or any other vehicle ECU connected via CAN (e.g., transmission control) of course based on appropriate priority rules. Sensor signals (e.g., engine speed sensor, boost pressure sensor) providing inputs for the control logic are read with the engine ECU’s A/D converter or by smart sensors equipped with CAN. Actuators can be governed via CAN or analog signals and with the individual power stages of the EDC which is common in the case of common rail injectors. A more detailed description of engine control units can be found in [36].

The current overall status of the diesel engine control technology is well summarized in [17]. The most widespread solution utilizes lookup tables and PID control. More advanced control techniques show clear advantages but their realization encounter barriers due to the high complexity and computational demand. Model-based controller design methods offer reduced calibration effort and need less costly testing. The modern control techniques are usually based on state-space representation. A good overview of control theory for linear systems is given in [8, 42]. For general nonlinear systems a comprehensive discussion is made in [9, 28, 37, 41]. The theory of the most popular control techniques is described in the following textbooks: optimal control [14], robust control and linear parameter varying approaches [61, 80], sliding mode control [67] and model predictive control [15, 55, 66]. Several control techniques were successfully used in different type of engine control problems see as examples [50] for PID control, [25] for LQ control, [52] for H-infinity control, [26] for Linear Parameter-varying (LPV) control and [3] for model predictive control.
As it was described above one of the main goals of diesel engine control is to reduce the engine emission. To be able to reach this goal the proper understanding of emission formulation mechanism is unavoidable. This section aims to summarize the main emission components and their formulation methods as well as contradictory requirements of their limitation. In ideal combustion of diesel fuel and air, the exhaust gas would include only water, carbon dioxide, nitrogen, and oxygen. However, at real conditions air pollutant materials arises. The main categories of air pollutants released into the atmosphere include carbon monoxide (CO), organic compounds which are unburned or partially burned Hydrocarbon (HC), PM and oxides of nitrogen (nitric oxide, NO, and small amounts of nitrogen dioxide, NO₂, collectively known as NOₓ). Diesel engines mostly run on lean mixture conditions, therefore, they are not a significant source of carbon monoxide. Moreover, CO gas content of the raw exhaust gases can be easily decreased by a relatively cheap oxidation catalyst below required levels. For similar reason, diesel exhaust gas hydrocarbon concentrations are lower by about a factor of 5 than typical Otto engine levels, and oxidation catalyst is also useful to regulate HC emission. [31] Hence, the most challenging is to control emission of nitrogen oxides and particulate matters in the diesel engine exhaust gases. Therefore, the formation processes will be briefly introduced in the next sections, and the effect of air-path parameters will be explained which is in the interest of this research work.

3.2.1 Nitrogen Oxides

Nitrogen oxide (NO), and nitrogen dioxide (NO₂) are usually grouped together as NOₓ emission, however, the combustion process mainly produces nitrogen-oxides (approximately 60-90%) and little NO₂ in the combustion chamber. As it was described above only NO₂ is relevant for air hygiene as a pollutant input. Smog results from the reaction of NO₂ and hydrocarbons, NO₂ irritates mucous membranes and is caustic when combined with moisture (acid rain)). In the air, NO oxidizes to NO₂, and oxidation catalytic converters in the engine exhaust systematically produce NO₂ from NO. [53] There are five main ways of NOₓ production during the combustion: the thermal or Zeldovich-mechanism, prompt or Fenimore-mechanism, NOₓ via N₂O, NOₓ via NNH and by the fuel-bounded nitrogen. For a detailed description of the different mechanisms of NOₓ formation see [77]. The most significant amount of nitrogen oxides in diesel engines is produced along the Zeldovich mechanism, therefore NOₓ produc-
Control input | Result on NO\textsubscript{x} formation
--- | ---
early start of injection | ↑
high rail pressure | ↑
higher boost pressure | ↑
smaller VNT area | ↑
increased EGR | ↓

Table 3.1: Tendencies in the influence of control inputs on NO\textsubscript{x} formation

O + N\textsubscript{2} \xrightleftharpoons[k_{1}]{_{\text{\textbullet}}} NO + N \hspace{1cm} (3.1)

k\textsubscript{1} = 1.8 \cdot 10^{14} \exp\left(-318kJ \cdot mol^{-1} / (RT)\right) cm^{3} / (mol \cdot s)

N + O\textsubscript{2} \xrightarrow[k_{2}]{_{\text{\textbullet}}} NO + O \hspace{1cm} (3.2)

k\textsubscript{2} = 6.4 \cdot 10^{9}T \exp\left(-26kJ \cdot mol^{-1} / (RT)\right) cm^{3} / (mol \cdot s)

N + OH \xrightarrow[k_{3}]{_{\text{\textbullet}}} NO + H \hspace{1cm} (3.3)

k\textsubscript{3} = 3.8 \cdot 10^{13}cm^{3} / (mol \cdot s).

It can be concluded that the thermal nitrogen oxide formation needs the following conditions: high temperature (T>1900 K, the activation energy of its first reaction is very high) and high oxygen concentration. Therefore, arrangements for lower cylinder temperature and lower oxygen concentration are effective in the NO\textsubscript{x} concentration reduction. For instance these can be lower compression ratio, retarded fuel injection, boost pressure reduction and EGR, etc. The effects of different control inputs available for a typical electronic diesel engine controller on NO\textsubscript{x} formation are summarized in 3.1.

### 3.2.2 Particulate Matter

A vehicle’s particulate matter emissions are the total mass of solids and attached volatile or soluble constituents. The typical composition of particulate matter with a standard oxidation catalyst includes approximately 75% soot, 13% lubrication oil, 5% fuel, 4% water and 3% sulfate. As it reveals the particulates mainly consist of soot, i.e., of elementary carbon. Organic compounds composed of unburned hydrocarbons that may stem from the lubricating oil or the fuel itself constitute the second largest fraction. The dew point of many hydrocarbons is fallen below under the aforementioned conditions for particulate matter sampling and weighing: the compounds condense
Control input | Result on PM formation
---|---
early start of injection | ↓
high rail pressure | ↓
higher boost pressure | ↓
smaller VNT area | ↓
increased EGR | ↑

Table 3.2: Tendencies in the influence of control inputs on PM formation [27]

and bond to the solid cores [53]. Sulfate stems from the sulfate content of the diesel fuel and lubrication oil. The above-described PM component composition of diesel exhaust gases highly depends on engine operation. For example at high loads much higher elementary carbon fraction can be observed. As a contrary at part loads, the quantity of hydrocarbons can increase. As it can be seen, the highest amount of particulate matter emission forms as soot. There is two main hypothesis for the soot production: elementary carbon hypothesis and polycyclic hypothesis. According to both theories of formation, primary particulates with diameters of less than 10 nm form first. These primary particulates agglomerate into the actual soot particulates. The bonding of highly mobile primary particulates causes the agglomerates to grow very rapidly at first. However, as the concentration of primary particulates decreases and the mobility of larger agglomerates diminishes, their growth in size decreases and a typical agglomerate size distribution appears. The size distribution is quite uniform even for different engines and usually a log-normal distribution with a value of approximately 80–100 nm. A more detailed explanation about the soot formation mechanisms can be find in [31, 77].

The majority of the formed soot oxidizes still in the combustion chamber if the temperature is high enough (above 1000 K) and a high amount of oxygen is available. As it can be seen the PM formation during the combustion process is mostly depending on the local oxygen availability in the combustion chamber and correlates well with the combustion air-to-fuel ratio. Therefore, the most typical method for reducing soot emissions is the improvement of the mixture formation so as to prevent the local over-rich portion of in-cylinder mixture, or the acceleration of the re-oxidation of the formed soot in high temperature conditions. This means, e.g., higher rail pressures, earlier start of injection, higher boost pressure, etc. 3.2 summarizes the effect of possible control inputs of electronic diesel control systems on PM formation.

As it reveals the reduction of PM and NOx has typically contradictory requirements in diesel engine control inputs which makes the diesel engine emission control especially challenging. Recent studies revealed relations between combustion temperature, equivalence ra-
tio, PM, and NO\textsubscript{x} formation, see, e.g., [2]. It shows the possibilities of smokeless and NO\textsubscript{x}-less combustion processes by using a large amount of cooled EGR under a near stoichiometric and even in a rich operating condition as shown in Fig. 3.1.

Figure 3.1: PM and NO\textsubscript{x} formation in function of temperature and equivalence ratio [2]
The thesis consists of five parts (including the Introduction). Each part begins with a motivation that describes the central problem statement and aim of the corresponding section. The parts are finished with a summary where the conclusions are listed distilled. The layout of the dissertation and the main scientific contributions are described below.

Part ii aims to reveal the co-operation and synergy effect of the flap valves installed in different locations of the engine air-path (e.g., upstream the turbine, downstream the turbine, etc.). To reach this goal as a first step a detailed model of the investigated engine was designed and validated. Thereafter, the individual and synergistic effect of the throttle valves installed at different locations of the intake and exhaust system of the engine was investigated on engine measures such as torque, fuel consumption, emission. Based on the analysis, the optimal engine air-path flap actuator setup was identified.

In Part iii an exhaust manifold pressure control was worked out. As a first step the system model was described which is based on first engineering principles. The model validation was presented with engine dynamometer measurements. After the definition of the control aims, a backpressure control structure was designed rooted from the above-defined state-space model which fulfills the requirements. Finally, the controller performance was demonstrated with engine dynamometer measurements in different use cases.

Part iv introduces the whole design process of the novel control structure of exhaust gas recirculation with the control inputs of the HP-EGR valve and exhaust throttling with a performance measure of the intake manifold oxygen concentration. The design process follows the steps used in the previous part for the backpressure controller: after the definition of a state-space system model based on first engineering principles and presentation of model accuracy a controller was synthesized which is able to track intake manifold oxygen concentration demands in a wide range. At last, controller performance was illustrated with engine dynamometer measurements.

Part v contains the final conclusions, proposed theses statements, directions for future research and summarizes the contribution of this thesis.
Part II

OPTIMAL AIR-PATH ACTUATOR SETUP
5

DETAILED DIESEL ENGINE MODEL
DEVELOPMENT AND VALIDATION

5.1 INTRODUCTION

The role of a detailed system model in modern diesel engine research has become increasingly unavoidable in the last few decades, especially in analysis, optimization, and control related tasks. The most important motive is the more and more rigorous emission standards such as Euro VI which introduced significant limitations, especially in case of nitrogen oxides and soot emission. Another challenging aspect is the reduction of the fuel consumption, which is motivated by the constantly increasing energy prices. It has high priority in case of commercial vehicles due to the substantial requirement of economical operation. In addition to these, the drivers prefer fast transient throttle response, which can’t be neglected. Additionally, non-stationary engine performance has also vital importance on emission due to highly transient test cycles in current regulations [63]. A detailed description of the above briefly listed motivations can be found in Part i. Fulfillment of engine requirements in the presence of the above challenges would be impossible without the availability of detailed engine models that can grant cheap, fast, infinite analysis platform for engine development instead of the costly, complicated and inflexible measurements.

A model is nothing more than an imitation of real processes. By realization, models can be separated into several classes, i.e., empirical, stochastic, deterministic, lumped parameter, etc. [29]. The suitable model class can be chosen based on the intended target. Engine models have a wide range of possible usage. Detailed models can be used for various effect analyses which can be a powerful tool in engine design and optimization. It can support modifications in the engine design, new actuator or booster tuning, combustion refinement, charge exchange process and air-path system configuration, etc. The possible applications depend of course on the chosen modeling principles. This way the usage of engine models can spare many expensive and time-consuming engine test bench measurement. Additionally hardly- or unmeasurable variables can be acquired. Another benefit can be utilized from the extensive, fast and reproducible simulations during optimization processes. These methods require results from numerous cases with various parameter values which would need an unrealizable extent of measured parts and measurement number see, i.e., Design of Experiments (DOE) [36]. Engine models can be run
not only as a single simulation but can be coupled with other models. This way the boundary conditions can be exchanged between the models. It can be especially useful in transient cases in which results can change quickly. Another benefit is that it is possible to couple different model classes or structures even if they are prepared by different persons and software, e.g., see [5].

In Chapter 2 several possible new engine function was listed which could be achieved by the interaction of flap valves installed in the engine air-path (e.g., EGR valve, charge air throttle valve, etc). For this purpose, an accurate tracking position controlled throttles are needed with intermediate positions (for examples see [1]). Although, there are numerous possible location for these valves in the engine intake, exhaust or EGR-path that seems to have a similar effect on the different measures of the engine operation such as boost pressure, EGR mass flow rate, cylinder charge, etc. The use of flap valves at each possible locations is not feasible due to high cost and of course due to duplicated effects in some cases. Therefore, the identification of the likely most beneficial layout (implying cost, fuel consumption, emission, transient response engine measures) is necessary. The targeted engine air-path setup which will be covered and investigated in this thesis is a direct injected, common-rail, turbocharged diesel engine equipped with a bypass controlled one-stage centripetal turbine, one-stage centrifugal compressor and high-pressure exhaust gas recirculation circuit with hot side EGR valve. For such an engine setup there are two possible location for the flap valves in the intake (upstream the compressor and downstream the compressor) and two possible locations for the flap valves in the exhaust (upstream the turbine and downstream the turbine). The effect of the hot-side and cold-side EGR valve does not aim to cover in this research. This part of this research seeks to identify the optimal engine flap valve setup considering the effects and synergies of flap valves at different locations in the air-path listed above. The first step towards the identification of the optimal engine setup which will be used as a basis for the further work is the development and validation of a detailed engine model. The model was implemented in GT-Suite environment which is a commonly used tool in engine research and development. This work is presented in Chapter 5. With the resulted model a detailed analysis of the actuation of the flap valves was carried out focusing on several different measures of the engine operation (e.g., boost pressure, NOX emission, IMEP, etc.) in Chapter 6. Based on the results the optimal engine setup was identified, which will serve as a basis for the work in the next parts of this thesis.
5.2 Test Engine and Measurement Setup

In this section, the test engine will be introduced that was used for modeling, identification, validation, analysis, controller development and testing throughout this thesis. The engine was installed on an engine dynamometer where all the operating parameters of the engine relevant for the system model were measurable. The modeled engine is a turbocharged and intercooled, medium duty commercial vehicle, common rail diesel engine. The emission level of the test engine is Euro 3. The main parameters are given in 5.1.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore [mm]</td>
<td>102</td>
</tr>
<tr>
<td>Stroke [mm]</td>
<td>120</td>
</tr>
<tr>
<td>Number of cylinders</td>
<td>4</td>
</tr>
<tr>
<td>Engine displacement [cm³]</td>
<td>3922</td>
</tr>
<tr>
<td>Number of valves</td>
<td>4</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17.3</td>
</tr>
<tr>
<td>Rated effective torque [Nm]</td>
<td>600</td>
</tr>
<tr>
<td>Rated effective power [kW]</td>
<td>125</td>
</tr>
</tbody>
</table>

Table 5.1: Basic engine parameters

In emission measurement processes where highly transient cycles are prescribed (i.e., World Harmonized Transient Cycles [United Nations Global Technical Regulation No. 4; 25 January 2007]), a remarkable amount of the total emission is accumulated during load steps, so the handle of these emission peaks plays an important role. To ensure a quick intervention, the engine is equipped with a high-pressure exhaust gas recirculation loop to provide the shortest connection between the intake and exhaust manifolds. Please note that this way of exhaust gas recirculation is only considered in this research. In the EGR duct, a cooler was installed to minimize the temperature of the in-cylinder gases which is essential for the maximum cylinder volumetric efficiency and minimum charge gas temperature.
The investigated engine is installed to an eddy current dynamometer. The accelerator pedal is actuated by a linear motor. The fuel consumption is measured gravimetrically. In the relevant locations of the intake and exhaust system pressure and temperature sensors are installed to ensure the possibility of the identification and validation of each sub-system: upstream the compressor, downstream the compressor but upstream the intercooler and in the intake manifold. On the exhaust side sensors are installed in the exhaust manifold, downstream the turbine but upstream the exhaust throttle and downstream the exhaust throttle respectively. In addition to the above mentioned standard pressure sensors in the intake and exhaust manifold high dynamic low pressure indicating sensors were also used to record the pressure waves during the charge exchange process. The measured low-pressure indicated pressures can also be used during the EGR duct identification process. Furthermore, in each cylinder an indicating sensor was installed as well. With the measured cylinder pressure signals, the characterization of the combustion process can be calculated, e.g., combustion temperatures, burn rates, etc. The instantaneous turbocharger rotor speed was measured by an optical (laser) sensor. Both the exhaust and intake manifolds broadband lambda sensors were used to measure the air-fuel ratio and calculate the exhaust gas recirculation rate. The intake manifold oxygen concentration was measured with a Universal Exhaust Gas Oxygen (UEGO) sensor the signal of which was corrected due to the boost pressure change. Fig. 5.2 shows the engine installed on the dynamometer in the test cell. A detailed description of the instrumentation the reader can find in Appendix A.
5.3 Model Description

The engine model was set up in Gamma Technologies GT-ISE simulation environment, in the GT-Power engine performance and acoustics application. The model building and validation process introduced below was published in [P2 and P4]. The accuracy of a wave action engine model depends heavily on the discretization of the intake and exhaust domains. To achieve high accuracy level, the complete intake and exhaust system (pipes, intercooler, manifolds, ports, etc.) were modeled in three-dimensional CAD software and then the 3D CAD model was transformed to 1D GT-Power flow objects by the GEM3D tool of GT-Suite. In this way, the intake and exhaust system could be converted into one-dimensional form as accurate as possible. The 3D model of the intake and exhaust ports can be shown in Fig. 5.3.

Figure 5.3: 3D CAD models of the engine intake and exhaust channels

The 1D flow domain discretization length was chosen to the recommended 0.4xbore diameter, namely 40 mm at the intake side, and 0.55xbore diameter, namely 55 mm at the exhaust side. The difference is due to the higher speed of sound in the exhaust system. The intercooler was decomposed as straight parallel pipes. The outlet air
temperature and the pressure drop were adjusted to the measurements. It has an essential influence on the volumetric efficiency (by intake air temperature) and also to the transients (by its volume). In the EGR loop, a check valve was applied downstream the EGR cooler. The above-described method of the one-dimensional intake and exhaust system modeling provides realistic flow system volumes, cross-sectional areas, and pipe lengths.

Valve lift profiles were measured on both intake and exhaust side with the help of a dial indicator. To be able to predict the mass flows accurately in and out of the cylinder the valve discharge coefficients were measured on a test bench, where the whole head along with the cylinder was attached to a Roots-blower. Between the blower and the cylinder, a tank with eligible volume was placed to damp the pressure oscillation created by the blower. The pressure difference between the intake manifold inlet and the cylinder was measured with a manometer. The mass flow was recorded with an orifice plate. Each valve lift from 1 mm to 10 mm was measured at different mass flow rates, and the discharge coefficients were finally calculated using the following formula:

\[
\dot{m}_v = A_{\text{eff,p}} \cdot \rho_{\text{is}} \cdot v_{\text{is}} = c_{D,p} \cdot A_{R,p} \cdot \rho_{\text{is}} \cdot v_{\text{is}}
\]

(5.1)

from which the discharge coefficient reads as follows:

\[
c_{D,v} = \frac{\dot{m}_v}{A_{R,p} \cdot \rho_{\text{is}} \cdot v_{\text{is}}}
\]

(5.2)

The resulted discharge coefficient values for the intake side are shown in Fig. 5.4.

![Figure 5.4: Intake valve discharge coefficient](image-url)
The turbo-compressor and its turbine were modeled with stationary performance maps. The compressor performance map can be seen in Fig. 5.5.

![Compressor performance map](image)

**Figure 5.5: Compressor performance map**

The turbocharger is wastegate controlled actuated by a membrane chamber. In case of higher engine speeds, the boost pressure does not exceed the desired value (the turbine will be bypassed). This boost pressure limitation was implemented in the model as a boost pressure sensor and a PID controller. After completing the intake and exhaust system, and turbocharger modeling the appropriate air pressure and temperature in the intake manifold, and the exhaust back pressure was calibrated.

FMEP was measured from the IMEP at idle operation points from the low to the high in 250 1/min engine speed steps.

The injection process can be specified by the injector geometry, rail pressure, injection timing, and duration. The rail pressure map was measured with the pressure sensor of the EDC. The injector geometry was measured by a measurement microscope. The injection duration and timing was analysed with a current clamp installed on the solenoid controller wire. This way the pilot injections could be investigated as well. Since the main goal was to predict emissions and cylinder pressures even at high EGR rates, a quasi-dimensional DI-Jet combustion model was chosen \[56, 79\]. It was calibrated to measured in-cylinder pressure and heat release curves.

These burn rates were calculated from measured indicated pressure traces in static operation points with the following formula \[62\]:

\[
\frac{dQ_b}{dp} = \frac{\kappa}{\kappa - 1} \cdot \frac{dV}{dp} + \frac{V}{\kappa - 1} \cdot \frac{dp}{dp} - \frac{dQ_{iw}}{dp} \tag{5.3}
\]
The instantaneous chamber volume belongs to a given crank angle:

\[ V_{\text{cyl}} = f(\varphi) = V_c + V_d = \frac{V_{d,\text{max}}}{\epsilon - 1} + V_d(\varphi) \]  
(5.4)

The only remaining unknown value from these parameters is the cylinder volume above the piston at a given crank angle, which is calculated as:

\[ V_d(\varphi) = s_\varphi \cdot A_p = \left[ r \cdot (1 - \cos \varphi) + l \cdot \left( 1 - \sqrt{1 - \lambda_c^2 \sin^2 \varphi} \right) \right] \cdot \frac{D^2 \pi}{4} \]  
(5.5)

Discharge coefficient of the EGR throttle valve was calculated from measured EGR and engine air mass flow rate using the following formulas. EGR flow rate can be calculated as:

\[ \sigma_{\text{egr}} = \sigma_{\text{engine}} \cdot x_{\text{egr}} \]  
(5.6)

The EGR rate is calculated from measured oxygen concentrations in exhaust and intake pipes [62]:

\[ x_{\text{egr}} = \frac{x_{\text{O}_2, im} - x_{\text{O}_2, air}}{x_{\text{O}_2, lo} - x_{\text{O}_2, air}} \]  
(5.7)

Then the EGR valve discharge coefficients can be calculated as [27]:

\[ c_{d,\text{egr}} = \frac{\sigma_{\text{egr}}}{A_{\text{egr}} \cdot \frac{p_{\text{em}}}{p_{\text{im}}} \cdot \Psi} \]  
(5.8)

The flow function \( \Psi \), calculated the following way:

\[ \Psi \left( \frac{p_{\text{im}}}{p_{\text{em}}} \right) = \sqrt{\frac{2 \cdot p_{\text{im}}}{p_{\text{em}}} \cdot \left[ 1 - \left( \frac{p_{\text{im}}}{p_{\text{em}}} \right) \right]} \]  
(5.9)

for subsonic conditions. Here stands for the intake manifold pressure.

Then the FMEP values were the only missing data to reach accurate effective brake torques. These were measured in the whole engine map with cylinder pressure indication.

5.4 Static Validation

The detailed engine model was validated first in stationary operational points. These points were chosen to cover the whole load map of the engine equidistantly. The selected engine speeds for static investigation are the idle speed of 900 RPM, and in 250 RPM increments from 1000 RPM to 2500 RPM (rated speed). At each speed, the load was incremented by 100 Nm. The model runs with a fuel dosage controller to ensure the prescribed brake torque. Additionally, the full load curve was investigated with the measured dosage preset. The accuracy of the boost pressure plays an essential role in the cylinder
filling process. Namely, the cylinder charge mass is proportional to the intake gas density. The deviation of the measured and simulated intake manifold pressures can be seen in Fig. 5.6.

With the tuning of the compressor and turbine maps, it was possible to achieve a boost pressure deviation below 10%. However, in a wide range, the variation is less than 5%. Only in the low speed, high load region approximates the 10% value. This region is important due to the proximity of the turbocharger surge line.

Another essential property of the intake gas is its temperature, determined by the intercooler. As seen in Fig. 5.7 the error of the intake air temperature is below 3%.
The exhaust manifold pressure has a lower effect on the volumetric efficiency than the boost pressure, but through the turbine power, it has a crucial influence on the compressor operation as well. Its accuracy is also essential concerning the recirculated exhaust gas mass flow rate because it is driven by the pressure drop between the intake and exhaust manifolds.

The exhaust manifold pressure deviation shows a similar contour plot to the boost pressure deviation because of the coupled operation of the compressor and turbine. The turbine pressure ratio was tuned with the mass flow rate scale of the turbine map. The deviations also remain below 10 % (see Fig. 5.8).
The turbine power is also affected by the exhaust manifold temperature. The position and shape of the burn rate function have the main influence, besides the air-fuel ratio. The temperature deviation of the exhaust gases is below 10% in a broad region of the engine map (see Fig. 5.9).

Figure 5.9: Exhaust manifold temperature deviation

Fig. 5.10 shows the engine intake mass flow rate differences from measured values. The correlation with the boost pressure deviation map can be observed. At low speeds and high loads, the in-cylinder mass flow rate is higher than measured due to the higher boost pressure. The maximum deviation is below 14%.

Figure 5.10: Engine intake mass flow rate deviation
The differences in the measured and modeled air-excess ratios also show a correlation with the inducted mass flow rates (see Fig. 5.11). At low speeds and high loads, the predicted lambda values are higher than the measured ones as a result of the higher intake air mass flow.

![Figure 5.11: Lambda deviation](image)

The most important influence on the brake specific fuel consumption is caused by the air-excess ratio. For this reason, a high correlation is seen between the lambda and BSFC deviation plots. The error is the highest also in the low speed – high load region of the map. At low loads, it is also difficult to fit the model to measurements because the share of the friction and the heat transfer losses are high. The highest deviations remain below 10% (see Fig. 5.12).

![Figure 5.12: Brake specific fuel consumption deviation](image)
The injection and combustion model was validated based on indicated cylinder pressure curves and heat release rates. The fit of the measurements and the simulation can be seen in Fig. 5.13. Finally, the validation of the effective engine torques was performed. The simulation results are depicted along the measurements in Fig. 5.14. The difference here is also below 5% so that the model can be accepted for the control investigations.

Figure 5.13: Cylinder pressure and heat release rate validation

Figure 5.14: Effective engine torque validation
5.5 Transient Validation

After the static evaluations, transient tests were performed using legally prescribed World Harmonized Transient Cycle (WHTC) (see [74]). Here for comparison, a section of the cycle from 990 to 1150 seconds is shown with and without exhaust gas recirculation. The engine operation points of the whole WHTC cycle for the investigated engine are depicted in Fig. 5.15.

The motoring points were measured and depicted with zero loads because the applied eddy current brake cannot produce negative torques (drive the engine). It can be seen that engine speeds over 1500 RPM are quite rare. The high number of low load points can also be observed.

Fig. 5.16 shows the comparison of simulated and measured intake and exhaust manifold pressures, along with the fuel consumption, brake torque and engine speed in the selected portion of WHTC without EGR.

The turbocharger inertia had a crucial effect on the pressure buildup dynamics both in the intake and exhaust manifold. Therefore, it needed a careful setup. Another sensitive parameter is the volume of the intake and exhaust systems downstream of the compressor and upstream the turbine, which were accurately modeled in the above-described way with GEM3D. It ensured the equality between the real and modeled volumes. All curves show a good fit in the whole investigated period. The Root-mean-square (RMS) error of the intake and exhaust manifold pressures without EGR was evaluated based on the following equations:
Figure 5.16: Model comparison in WHTC without EGR between 990 and 1150 seconds

\[
\varepsilon_{p_{im, \text{woEGR}}} = \frac{1}{T_{\text{cycle}}} \int_0^{T_{\text{cycle}}} \left( \frac{p_{im, \text{meas,woEGR}} - p_{im, \text{sim,woEGR}}}{p_{im, \text{meas,woEGR}}} \right)^2 dt = 2.57\% \tag{5.10}
\]

\[
\varepsilon_{p_{em, \text{woEGR}}} = \frac{1}{T_{\text{cycle}}} \int_0^{T_{\text{cycle}}} \left( \frac{p_{em, \text{meas,woEGR}} - p_{em, \text{sim,woEGR}}}{p_{em, \text{meas,woEGR}}} \right)^2 dt = 1.44\% \tag{5.11}
\]

In the following comparison, the performance of the detailed model was investigated in the same cycle with the use of the HP-EGR duct. The EGR valve was actuated based on a predefined lookup table in which the EGR valve position was defined in function of the engine speed and brake mean effective pressure. At low loads, a fully opened EGR valve was set which was gradually closed while moving towards higher loads.

Similarly the RMS errors with EGR show a good fit:

\[
\varepsilon_{p_{im,wEGR}} = \frac{1}{T} \int_0^{T} \left( \frac{p_{im, \text{meas,wEGR}} - p_{im, \text{sim,wEGR}}}{p_{im, \text{meas,wEGR}}} \right)^2 dt = 2.63\% \tag{5.12}
\]

\[
\varepsilon_{p_{em,wEGR}} = \frac{1}{T} \int_0^{T} \left( \frac{p_{em, \text{meas,wEGR}} - p_{em, \text{sim,wEGR}}}{p_{em, \text{meas,wEGR}}} \right)^2 dt = 1.26\% \tag{5.13}
\]

From the comparison of EGR rates one can see that the model cannot reconstruct precisely the high-frequency dynamic fluctuation of
Figure 5.17: Model comparison in WHTC with EGR between 990 and 1150 seconds

the exhaust gas rate, but in mean values, it approximates well the exhaust gas fraction of the cylinder charge.
6.1 Introduction

In this Chapter, three operation points were chosen from the European Stationary Cycle (ESC) emission measurement cycle in which engine performance measures were analyzed as a consequence of EGR mass flow rate support by throttling at different locations of the engine air path (downstream of the compressor and upstream and downstream of the turbine) as suggested in Section 2.2. The selected operation points are the ‘A’ speed at 25% load, the ‘B’ speed at 50% load and the ‘C’ speed at 75% load. For the particular engine, the speeds and the effective torques are A25=1500 RPM and 150 Nm, B50=1900 RPM and 290 Nm, C75=2300 RPM and 375 Nm which cover the most relevant operation range of the engine. To satisfy the low NOx production limitation of the Euro VI emission norm it is essential to reach high EGR rates. For these high rates in several engine operation mode, it is not enough to fully open the EGR valve. To increase the EGR rate, additional interventions are needed. These can be carried out with flaps at different locations of the intake and exhaust system as it was described in Section 2.2. The investigated placements are the following: downstream of the compressor, upstream of the turbine and downstream of the turbine. At each operation point, the different EGR supports were studied. First, as a reference, the EGR valve was fully closed. In a second simulation, the EGR valve was fully opened (the choke flaps were fully opened), which is illustrated with blue color. It resulted in 16.4 % EGR rate, which is the maximum EGR case in each operation point. The goal is to compare the different exhaust gas recirculation choices. The simulation study presented below was published in [P1] preliminary.

6.2 Results in a25 operation point

In the initial case, the EGR valve was closed. It was depicted with brown color in the diagrams. In the second case only the EGR valve was opened (the choke flaps were fully opened), which is illustrated with blue color. It resulted in 16.4 % EGR rate, which is the maximum
achievable without any support. The EGR rate can be calculated with the following equation:

\[ x_{egr} = \frac{\sigma_{egr}}{\sigma_{egr} + \sigma_{air}} \]  

(6.1)

In the last three cases, the desired EGR rate was increased to 25 % by the support measures. The throttle position was set by a controller to achieve the desired 25 % EGR rate. The effective torque is maximal at the no EGR case and decreases with higher EGR rates. More EGR results in prolonged heat release and hence less IMEP360. The turbine power depends strongly on the mass flow rate, so it correlates to the fresh air mass flow rate. PMEPs are relatively low values since the intake and exhaust pressure is nearly identical to the initial case (positive PMEP represents losses).

The pressure drops on the chokes are similar values in each case. The maximal intake manifold pressure with the 25% EGR rate was found when the pressure drop was applied upstream the turbine. In the initial case, the amount of the soot is minimal due to high air-fuel ratio. Applying 16.4 % EGR rate the NOx reduced by 38%. The further 10 % EGR rate increase could not decrease the NOx production but promoted the soot formation.

![Figure 6.1: Torque, BSFC, intake gas mass flow rates in the A25 operation point](image)

Figure 6.1: Torque, BSFC, intake gas mass flow rates in the A25 operation point
Figure 6.2: IMEP\textsubscript{360}, PMEP, lambda and turbine power in the A25 operation point

Figure 6.3: Manifold pressures and pressure drop values on the chokes in the A25 operation point

Figure 6.4: NO\textsubscript{x}-soot trade-off in the A25 operation point
6.3 RESULTS IN B50 OPERATION POINT

With fully open EGR valve 14.5% EGR rate was developed. It was increased to 25% in the last three cases with support. The torque and fuel consumption shape show a similar trend to the A25 case: the initial torque is the highest and the support downstream of the compressor results in the least effective support. The highest overall intake mass flow rate of the 25% EGR cases is when the support is applied upstream to the turbine. The difference between the intake and exhaust manifold pressure is maximal in the first case without EGR. If the EGR valve is opened, the pressure will be equalized through the EGR system. It means less charge exchange losses and less fuel consumption.

The NO$_x$ concentration decreases significantly (by 70%) in the higher EGR cases. The minimum value results with the intake support but due to low oxygen concentration the soot emission is high.

![Figure 6.5](image.png)

**Figure 6.5:** Torque, fuel consumption, intake gas mass flow rates in the B50 operation point

![Figure 6.6](image.png)

**Figure 6.6:** IMEP$_{360}$, PMEP, lambda and turbine power in the B50 operation point
Figure 6.7: Manifold pressures and pressure drop values on the chokes in the B50 operation point.

Figure 6.8: $NO_x$-soot trade-off in the B50 case operation point.

6.4 RESULTS IN C75 OPERATION POINT

The available EGR rate without any support was 19.9%. The performance results are shown in Fig. 6.9. The brake torque values are here the most balanced. With fully open EGR valve but no support (blue column) the simulation results in higher brake power. As seen in 6.10 the indicated mean pressure is less in the second case than the initial was due to the recirculated inert gases. Despite this fact, the net power can be increased due to the minimized charge exchange losses (PMEP).
Figure 6.9: Torque, fuel consumption, intake gas mass flow rates in the C75 operation point

The reason for the lower PMEP values in EGR cases can be traced in Fig. 6.11. With opening the EGR valve, the intake and exhaust manifold will be connected to each other, and the pressures can be balanced through the EGR loop. The NOₓ-soot trade-off shaped similarly as in the previous simulations. The NOₓ reduction is even more pronounced with EGR.

Figure 6.10: IMEP_{360}, PMEP, lambda and turbine power in the C75 operation point
Figure 6.11: Manifold pressures and pressure drop values on the chokes in the C75 operation point

Figure 6.12: NO$_x$-soot trade-off in the C75 operation point

6.5 Conclusion

This chapter investigates the EGR rate support alternatives by air-path throttling regarding engine performance. In A25 operation point, the NO$_x$ formation was moderately reduced by the exhaust gases, and a further EGR rate increase was not sufficient: only soot emission increased. The lowest nitrogen-oxide concentration resulted with the support downstream the compressor but with an increased PM emission. Turbine up and downstream support were similarly effective, but with a much lower PM. It was interesting to see that an EGR rate increase is how far not the only factor for low NO$_x$ emission. On the other hand, other effects of higher EGR rate realization makes severe consequences to the engine performance and soot emission. The B50 operation point results show similar effects, but the EGR rate increase with the support continued to decline the nitrogen oxides. The best value occurred again with the flap in the intake system, but the soot is unacceptable. Turbine supports are the same way preferred with low
PM emissions and similar NO\textsubscript{x}. In C75 operation point, improvement of the brake torque can be noticed with the fully open EGR valve. It is caused by the balanced intake and exhaust manifold pressure. The pumping losses are minimized this way. The NO\textsubscript{x}-soot trade-off diagram looks similar to the previous cases. As a summation, it can be assessed that the support downstream of the compressor is the best EGR rate increase measure if the goal were only to minimize the nitrogen-oxide emissions. However, it is disappointing in PM emission and fuel consumption. In perspective of the fuel consumption, particulate matter and NO\textsubscript{x} the best support option is the upstream to the turbine throttling. It ensured the lowest BSFC, the highest boost pressure, and intake fresh air mass flow. Although, the predicted NO\textsubscript{x} formation is only slightly higher than with downstream the compressor throttling while it results in significantly less soot due to the higher air-to-fuel ratios. However, a drawback of this solution is the difficult realization of the actuator due to the high exhaust gas temperature and pressure upstream of the turbine. For such a high mileage which is expected for commercial vehicles, it cannot be manufactured at a reasonable cost. Putting the actuator location downstream of the turbine the temperature and pressure levels sink dramatically which means a significantly cheaper design of the flap valve and the actuator. Whilst engine BSFC, fresh air mass flow rate and NO\textsubscript{x}-soot trade-off results are only marginally worse of the downstream the turbine actuator layout than upstream the turbine. Considering the above-described design difficulties and the only slightly poorer engine performance results than the best upstream the turbine throttling location, a reasonable compromise could be achieved with controlled flap downstream of the turbine.

As a conclusion, choking at the exhaust side was more beneficial than the intake side choking: lower BSFC, higher boost pressure and charge air mass flow rate and better soot-NO\textsubscript{x} trade-off was achievable. Similarly to functions listed in Section 2.1 the upstream the turbine location throttling was the most beneficial for the cylinder charge control too. Due to the above-mentioned disadvantages of the upstream the turbine location (the actuator is exposed to high temperature and pressure) the downstream the turbine location with the hot side HP-EGR valve was chosen as the preferable engine layout for cylinder charge control. Hence, the cylinder charge control solution was worked out for this selected engine setup, and its performance is demonstrated with engine dyno measurements in Part iv.
SUMMARY

In this part of the thesis, an optimal engine air path setup with throttle valves was developed. Based on the targeted possible new engine functions of the brake blending, AMT support, exhaust gas thermomanagement, cylinder charge composition control (summarized in Chapter 2) control strategies were worked out. To derive the most advantageous throttle locations in the engine intake and exhaust system from the several possibilities (downstream of the compressor and upstream and downstream of the turbine) a simulation study was performed. As a first step, a detailed engine model was built up in GT Suite environment and validated with engine dyno measurements. With the help of the validated model a comparison study was made which showed significant differences in engine measures (focusing especially on fuel consumption and emission) between different engine air-path setups. Based on the evaluation a HP-EGR valve in cooperation with exhaust throttling downstream of the turbine was chosen as an optimal engine air-path throttling setup as a practical trade-off.

To achieve the realization of the possible targeted new engine functions control strategies were developed. For the realization of brake blending, AMT support and exhaust gas thermomanagement functions an exhaust manifold backpressure control aim seems to be a suitable choice. For adjustment of the cylinder gas charge composition, the control of the EGR mass flow with the actuators of EGR valve and exhaust throttle downstream the turbine offers the most beneficial solution. The next parts of the dissertation describes the achievement of these two control aims.
Part III

EXHAUST THROTTLE BACKPRESSURE CONTROL
MODEL DEVELOPMENT, IDENTIFICATION AND VALIDATION

8.1 INTRODUCTION

On commercial vehicle diesel engines, a widely used and cost-effective endurance brake is the exhaust brake. These brake retarders are typically designed as butterfly valves and applied for generating back-pressure for the engine exhaust. The brake power can be increased by higher backpressure in the exhaust manifold, but its value is limited by the valvetrain design [33]. The allowed overpressure is engine specific so it is an exhaust brake design parameter. To handle it several exhaust throttle designs were invented. For example: using a pressure regulator valve which can bypass the restrictor valve or flap axle with an offset, etc. see, e.g., [72] and [71]. To fulfill the requirements as endurance brakes, exhaust throttles on modern commercial vehicle diesel engines have only two states: fully opened and fully closed. The generation of intermediate pressure levels and tracking pressure demands with a high response would allow the use of exhaust throttles in the following possible new application fields suggested by the author. With appropriate control of the engine exhaust manifold pressure the engine brake power could be adjusted arbitrarily. In this way, the backpressure controlled exhaust brake operation could substitute a service brake at moderate brake torque demands. This is called brake blending. So the exhaust throttle could be integrated into the service brake operation and could increase the lifetime of brake pads. Brake blending experiments with a compression release brake are presented in [51]. Another benefit of an exhaust backpressure controller could be utilized during the upshifting of gears with an AMT gearbox. Before the clutch engagement the engine speed needs to be synchronized to the gearbox input shaft speed. The time demand for this synchronization process, namely the engine deceleration, could be decreased with suitable backpressure generation in the exhaust manifold. This results in quicker vehicle acceleration. A similar application with an engine compression release brake can be seen in [68]. Engine exhaust throttles also could provide multiple functionalities assisting the fulfillment of new requirements resulting mainly from evermore rigorous emission standards. Regulations that have recently come into force (Euro 6 and US EPA 13) include significant limitations, especially on soot and nitrogen oxides. This creates a constant challenge to engine developers. The two ways which can be utilized to produce more environmental friendly engines are exhaust
gas aftertreatment (i.e., SCR, DPF, etc.) and the restriction of pollutant formation during the combustion process (i.e. exhaust gas recirculation). To meet new requirements in the future, the use of both possible methods will probably be unavoidable \[73\]. To maintain high efficiency in engine exhaust aftertreatment systems, it is desirable to avoid cooling by cold exhaust gases in low-load engine conditions and also rapid heating. For details see \[53\]. In this case, controlled exhaust backpressure could be used for exhaust gas thermomangement. In HP-EGR systems the back-flowing exhaust gas mass flow rate is driven by the pressure difference between the exhaust and the intake manifold. The resulting conditions depend mainly on the turbocharger and its cooperation with the engine. With the application of exhaust throttling, the pressure drop on the EGR duct could be adjusted, and EGR rate could be increased. Additional nitrogen oxide emission reduction and combustion method improvement would also be possible. An exhaust backpressure controller with a variable geometry turbine for turbocharged SI engines was presented in \[23\]. Similar EGR rate increase with an intake throttle was introduced in \[24\]. The contribution of this work in this part is a backpressure controller function which allows the use of exhaust throttles in the above mentioned possible new application fields. The requirements for the prior future functions are defined, and a controller is designed which meet them. A first engineering principle-based, mean-value, nonlinear engine and actuator model is described and validated with engine test bench measurements. Based on the nonlinear model, a feedforward term was obtained by model inversion. An LQ servo controller was designed as a feedback. In order to handle the saturation of the control signal a high gain anti-windup is applied. The performance of the controller is demonstrated with the help of measurements on a medium-duty diesel engine installed on an engine test bed. Three different test cycles are created and used in order to simulate brake blending, thermomangement, and EGR support operation. Finally, the compliance of the requirements is evaluated.

### 8.2 System Description and Experimental Setup

The controllers were tested on a medium-duty commercial vehicle, a common rail, a turbocharged and an intercooled diesel engine. The engine was equipped with a cooled HP-EGR system. The exhaust throttle valve was installed directly downstream of the turbine, which provides the minimum volume between the engine exhaust valves and the throttle flap for minimizing pressure rise times. The layout of the test engine is shown in Fig. 8.1.
The engine was installed on an engine test bench where all the operating parameters of the engine relevant for the system model and backpressure controller were measurable. The controller design was performed in a MATLAB/Simulink environment. The resulted controller was implemented and tested on a rapid control prototyping unit. The presented exhaust throttle controller is a part of the overall vehicle control system including the EDC and the EBS. The architecture is shown in Fig. 8.2.

The EBS controller requests a brake torque demand \( M_{brk, dem} \), which is received by the EDC and converts it into an exhaust manifold pressure demand \( p_{em, dem} \) to the Exhaust Throttle Controller (XTC). The aim is to achieve the expected deceleration of the vehicle. Moreover, the EDC can generate exhaust manifold pressure demand for its internal functions (e.g., thermomanagement, etc.) as well. The XTC provides the actual measured \( p_{em} \) signal as feedback. The experimental setup is depicted in Fig. 8.3.
8.3 System Model

The model building and validation process presented below was published in the following papers by the author: \([\text{P7, P8}]\). Preliminary modeling assumptions were taken to obtain the simplest model form according to our goals:

\(A1\). The potential energy is neglected.

\(A2\). Constant physical and chemical properties are assumed over each balance volume of the model, such as specific heat, specific gas constant and adiabatic exponent etc.

\(A3\). The adiabatic exponent, the specific gas constant and consequently the specific heat of the air and the exhaust gas are equal, which is a reasonable assumption because diesel engines operate with a lean mixture.

\(A4\). There are no mass and energy storage effects in the combustion chamber.

\(A5\). The fluids can be modeled as an ideal gas.

\(A6\). The temperature of the outflowing gas from the receiver is equal to the receiver’s temperature.

\(A7\). The inlet temperature and pressure of the compressor is equal to the ambient temperature and pressure: \(T_{c,in} = T_{amb}\) and \(p_{c,in} = p_{amb}\).

\(A8\). The mass of the fuel was neglected.
The intended use is control system design, so the model should be written in state-space form. The systematic modeling procedure is based on first engineering principles and follows the modeling procedure published in [58]. The hierarchical structure of a dynamic model can be separated into set of the following model elements:

- balance volumes over which conservation balances are constructed (the highest level),
- balance equations,
- terms in balance equations corresponding to mechanisms,
- constitutive equations,
- variables and parameters (the lowest level)

The modeled engine can be separated into the following four balance volumes: the intercooler, the intake manifold, the exhaust manifold and the volume between the turbine and the exhaust brake. However, for backpressure control the modeling of the exhaust manifold only seemed to be a suitable simplification. See this balance volume depicted in dashed line in Fig. 8.1.

Balance volumes can be modeled as receivers with mass flow as input and output and for which the thermodynamic states assumed to be the same over the entire volume. The model equations can be derived from the following mass conservation law as balance equation and additional equations that give the connections between the conservation differential equation and the thermodynamic parameters. The temperature of the out-flowing gas is assumed to be equal to the gas temperature in the receiver.

The mass conservation law:

$$\frac{dm}{dt}(t) = \sigma_{in}(t) - \sigma_{out}(t) \tag{8.1}$$

To reach the differential equation of the pressure, the ideal gas law can be used:

$$p(t) \cdot V = m(t) \cdot R \cdot T(t) \tag{8.2}$$

For balance volumes for which the temperature of the inlet gas and the temperature of the gas in the receiver are nearly equal, the isothermal assumption is a good approximation. The differential equation for the pressure can be obtained by differentiating Eq. 8.2 and substituting Eq. 8.1 in it in the following form:

$$\frac{dp(t)}{dt} = \frac{R \cdot T}{V} \left[\sigma_{in}(t) - \sigma_{out}(t)\right] \tag{8.3}$$

To ensure the model simplicity the differential equation for the temperature calculation was avoided and the temperature was pre-defined. Using the derived differential equation Eq. 8.3 the pressure level can be defined in balance volumes.

As a balance volume, the exhaust manifold was chosen (marked with a dashed line in Fig. 8.1). The effect of the turbocharger turbine
was neglected because of the small pressure drop in relevant operation scenarios of the backpressure controller. The isothermal differential equation for the pressure as state variable was defined as follows:

\[
\frac{dp_{em}}{dt} = \frac{R_{\text{air}}}{V_{em}} \cdot \left[ \sigma_{eo} - \sigma_{ei} - \sigma_{egr} \right].
\] (8.4)

The exhaust gas recirculation valve in the above mentioned typical operation range of the backpressure controller is usually closed. For example, in the brake blending mode, there is no fuel injection into the cylinders. Consequently, there is no nitric oxide formation, so there is no purpose for the use of the EGR. The same can be said for the AMT support usecase. Therefore, for the sake of simplicity, the \( \sigma_{egr} \) was neglected in the model described below. Alternatively, the EGR mass flow can be calculated from the measured position of the EGR valve and the intake and exhaust manifold pressure differences similarly to Eq. 11.14. This way, the EGR mass flow could be treated as a measurable disturbance.

The exhaust gas mass flow rate is the sum of the engine intake gases and the injected fuel quantity:

\[
\sigma_{eo} = \sigma_{ei} + \sigma_{fuel}.
\] (8.5)

Due to high amount of excess air the fuel mass flow rate is relative small compared to the intake air, especially at low engine loads. So \( \sigma_{fuel} \) was treated as zero. Therefore the use of the specific gas constant of air as well as a constant exhaust gas temperature is a good simplification.

The engine can be modeled as a positive displacement pump, so the induced gas mass flow rate into the cylinders is formulated in the following form:

\[
\sigma_{ei} = \eta_{\text{vol}} (p_{im}, p_{em}, n_e) \cdot \varrho_{im} \cdot n_e \cdot V_d = \eta_{\text{vol}} (p_{im}, p_{em}, n_e) \cdot \frac{p_{im}}{R_{\text{air}} \cdot T_{im}} \cdot n_e \cdot \frac{V_d}{i}.
\] (8.6)

The engine volumetric efficiency was approximated with the following multilinear formula:

\[
\eta_{\text{vol}} (p_{im}, p_{em}, n_e) = \eta_{\text{vol}, n_e} (n_e) \cdot \eta_{\text{vol}, \Delta p} (p_{im}, p_{em}).
\] (8.7)

The value of the parameter depending on the engine speed is treated as a constant, which due to the narrow engine speed range is a good approximation for a commercial vehicle diesel engine.

The value of the parameter depending on the pressure difference takes into consideration the effect of the increased exhaust manifold backpressure. Due to trapped exhaust gases in the cylinder, it causes a decrease of the volumetric efficiency. The approximation is based on the suggestion in [27].
So the engine outlet gas mass flow rate can be written as:

\[
\sigma_{eo} = \eta_{vol,n_e} \cdot \left[ \frac{V_c + V_d}{V_d} - \left( \frac{p_{em}}{p_{im}} \right)^{\frac{i}{\kappa}} \cdot \frac{V_c}{V_d} \right] \cdot \frac{p_{im}}{R_{air} \cdot T_{im}} \cdot n_e \cdot \frac{V_d}{i}. \tag{8.9}
\]

The mass flow through the exhaust throttle valve can be calculated with the standard orifice equation:

\[
\sigma_{et} = c_{d,et} \cdot A_{et}(\varphi_{et,act}) \cdot \frac{p_{em}}{\sqrt{R_{air} \cdot T_{em}}} \cdot \Psi \left( \frac{p_{em}}{p_{amb}} \right). \tag{8.10}
\]

The flow function \( \Psi \left( \frac{p_{em}}{p_{amb}} \right) \) is defined depending on the flow conditions:

\[
\Psi \left( \frac{p_{em}}{p_{amb}} \right) = \begin{cases} 
\sqrt{\frac{2}{\kappa+1}} & \text{for } p_{amb} < p_{cr} \\
\left( \frac{p_{amb}}{p_{em}} \right)^{\frac{1}{\kappa}} \cdot \sqrt{\frac{2\kappa}{\kappa-1} \cdot \left[ 1 - \left( \frac{p_{amb}}{p_{em}} \right)^{\frac{\kappa-1}{\kappa}} \right]} & \text{for } p_{amb} \geq p_{cr}
\end{cases}
\]

where the critical pressure is:

\[
p_{cr} = \left[ \frac{2}{\kappa + 1} \right]^{\frac{1}{\kappa}} \cdot p_{em}. \tag{8.11}
\]

A nonlinear formulation for the geometrical area of a butterfly valve can be found in [16] and [32] as described below:

\[
A_{et}(\alpha) = \frac{D_{th}^2}{4} \cdot \frac{\pi}{4} \cdot \left( 1 - \frac{\cos \alpha}{\cos \alpha_0} \right) + A_{leak}. \tag{8.13}
\]

Taking into consideration the throttle shaft diameter the expression becomes more complex. As a consequence of the nonlinear behavior of the system, the geometrical area of the throttle valve is only relevant in the range of fully closed flap positions. In this region the throttle area as a function of the throttle flap position can be approximated with a linear expression as defined below:

\[
A_{et}(\varphi_{et,act}) = A_{et,0} - A_{et,0} \cdot \varphi_{et,act} + A_{leak}. \tag{8.14}
\]

As can be seen, the system behavior is not continuous due to the valve mass flow dependence on the flow conditions. To describe the model as compactly as possible a nominal hybrid mode was chosen in order to obtain a unique model structure. The backpressure controller operates mostly at sonic flow speeds therefore in the nominal hybrid mode the mass flow through the valve is:
\[ \sigma_{et} = c_{d,et} \cdot (A_{et,0} - A_{et,0} \cdot \varphi_{et,act} + A_{leak}) \cdot \frac{p_{em}}{\sqrt{R_{air} \cdot T_{em}}} \cdot \sqrt{\frac{2}{\kappa (\kappa + 1)}}. \] (8.15)

This simplification overestimates the mass flow rate through the valve in the subsonic flow condition so it will result in lower modeled pressure levels. Although, these states usually happen only in transients during the brake blending, AMT support and thermomenagement usecases. In these cases controller robust performance is expected to handle the model inaccuracy.

As a consequence of the nonlinear behavior of the system, the geometrical area of the throttle valve is only relevant in the range of fully closed flap positions. In this region the throttle area, as a function of the throttle flap position, can be approximated with a linear expression as defined in Eq. 8.15 in the first round bracket. More accurate nonlinear formulations for the geometrical area of a butterfly valve can be found in [16] and [32]. The actuator dynamics was modeled as a first-order lag.

Due to its comparable rise times with the balance volume pressure, the inclusion of the actuator dynamics model is inevitable. Seeking for the simplest model it was defined as a first-order lag in the form below:

\[ \frac{d\varphi_{et,act}}{dt} = -\frac{1}{T_{et}}\varphi_{et,act} + \frac{1}{T_{et}}\varphi_{et,dem}. \] (8.16)

For controller design the model was written in state space-form as:

\[ \frac{dx}{dt} = f(x, d) + g(x, d) u. \] (8.17)

The state vector is:

\[ x = [p_{em}, \varphi_{et,act}]^T, \] (8.18)

the input vector is:

\[ u = [\varphi_{et,dem}]^T, \] (8.19)

and the measurable disturbance vector is:

\[ d = [n, p_{im}]^T. \] (8.20)

With the above defined notations the state-space model can be defined as follows:

\[ \begin{bmatrix} p_{em} \\ \varphi_{et,act} \end{bmatrix} = \begin{bmatrix} f_1(x, d) \\ f_2(x, d) \end{bmatrix} + \begin{bmatrix} g_1(x, d) \\ g_2(x, d) \end{bmatrix} \begin{bmatrix} \varphi_{et,dem} \end{bmatrix}, \] (8.21)
\[ f_1(x, d) = \frac{R_{\text{air}} \cdot T_{\text{em}}}{V_{\text{em}}} \cdot \left[ \frac{V_c + V_d}{V_d} - \left( \frac{p_{\text{em}}}{p_{\text{im}}} \right)^{\frac{i}{2}} \cdot \frac{V_c}{V_d} \right] \cdot \frac{p_{\text{im}}}{R_{\text{air}} \cdot T_{\text{im}}} \cdot n_e \cdot \frac{V_d}{i} \]

\[ - \varepsilon_{d, et} \cdot (A_{et, 0} - A_{et, 0} \cdot \phi_{et, \text{act}} + A_{\text{leak}}) \cdot \frac{p_{\text{em}}}{R_{\text{air}} \cdot T_{\text{em}}} \cdot \sqrt{\kappa \left( \frac{2}{\kappa + 1} \right)} \left( \frac{\kappa + 1}{2} \right)^{\frac{i+1}{i}} \]

\[ g_1(x, d) = 0, \quad (8.23) \]

\[ f_2(x, d) = -\frac{1}{T_{et}} \phi_{et, \text{act}}, \quad (8.24) \]

\[ g_2(x, d) = \frac{1}{T_{et}}, \quad (8.25) \]

The measured and performance outputs are respectively:

\[ y = \left[ \phi_{et, \text{act}}, \ p_{\text{em}} \right]^T, \quad (8.26) \]

\[ z = \left[ p_{\text{em}} \right]^T. \quad (8.27) \]

### 8.4 Model Parameter Identification

There are 13 parameters \( R_{\text{air}}, T_{\text{em}}, V_{\text{em}}, \eta_{\text{vol}, n_e}, V_c, V_d, \kappa, T_{\text{im}}, i, \varepsilon_{d, et}, A_{et, 0}, A_{\text{leak}}, T_{et} \) in the above-described model that need to be identified. The parameter values were defined based on the following systematic process. Some parameters are related to either the engine or the flap geometry or associated with a specific theory or the above-described modeling assumptions.

- \( V_c \) and \( V_d \) can be found in the engine datasheet, see 5.1.
- The whole intake and exhaust system was modeled in a 3D CAD system so the exhaust manifold volume could be measured directly, see 5.3.
- The whole flap valve geometry was also available as a 3D CAD model so the geometric flow area could be measured directly in function of the flap position and in this way the linear regression for the area modeling could be performed. Alternatively, the nonlinear approximation formula 8.13 could also be used.
- A four-stroke engine was used for testing therefore \( i \) was specified as 2.
• During the typical working scenarios of the backpressure controller (brake-blending or AMT support mode) there is no fuel injection into the cylinders, or at least the mixture is highly lean (thermomanagement mode) so the usage of the specific gas constant and adiabatic exponent of the air is a good approximation.

• Due to the reasons in the previous point, the definition of the intake manifold pressure as ambient is a good approximation or the EDC measurement can also be used.

The other parts of the parameters to be identified are unknown static parameters.

• The engine volumetric efficiency was calculated based on intake mass flow rate measurements with a hot film mass flow meter and fitted with the least-squares method.

• The flap valve discharge flow coefficient was fitted by the least-squares method to measured engine mass flow rate data considering the above described geometrical flow area definition.

• The exhaust manifold temperature enters the state equations in a nonlinear way, so its value was approximated with the simplex numeric optimization method based on random engine backpressure identification measurement cycles.

There is one dynamic parameter which value can be identified with transient tests.

• The actuator time constant ($T_{et}$) can be individually identified with a position control measurement.

The model identification process described in this chapter can be followed on any engine on which the backpressure controller application is targeted. This way it is suitable for serial automotive applications. The resulted parameters are listed in Table 8.1.
Table 8.1: List of engine backpressure model parameters

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adiabatic exponent</td>
<td>κ</td>
<td>1.4</td>
<td>-</td>
</tr>
<tr>
<td>Engine compression volume</td>
<td>$V_c$</td>
<td>0.0002393</td>
<td>$m^3$</td>
</tr>
<tr>
<td>Engine displacement</td>
<td>$V_d$</td>
<td>0.003922</td>
<td>$m^3$</td>
</tr>
<tr>
<td>Engine volumetric efficiency</td>
<td>$\eta_v$</td>
<td>0.8</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust manifold temperature</td>
<td>$T_{em}$</td>
<td>370</td>
<td>K</td>
</tr>
<tr>
<td>Exhaust manifold volume</td>
<td>$V_{em}$</td>
<td>0.0051</td>
<td>$m^3$</td>
</tr>
<tr>
<td>Exhaust throttle area parameter</td>
<td>$A_{et,0}$</td>
<td>0.0022</td>
<td>$m^2$</td>
</tr>
<tr>
<td>Exhaust throttle discharge coefficient</td>
<td>$c_{d,et}$</td>
<td>0.65</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust throttle leakage area parameter</td>
<td>$A_{leak}$</td>
<td>1.6250e-05</td>
<td>$m^2$</td>
</tr>
<tr>
<td>Flap actuator time constant</td>
<td>$T_{et}$</td>
<td>0.025</td>
<td>s</td>
</tr>
<tr>
<td>Intake manifold temperature</td>
<td>$T_{im}$</td>
<td>300</td>
<td>K</td>
</tr>
<tr>
<td>Number of revolutions per cycle</td>
<td>$i$</td>
<td>2</td>
<td>-</td>
</tr>
<tr>
<td>Specific gas constant</td>
<td>$R$</td>
<td>287</td>
<td>$J/kg \cdot K$</td>
</tr>
</tbody>
</table>

8.5 Model Validation

The model described in the previous section was validated with test bench measurements. The model accuracy was evaluated in test cycles at three different engine speeds which cover the relevant speed range of the engine: namely at 1000, 1500 and 2000 RPM. During the test cycle, constant, random, and 250 ms position demands were sent to the engine exhaust throttle valve. The range of the randomly generated throttle position demand signal was chosen to ensure perceptible pressure generation in the exhaust manifold. In the following figure, the comparison of the measured and modeled exhaust pressure as well as throttle valve positions are depicted at an engine speed of 1500 RPM.
Figure 8.4: Model validation at 1000 RPM engine speed

Figure 8.5: Model validation at 1500 RPM engine speed
To reach the modeling aim, namely, to define a model for controller design, fitting of the pressure state variable is desired. So as to show off the accuracy of the model the deviations were evaluated as root-mean-square errors in the validation cycles at each engine speed:

\[
\varepsilon_{p_{em}} = \sqrt{\frac{1}{T_{cycle}} \int_0^{T_{cycle}} \left( \frac{p_{em,m} - p_{em}}{p_{em,m}} \right)^2 dt}. \quad (8.28)
\]

The length of the entire sequence was 20 seconds in each case, and the suffix ‘\(m\)’ denotes the measured value. The RMS errors of the modeled exhaust manifold pressures were 9.5% at 1000 RPM, 8.9% at 1500 RPM, and 8.5% at 2000 RPM. The errors remain below 10% at each engine speed. Therefore, the model can be used for controller design.

Figure 8.6: Model validation at 2000 RPM engine speed
9.1 CONTROL AIMS AND REQUIREMENTS

The controller design and test procedure introduced below was published in the following papers by the author: [P5, P6, P7].

The control requirements which must be fulfilled by the engine exhaust backpressure controller were initially laid down as follows:

\( R_1 \). The steady state error of the exhaust manifold pressure must be below 0.2 bars at every engine operation point.

\( R_2 \). Application to various engines without measurements with the knowledge of the following parameters only should be possible: engine displacement, compression ratio, throttle valve diameter.

\( R_3 \). The overshoots should be below 0.2 bars to avoid potential engine damage caused by a too high backpressure level.

\( R_4 \). The controller must ensure the asymptotic stability of the system.

\( R_5 \). The rise time of the exhaust manifold pressure should be below 0.5 s to ensure the controller applicability for brake blending functions.

\( R_6 \). The peak-to-peak amplitude of the throttle flap position in steady state backpressures must be below 2 % to avoid actuator wear.

\( R_7 \). The complexity of the control algorithm should be low enough to allow its application in an embedded environment, where the clock rate of the applied single core processor is in the range of 40-50 kHz and the available memory is below 64 kbyte.

9.2 EXHAUST PRESSURE OSCILLATION ANALYSIS

In the previous section, a mean-value model was described for the exhaust manifold pressure. The model neglects the discrete cycles of the engine and assumes one constant source of mass flow into the exhaust manifold. In fact, the inflowing mass is distributed between the exhaust ports of the cylinders and shows highly varying values during the individual exhaust processes. This excitation effect causes periodic pressure oscillations in the exhaust manifold the amplitude of which (0.5 bars) can be compared to the expected control accuracy.
The frequency of the pressure pulsation is equal to the Engine Firing Rate (EFR) [57] and can be calculated by the following formula [65]:

$$EFR = \frac{n_e}{i} \cdot z. \quad (9.1)$$

The exhaust manifold pressure pulsation was analyzed based on test bench measurements, and the Power Spectral Density (PSD) of the pressure signal was evaluated at different engine speeds. The resulting graph can be seen in Fig. 9.1. The measurement returns the expected base harmonics frequencies based on Eq. (9.1). It can also be seen that the magnitude of the higher order harmonics is negligible (except 700 RPM). The highest power spectral density values are reached at 1800 RPM and 2000 RPM. Note that their magnitude values are depicted divided by six and three to ensure readability.

![Figure 9.1: Pressure oscillation PSD analysis of the exhaust manifold pressure signal](image-url)

The above described nonlinear model (Eq. (8.21)) was linearized with the Jacobian method around a typical operation point:

- $n_e = 1500$ RPM because the engine speed range is 700-2500 RPM and this speed is frequently used during the operation of the back-pressure controller.
- $p_{em} = 3$ bars and $\phi_{et,act} = 0.92$.

Fig. (9.2) shows the comparison of the linearized and the nonlinear system. As it reveals, the linearized system predicts acceptably the nature of the backpressure dynamics in the above-defined engine speed operation range. Of course, there is some error (which extent is increasing at the edges of the engine speed operation range) caused by the linearization around the equilibrium point, but these inaccuracies are targeted to handle by the robustness of the controller. Therefore, the above described Jacobian linearised model is accepted for the controller design.
Figure 9.2: Comparison of the linearized and nonlinear system response

The Bode plot of the system can be seen in Fig. 9.3.

Figure 9.3: Bode plot of the linearized model

The cutoff frequency of the plant is around 1 Hz (marked by the arrow). The frequency of the pressure oscillation is engine speed dependent and begins from 30 Hz at 700 RPM engine speed. The use of a filter on the feedback exhaust manifold pressure signal would introduce a phase lag in the closed loop. The value of the phase lag would be remarkable in the range of the open loop system’s bandwidth due to the small difference between the system cutoff and exhaust pressure pulsation frequency. It would cause slower closed loop response time or make the system unstable by consuming its phase margin. The use of filters without phase shifting (e.g., Kalman filter) is not feasible because the pressure pulsation occurs at a specific frequency and the computational demand of such filters is high. Therefore, filtering only is an inadequate solution. However, based on $R_5$ and $R_6$, the flap position oscillation need to be held below a certain limit while ensuring the predefined closed loop response time. The proposed controller which can fulfill these requirements in the presence
of the exhaust pressure oscillation is described in the following section.

9.3 **CONTROLLER DESIGN**

In $\mathcal{R}_4$ the asymptotic stability was required. Hence, only full state feedback controllers can be considered for controller design.

The first obvious choice is the linear quadratic regulator which is from the field of optimal control theory. It guarantees robustness with 60-degree phase margin if all the state variables are measured. It can be advantageous against modeling inaccuracies (see in Fig. (9.2)). The augmentation with an integrator (LQ servo) increases further the tracking and robustness capabilities. Moreover, the tuning of the LQ controller can be done with a few numbers of tuning parameters that make the design process straightforward. In this case, the disturbance vector elements were treated as parameters in the Jacobian linearization. Therefore, the engine speed and boost pressure measurement was not used directly to the control.

As a second step, the performance of the LQ servo is enhanced with a model inversion based feedforward that is independent of the disturbance effect of the exhaust pressure oscillation. However, the LQR controller may be suboptimal in case of non-white noise disturbance inputs (just like engine speed and boost pressure variation), but the integral control can be used to mitigate the effect of non-white noise disturbance. Another possibility to deal with the disturbance input is the inclusion of a feedforward control (see for LQ disturbance rejection details [14]), which was done in case of the second controller where for the online calculation of the feedforward term the measured signals of the disturbance vector (see Eq. (8.20)) was directly used.

As a third controller type to be designed an H-infinity controller comes from the robust control theory field. The effect of the model parameter changing was covered with an LPV model structure. Moreover, the measurable disturbance vector was included in the controller design. These signals were used online for calculation of the control input.

As a nonlinear control method, the sliding mode controller was chosen. The measurable disturbance signal was used to the online calculation of the equivalent control term directly.

This way four different controllers were designed for the given control problem and model to cover all the main fields of modern control theory. Please note that computationally demanding control design methods (e.g., Model Predictive Control - MPC, etc.) was excluded from this comparison due to the predefined requirement $\mathcal{R}_7$. The feedback signals (the exhaust manifold pressure and the flap actuator position) was measured in the case of all the four designed controller
(see Appendix A for sensor setup details). All of the controllers were tested and compared in test bench measurements taking into consideration the predefined control aims and requirements. The controllers were synthesized and implemented in the rapid prototyping unit in discrete time with a sample time of 1 ms.

9.3.1 LQ servo control structure

The first proposed controller is an LQ servo which is from the field of linear, optimal control. This type of controller was successfully used in diverse nonlinear control problems, see, e.g., [7] and [69].

The LQ servo control is an extended version of the LQ (Linear Quadratic) control, which has full state feedback and can follow a reference signal. The LQ provides an optimal feedback rule for a given cost function of the state and input energy so that the input energy can be considered for the control design.

\[ J(\bar{x}, u) = \frac{1}{2} \int_0^\infty (\bar{x}^T Q \bar{x} + u^T R u) \, dt, \]  

(9.2)

where \( Q = Q^T \geq 0 \) and \( R = R^T > 0 \).

The LQ servo includes an additional artificial state with an integrator, which can ensure the tracking capability of the controller. The new state variable to be introduced is obtained as follows:

\[ \dot{z} = e = x_{\text{ref}} - x_p = p_{\text{en, dem}} - p_{\text{en}} = p_{\text{en, dem}} - C_p x. \]  

(9.3)

So the augmented system can be written as follows:

\[ \begin{bmatrix} \dot{x} \\ \dot{z} \end{bmatrix} = \begin{bmatrix} \hat{A} \\ A \\ -C_p \end{bmatrix} \begin{bmatrix} x \\ z \end{bmatrix} + \begin{bmatrix} B \\ 0 \\ I_p \end{bmatrix} u + \begin{bmatrix} 0 \\ 0 \end{bmatrix} x_{\text{ref}}. \]  

(9.4)

Suppose that the augmented system \((\hat{A}, \hat{B}_1)\) is stabilizable and the complete state of the augmented plant can be accurately measured or estimated at all times and is available for feedback.

In this way, the associated optimal control can be obtained as:

\[ u = -[K_x \quad K_z] \begin{bmatrix} x \\ z \end{bmatrix}; \quad [K_x \quad K_z] = R^{-1} \hat{B}_1^T P, \]  

(9.5)

where \( P \geq 0 \) is the unique solution of the Control Algebraic Ricatti Equation (CARE) of

\[ P\hat{A} + \hat{A}^T P - P\hat{B}_1 R^{-1} \hat{B}_1^T P + Q = 0. \]  

(9.6)
Thus the closed-loop augmented system in state space form is obtained as follows [14]:

\[
\begin{bmatrix}
\dot{x} \\
\dot{z}
\end{bmatrix} =
\begin{bmatrix}
A - BK_r & BK_z \\
-C_P & 0
\end{bmatrix}
\begin{bmatrix}
x \\
z
\end{bmatrix} +
\begin{bmatrix}
0 \\
I
\end{bmatrix}
\begin{bmatrix}
x ref.
\end{bmatrix}.
\] (9.7)

The weighting matrices \( R \) and \( Q \) are the tunable parameters of the LQ servo control with appropriate dimensions. Based on Bryson’s law (see in [13]) a suitable initial choice for the elements of the weighting matrices can be obtained that can be tuned to achieve the expected performance of the system. With the weighting matrices and with the Jacobian linearized system described in the previous section the CARE was solved, and the controller was synthesized.

9.3.2 Model inversion feedforward control with LQ servo feedback design

The model inverse can be used as a feedforward control [19]. This model inversion feedforward control can improve tracking performance in case the modeling error is below a certain limit [18].

Although the pressure waves from the exhausting processes of the individual cylinders are in fact non-modeled dynamics of the plant (a mean value model was prescribed), the effect of these pressure waves can be treated as a disturbance on the output of the plant as it is depicted in Fig. 9.4.

![Figure 9.4: General Feedback scheme](image)

The disturbance effect on the plant input should be minimized. Otherwise it can cause extensive control activity and wear of the flap bearings and the motor drive mechanism. The disturbance effect on the input can be defined as follows based on [80]:

\[
u = KS_0d = K(I + L_o)^{-1}d
\] (9.8)

Where \( S_0 \) is the output sensitivity transfer matrix and \( L_o \) is the output loop transfer matrix. It can be seen that with the decreasing of the feedback gain value the disturbance effect can be lowered. The designed controller has to be a tracking controller with a high dynamic response, so a stabilizing controller is needed. It can be realized with full state feedback. For these reasons, the LQ servo controller was
chosen. Based on the resulted linear state-space model two types of controllers were designed: one with only feedback and one supplemented with a feedforward gain to reduce the feedback gain. The weighting matrices were defined and tuned independently so that the feedback gain from the exhaust manifold pressure be smaller in the case of the feedforward and feedback controlled system. The dynamical properties of the two systems were targeted to keep similar. The design methods and the initial choices of the weighting matrices were based on the suggestions in [14] and [7]. The feedforward control is derived from the expression below based on [14]:

\[ K_{ff} = - \left[ C (A - BK)^{-1} B \right]^{-1} \] (9.9)

The designed control structure in case of the feedforward and feedback controlled system is depicted in Fig. 9.5.

In order to accelerate the closed loop system while avoiding the use of high feedback gains (due to the risk of high control activity due to exhaust manifold pressure pulsation) the definition of a non-linear feedforward control term is targeted. It can be derived based on model inversion with the manipulation of the first state equation \((f_1(x,d))\) of Eq. (8.21) considering static condition \((p_{em} = 0)\) and threatened the actual throttle angle as the demand:

\[
\frac{R_{air} \cdot T_{em}}{V_{em}} \left[ \eta_{vol,n_e} \cdot \left[ \frac{V_c + V_d}{V_d} \cdot \left( \frac{p_{em}}{p_{im}} \right) \cdot \frac{V_c}{V_d} \right] \cdot \frac{p_{im}}{R_{air} \cdot T_{im}} \cdot n_e \cdot \frac{V_d}{i} \right] - c_{d,et} \left( A_{et,0} - A_{et,0} \cdot \varphi_{et,dem,ff} + A_{leak} \right) \cdot \frac{p_{em}}{\sqrt{R_{air} \cdot T_{em}}} \cdot \frac{1}{\sqrt{\kappa \left( \frac{2}{\kappa + 1} \right)^{\frac{\kappa+1}{\kappa-1}}}} = 0. \] (9.10)

After some algebraic manipulation the feedforward term can be calculated as:

\[
K_{ff} = \varphi_{et,dem,ff} = \left( \begin{array}{c}
- \eta_{vol,n_e} \cdot \eta_{vol,Ap} \cdot \frac{p_{im}}{R_{air} \cdot T_{im}} \cdot n_e \cdot \frac{V_d}{i} + A_{leak} + A_{et,0} \\
\frac{c_{d,et} \cdot \frac{p_{em}}{\sqrt{R_{air} \cdot T_{em}}} \cdot \sqrt{\kappa \left( \frac{2}{\kappa + 1} \right)^{\frac{\kappa+1}{\kappa-1}}} + A_{et,0}}{A_{et,0}}
\end{array} \right) \] (9.11)

The LQ servo controller above described was augmented with the \(K_{ff}\) feedforward term. The schematic of the controller is depicted in Fig. 9.5.
9.3.3 H-infinity control schematic

To design an H-infinity controller which can handle the nonlinear behavior of the system and guarantee robustness, the LPV state space model is a suitable form. This structure was successfully applied in diesel engine air path modeling see, e.g., [40]. The LPV model conversion is described in the following sub-section.

9.3.3.1 Conversion into linear parameter-varying state space model

H-infinity controller synthesis can be specified with separated actuator dynamics from the controlled plant. Therefore, only the first state equation in Eq. (8.21) was converted into the LPV form.

\[
\frac{dp_{em}}{dt} = \frac{R_{air} \cdot T_{em}}{V_{em}} \cdot \eta_{vol} \cdot \frac{p_{im}}{R_{air} \cdot T_{im}} \cdot n_{e} \cdot \frac{V_{e}}{\sqrt{\frac{2}{\kappa} \cdot \kappa}} - \frac{p_{em}}{\sqrt{R_{air} \cdot T_{em}}} \cdot \sqrt{k} \cdot \left( \frac{2}{\kappa + 1} \right)^{\frac{\kappa - 1}{\kappa}}.
\]

(9.12)

The linear conversion from the flap valve position to the valve area can be calculated by an algebraic function. Seeking the simplest model structure as an input, the throttle area was chosen (\(u = [A_{et}]\)). Review of Eq. (9.12) reveals that only the exhaust manifold pressure enters the state equation in a nonlinear way. Therefore it has been considered as a parameter: \(\varrho = p_{em}\). The boost pressure and the engine speed are measurable disturbance inputs to the plant, which multiply the first part of Eq. (9.12). Therefore they have been defined the product: \(w = [p_{im} \cdot n_{e}]\). So the model was defined in the following form:

\[
\frac{dx}{dt} = A(\varrho)x + B_{1}(\rho)w + B_{2}(\rho)u,
\]

(9.13)

in which the parameter dependent matrices are given as:

\[
A(\varrho) = [0],
\]

(9.14)
\[
B_1(\rho) = \left[ \frac{T_{em}}{V_{em}} \cdot \eta_{vol} \cdot \frac{V_d}{T} \right], \quad (9.15)
\]
\[
B_2(\rho) = \left[ -\frac{R_{air} T_{em}}{V_{em}} \cdot c_d \cdot \rho \cdot \sqrt{T_{em}} \cdot \sqrt{\kappa} \left( \frac{2}{\kappa+1} \right)^{\frac{\kappa}{\kappa+1}} \right]. \quad (9.16)
\]

Note that the \( A \) matrix is equal to zero because the state variable was considered as a parameter, so a quasi-LPV model was specified.

The performance output:
\[
z(t) = C_{11}x + D_{11}(\rho)w + D_{12}(\rho)u. \quad (9.17)
\]

The measured output is:
\[
y(t) = C_{21}x + D_{21}(\rho)w + D_{22}(\rho)u. \quad (9.18)
\]

So the state space matrices become:
\[
C = \begin{bmatrix} 1 \\ 1 \end{bmatrix}; \quad D = \begin{bmatrix} 0 & 0 \\ 0 & 0 \end{bmatrix}. \quad (9.19)
\]

### 9.3.3.2 \( \mathcal{H}_\infty \) controller synthesis

A two degree of freedom closed loop interconnection system was considered for the formulating the \( \mathcal{H}_\infty \) controller synthesis problem based on the suggestions in [70] and [12]. The structure is depicted in Fig. 9.6, where \( r \) denotes the reference input, \( u \) is the control input, and \( z_e \) is the performance output. The structure of the controller includes the \( K_y \) feedback and \( K_r \) pre-filter part, so the controller made up of two parts as \( K = \begin{bmatrix} K_r & K_y \end{bmatrix} \).

![Figure 9.6: \( \mathcal{H}_\infty \) controller design structure](image)

Although the pressure waves from the exhaust processes of the cylinders are in fact non-modeled dynamics of the plant, the effect of these pressure waves can be treated as a disturbance on the output of the plant. Therefore its effect on the control input is parametrized by the controller as \( u = KS_0d = K(1 + PK)^{-1}d \). For details see [80] and [P5]. So the controller must be synthesized with the minimal gain from the \( p_{em} \) to the control input in the frequency region of the
possible EFR. To avoid high control activity and potential actuator and flap mechanism damages the actuator weighting function was chosen as a third order high pass Butterworth filter with a cut off frequency of 30 Hz as follows: $W_u = \frac{10^{-3}}{s^3 + 3 \cdot 10^{-7} \cdot s^2 + 3.11 \cdot 10^{-4} \cdot s + 6.7 \cdot 10^{-6}}$. The required transfer function from the reference to the exhaust manifold pressure has been defined as a first-order transfer function: $T_{ry} = \frac{1}{1/(5 \cdot 2\pi) \cdot s + 1}$. $W_{cmd}$ describes the magnitude and the frequency dependence of the reference command generated by the normalized reference signal $r$. Allowing a step reference input and targeting a maximum backpressure of 5 bars constant transfer function $W_{cmd} = 5 \cdot 10^5$ was defined. The actuator dynamics were treated as a first-order transfer function as defined above: $G_{act} = \frac{1}{0.025 \cdot s + 1}$. The input multiplicative uncertainty $W_{m1} = \frac{0.2s}{s + 188.5}$ and $W_{m2} = 0.003$ model was applied to deal with the unmodeled dynamics (mainly the exhaust pressure oscillation) of the plant, resulting in 20% uncertainty at frequencies above 30 Hz. In order to meet requirements to keep steady-state errors below 0.2 bars the performance output weighting function was defined as $W_e = \frac{1}{0.2 \cdot 10^{10} \cdot 159.15 \cdot s + 1}$. To derive the robust performance of the closed-loop system and synthesize the controller the above structure was reformulated in the LFT form. In this way the so-called $\Delta - P - K$ structure was reached as depicted in Fig. 9.7.

![Figure 9.7: $\Delta - P - K$ structure for controller design](image)

Using the weighting functions of the nominal performance and the robust stability specifications a sub-optimal $H_\infty$ controller was designed. With the 2-Riccati solution method, an 8th order controller was reached with $\gamma = 0.995$. In order to meet the requirements and develop the possible lowest order controller the Hankel singular value-based order reduction procedure was applied. The lowest order controller, which showed no perceptible performance deviance from the initial 8th order controller, was 5th order. Fig. 9.8 shows the resulting controller transfer function from the exhaust manifold signal to the control input. Note that in the critical frequency region of the pressure oscillations (possible EFR frequency region) a rejection occurs.
To reject the effect of the measurable disturbance the following feed-forward control was specified:

\[ u_{ff} = -\frac{B_1(\rho)}{B_2(\rho)}w. \quad (9.20) \]

The controller was implemented with the algebraic linear function

\[ f(A_{ct}) = \varphi_{ct,dem} = -\frac{A_{ct} + A_{ct,0} + A_{act}}{A_{ct,0}}, \quad (9.21) \]

as depicted below. It ensures the conversion from the throttle area to the actuator position demand.

![Bode Diagram](image)

**Figure 9.8:** $H_\infty$ controller transfer function from $p_{em}$ to control input

\[ K_f(A_{ct}) \quad G \]

\[ A_{ct} \quad f(A_{ct}) \quad \varphi_{ct,dem} \]

\[ w \]

\[ r \]

\[ \rightarrow p_{cm} \]

**Figure 9.9:** $H_\infty$ control structure

### 9.3.4 Sliding mode control

As the fourth control structure, a first-order sliding mode controller was designed. The actuator dynamics were neglected to ensure the control simplicity and low computational cost. In this way, only the first row of Eq. (8.21) was taken into consideration, and the equivalent control was defined as $\hat{u} = K_{ff}$ similarly to Eq. (9.11).

The sliding mode control law was formulated based on suggestions in [67] as follows:

\[ u = \hat{u} - k \cdot sgn(s), \quad (9.22) \]
where ‘sgn’ denotes the signum function.

The sliding surface is defined as

\[ s(x, t) = \left( \frac{d}{dt} + \lambda \right)^{n-1} \tilde{x}, \]  

(9.23)

where

\[ \tilde{x} = x_{\text{ref}} - x, \]  

(9.24)

and \( n \) is the model order.

In the case of the described first order simplified model the sliding surface can be written as:

\[ s(x, t) = \lambda \cdot (p_{em} - p_{em, \text{dem}}). \]  

(9.25)

So the control input is:

\[ u = \hat{u} - k \cdot \text{sgn} (\lambda \cdot (p_{em} - p_{em, \text{dem}})). \]  

(9.26)

To reduce high control activity and chattering, caused mainly by exhaust pressure oscillation, the sigmoid function was used instead of signum as below:

\[ u = \hat{u} - k \cdot \frac{2}{1 + e^{-q \cdot (p_{em} - p_{em, \text{dem}})}} - 1. \]  

(9.27)

9.3.5 Saturation handling

In order to handle the saturation of the throttle valve actuator, a high gain anti-windup was applied in each control structure, except the sliding mode controller, which contains no integrator. The anti-windup compensation is provided by subtracting the difference between the actual and the saturated control signals through a high gain matrix to the controller input. For a detailed description and other applications see [44] and [70].

9.4 Test results and discussion

The performance of all four controllers was evaluated with engine dyno measurements in three different tests. Please see Section 8.2 and Appendix A for the detailed description of the experimental setup. These cycles simulate different applications of the exhaust throttle relevant during its operation. The first cycle affects a brake blending application. The engine speed and the backpressure demands send to the engine dyno speed controller, and the backpressure controller comes from a complete vehicle simulation where a medium-duty truck stops during a city cycle. During this process, the engine
speed decreases from almost 2000 RPM to 600 RPM idling, and two downshifts take place. The backpressure demand changes based on the driver’s deceleration demand. The performance comparison of the four backpressure controllers designed above in the same brake blending engine dyno test cycle measurement can be seen in the following figures.

Figure 9.10: Test result of the LQ servo controller in the brake blending cycle

Figure 9.11: Test result of the LQ servo controller with feedforward in the brake blending cycle

The LQ servo controller depicted in Fig. 9.10 has a stable behavior and follows the demand without static error. The disturbing effect of the decreasing engine speed causes no noticeable error in the backpressure. The oscillation in the flap position demand is moderate with
Figure 9.12: H-infinity controller result in the brake blending cycle

a maximum of 1.8% amplitude, but the tracking is relatively slow (see system response for high demand slopes at approx. 2.5 s, 6.25 s, 10.5 and 12 s). As initially expected this can be improved with the use of the feedforward control (see in Fig. 9.11) with conservation of the mentioned favorable properties. The signal tracking of the H-infinity controller (depicted in Fig. 9.12) is the fastest, but it shows oscillations in the control input signal which affect the backpressure as well. The flap position oscillation reaches the 3-4% range. In the test of the sliding mode controller in Fig. 9.13, smooth tracking with low oscillation (max. 1.4%) in the flap position can be observed.

The second cycle simulates on the engine dyno a typical thermo-management operation where after a cold start the engine has to be warmed up as fast as possible. To facilitate this, the engine management applies proper backpressure in the engine exhaust which is realized in several steps in following test measurement:

The LQ controllers show fast and accurate performance without any under- or overshoot in Fig. 9.14. As in the previous test case, the feedforward accelerates the response times from 0.65 s to 0.45 s (see, e.g., pressure step at 1 s in Fig. 9.15). The H-infinity controller is in this case also fast but shows small overshoot (see the step at 4 s in Fig. 9.16 with 0.47 bar overshoot) and noticeable oscillation in the input signal. The sliding mode controller shows fast settling as well, but small over- and undershoots are also observable (note 0.35 bar overshoot after pressure demand step up at 4 s in Fig. 9.17).

The third cycle applies small pressure demand steps and sinusoidal at a close to atmospheric pressure level. It simulates on the dyno test the exhaust throttle support of the HP-EGR recirculation by suitable backpressure generation.
The above figures illustrate that all of the four controllers can follow demand signals even in the subsonic range. Despite the fact that a sonic nominal hybrid mode was chosen in Eq. 8.15 it doesn’t cause notable inaccuracies. Remarkable static errors cannot be noticed but in the case of the sliding mode controller depicted in Fig. 9.21, the lack of the integrator can cause inaccuracies. Especially in the case of worn flap valves. An improvement of the tracking performance of the LQ controller with the feedforward is observable, especially in fact of the sinusoidal demands (compare 9.18 and 9.19 from 4 s). Flap position oscillation in case of the H-infinity controller is also observable at moderate pressure levels in Fig. 9.20.

The compliance with the predefined requirements for each of the investigated controllers is summarized in Table 9.1. The acceptance of values that are below the limit was marked with boldface text. As it reveals, only the LQ servo controller augmented with the model inversion based feedforward was able to fulfill all the predefined requirements. The simple LQ servo shows similarly good results but the rise time achieved is to slow. The computational demand, as well as the overshoot and actuator shaft oscillation of the H-infinity controller, doesn’t meet the required level. The sliding mode controller has a favorable performance. However, its rise time is slower than the limit. As a consequence, the LQ servo controller with the feedforward was chosen as the proposed control structure for the exhaust manifold backpressure control problem for which all the predefined requirements have met.
Figure 9.14: LQ servo controller performance in the thermomanagement cycle

Figure 9.15: LQ servo controller with feedforward in the thermomanagement cycle
Figure 9.16: Thermomanagement test for the H-infinity controller

Figure 9.17: Sliding mode controller in the thermomanagement test
Figure 9.18: EGR support test: LQ servo controller performance

Figure 9.19: EGR support test: LQ servo controller with feedforward performance
Figure 9.20: H-infinity controller in the EGR support test

Figure 9.21: Sliding mode controller result in the EGR support cycle
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<td>±0 bar</td>
<td>±0.08 bar</td>
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<td>0.2 bar</td>
<td>0.12 bar</td>
<td>0.47 bar</td>
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<td>Rise time</td>
<td>0.5 s</td>
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<td>0.45 s</td>
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<td>1.8 %</td>
<td>1.8 %</td>
<td>3.5 %</td>
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<td>Complexity</td>
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<td>+</td>
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SUMMARY

This part described the design of a controller for diesel engine exhaust manifold pressure adjustment with a flap valve. Due to their cost effectiveness exhaust throttles are widely used as endurance brake devices. Possible new applications of backpressure controlled exhaust throttle valves, e.g., brake blending, exhaust gas temperature management, exhaust gas recirculation support, and the need for fast response times were the motivation for this work.

The control aims and requirements were laid down in five points. To ensure applicability on different engines and reducing the calibration effort model-based design was chosen. First of all, a two-state, nonlinear model was described. The model was validated based on test bench measurements showing a reasonable accuracy. The specified model is a mean value model which neglects the effect of the pulsation of the exhaust manifold pressure, caused by the time-shifted exhaust processes of the individual cylinders. The amplitudes of the pressure oscillation are comparable to the expected accuracy, and the oscillation frequency is close above to the system cutoff frequency. For this reason, filtering is not feasible without deterioration of the system response time which was a challenge for the controller design. The LQ servo controller was augmented with a model inversion-based feedforward term, which was led by the requirements of high response time (R5) while the actuator position oscillation remains at a certain limit (R6) to avoid actuator wear. For the LQ controller design, a two-state model was linearized around a typical operation point. A trade-off was achieved between the system response time and the control activity with the appropriate tuning of the feedback gain with the weighting matrices. To avoid integral windup due to actuator saturation and controller integrator, a high gain anti-windup was applied.

The performance of the controllers was tested on an engine dyno. Three different test cases were specified to investigate the controller’s performance in relevant operation fields: a brake blending test cycle with changing engine speed, a thermomangement cycle with medium pressure demand steps and an EGR support test cycle close to atmospheric pressure levels. The proposed LQ servo controller with model inversion based feedforward fulfills all the defined requirements.
Part IV

CYLINDER CHARGE OXYGEN CONCENTRATION CONTROL
11.1 INTRODUCTION

There are several alternative placements of throttle valves in the engine intake and exhaust system, e.g., upstream or downstream to the compressor, etc. Models for EGR rate increase with intake throttling are presented in [75],[59],[24]. Preliminary investigations show concluded in Section 2.2, Chapter 6 and in [P1], that considering the maximum cylinder charge, the most advantageous placements of the throttling is downstream to the turbine. Similar advantages of exhaust side throttling in Low-Pressure Exhaust Gas Recirculation (LP-EGR) systems are shown in [60]. Based on the above results this chapter of this thesis gives a novel proposal for the EGR rate increase through exhaust throttling.

For handling the pollutant formation and the combustion process the precise adjustment of the cylinder charge composition is necessary. To solve this problem (based on the consequences in Section 2.2, Chapter 6, and Chapter 2.2.1) a control method is proposed in which can adapt the intake gas oxygen concentration to a targeted level with the actuation of the EGR valve and if necessary with the exhaust throttle.

In recent years for the Multiple Input Multiple Output (MIMO) control problem of precisely adjusting the air-path parameters (e.g., gas mass flow rate, boost pressure, etc.) with various new actuators (e.g., VGT, bypass valves, etc.) several control approaches were developed. Model Predictive Control (MPC) techniques were successfully applied in [34]. The highly nonlinear model of a two-stage turbocharged diesel engine air path was linearized by feedback in [49]. H-infinity techniques were employed for a three-input-three-output problem in [78]. A hybrid robust air-path control was developed in [76] for diesel engines running conventional and LTC combustion modes.

As it is revealed from the above discussion of air-path controllers a wide variety of control techniques are available for physics-based, mean-value, nonlinear models. Moreover, calibration effort can be minimized on different engines with the use of a model based on first engineering principles. Hence, a model-based controller is targeted. The model described below is the first step toward this aim.

In this chapter, a physics-based, mean-value, nonlinear model will be described. Three balance volumes were chosen: the intake manifold, the exhaust manifold and the volume between the turbine and
the exhaust throttle. The model has five state variables, two control, and two measured disturbance inputs. The model validation was carried out based on engine dyno measurements with a medium-duty diesel engine. The test cycle was taken from the WHTC. In Section 11.2 the engine air path system and the experimental setup will be introduced. The modeling aims and requirements are laid down in Section 11.3 followed by the modeling assumptions in Section 11.4. The model is introduced in detail in Section 11.5 and is converted for the controller design to state space form in Section 11.6. The validation measurements are described in Section 11.8. The air-path mode for the purpose of intake manifold oxygen concentration controller design was published previously by the author in [P9].

11.2 SYSTEM DESCRIPTION AND EXPERIMENTAL SETUP

The model was validated on a common rail, medium-duty commercial vehicle, turbocharged and intercooled diesel engine. The engine was equipped with a cooled HP-EGR system. The exhaust throttle valve was installed directly downstream of the turbine, which provides the minimum volume between the engine exhaust valves and the throttle flap in order to minimize pressure rise times. Relevant parameters of the engine and dynamometer are listed in Section 5.2. The layout of the test engine is shown in Fig 11.1.

![Figure 11.1: Engine layout](image)

The engine was installed on an engine test bench where all the operating parameters of the engine relevant for the system model were measurable. The intake manifold oxygen concentration was measured with a UEGO sensor the signal of which was corrected due to the boost pressure change. A detailed description can be found in Appendix A.

11.3 MODELING AIMS AND REQUIREMENTS

The modeling goals have a major impact on the complexity and the mathematical form of the model. The model developed in this paper
will be the base of an intake manifold oxygen concentration controller. Therefore the following requirements were laid down on the basis of this control-oriented intended usage:

\( R_1 \). The model description must be based on the chemistry, and thermodynamics of the engine air path system and the model variables and parameters must be of physical relevance.

\( R_2 \). A deterministic input-output model must be specified.

\( R_3 \). The model should be restricted to the index-1 model class. That is, the model should be a set of differential algebraic equations (DAEs), where the algebraic equations can be inserted into the differential ones.

\( R_4 \). The model should be represented in state space form.

\( R_5 \). The model should be capable of describing the dynamics of the intake manifold oxygen concentration within a deviation of 10% from the measurement in the whole operation domain of the engine. The accuracy should be evaluated as an \( L_2 \) error.

11.4 SIMPLIFYING ASSUMPTIONS AND INPUT CONSTRAINTS

Simplifying assumptions have been made to reduce model complexity and to reach modeling goals.

\( A_1 \). The potential energy is neglected.

\( A_2 \). Constant physical and chemical properties are assumed over each balance volume of the model, such as specific heat, specific gas constant and adiabatic exponent etc.

\( A_3 \). The adiabatic exponent, the specific gas constant and consequently the specific heat of the air and the exhaust gas are equal, which is a good assumption because diesel engines operate with a lean mixture.

\( A_4 \). There are no mass and energy storage effects in the combustion chamber.

\( A_5 \). The fluids can be modeled as an ideal gas.

\( A_6 \). The temperature of the outflowing gas from the receiver is equal to the receiver’s temperature.

\( A_7 \). The inlet temperature and pressure of the compressor is equal to the ambient temperature and pressure: \( T_{c,in} = T_{amb} \) and \( p_{c,in} = p_{amb} \).

\( A_8 \). The outlet pressure of the turbine is equal to the ambient pressure: \( p_{t,out} = p_{amb} \).
The model described below was derived from the preliminary model published in [P8] with further development, completion and the omission of the pneumatic booster system and in its present form in [P9].

In the engine air path system three balance volumes were chosen: the intake manifold (I.), the exhaust manifold (II.) and the volume between the turbine and the exhaust throttle (III.) (each depicted by dashed lines in Figure 11.1).

### 11.5.1 Equations for the intake manifold balance volume

For the balance volume pressures, the isothermal equation was defined based on the mass conservation and the ideal gas law. The heat loss through the walls can be neglected due to the small temperature differences. The EGR gas temperature does not differ significantly from the air temperature flowing out of the intercooler since it is cooled by the engine coolant. Moreover, it was found in [75] that a dynamic model does not improve the model quality. Therefore the temperature was treated as a constant seeking of simplicity.

The intake manifold has two mass flow inlets from the compressor and the EGR valve and one mass flow outlet to the cylinders, so the differential equation for the pressure state was formulated as follows:

\[ \frac{dp_{im}}{dt} = \frac{R \cdot T_{im}}{V_{im}} \cdot \left[ \sigma_c + \sigma_{egr} - \sigma_{ei} \right]. \quad (11.1) \]

The compressor mass flow rate was computed based on the formula suggested in [4]. The overall efficiency of the turbocharger was treated as a constant which is a good approximation and provides a simple model structure compared to more detailed turbocharger models, e.g., see [54].

\[ \sigma_c = \eta_t \cdot \sigma_t \cdot c_p \cdot T_{em} \cdot \left( \frac{p_{em}}{p_{to}} \frac{\kappa - 1}{\kappa} - 1 \right) \quad (11.2) \]

\[ \frac{R \cdot T_{amb}}{\kappa - 1} \cdot \left( \left( \frac{p_{em}}{p_{amb}} \right)^{\frac{\kappa - 1}{\kappa}} - 1 \right) \]

The engine can be modeled as a positive displacement pump, so the induced gas mass flow rate into the cylinders is formulated in the following form:

\[ \sigma_{ei} = \eta_{vol} \cdot \frac{p_{im}}{R \cdot T_{im}} \cdot n_e \cdot \frac{V_d}{i}. \quad (11.3) \]

The differential equation for the intake manifold oxygen volume fraction can be derived from the definition of the intake manifold air mass fraction:

\[ w_{im,air} = \frac{m_{im,air}}{m_{im}} = \frac{m_{im,air}}{m_{im,air} + m_{im,eg}}. \quad (11.4) \]
After differentiation and manipulation, the following expression resulted for the derivative of the intake manifold air mass fraction:

$$\frac{dw_{im,air}}{dt} = \frac{m_{im,air} (1 - w_{im,air}) - w_{im,air} m_{im}}{m_{im}}. \quad (11.5)$$

Based on the mass conservation law the change of the gas mass in the receiver can be calculated from the in- and outflowing mass flows (8.1, $\frac{dm}{dt}(t) = \sigma_{in}(t) - \sigma_{out}(t)$):

$$\frac{dw_{im,air}}{dt} = \left(\sigma_c - \sigma_{e_{i,air}} + \sigma_{egr,air}\right) (1 - w_{im,air}) - \left(\sigma_{egr,air} - \sigma_{e_{i,mg}}\right) w_{im,air}. \quad (11.6)$$

With using the ideal gas law (8.2, $p(t) \cdot V = m(t) \cdot R \cdot T(t)$) for the denominator and with some manipulation of the numerator the following expression can be written for the intake manifold air mass fraction:

$$\frac{dw_{im,air}}{dt} = \frac{R \cdot T_{im}}{V_{im} p_{im}} \left[\sigma_c (1 - w_{im,air}) + (w_{em,air} - w_{im,air}) \sigma_{egr}\right] \quad (11.7)$$

Considering the relation between the gas mass fraction and volume fraction ($x_i = \frac{w_i}{M_i \bar{M}}$) the oxygen volume fraction of the intake and exhaust manifold can be written as:

$$x_{O_2,im} = w_{im,air} \frac{M_{air}}{M_{O_2}}. \quad (11.8)$$

$$x_{O_2,em} = w_{em,air} \frac{M_{air}}{M_{O_2}}. \quad (11.9)$$

With differentiation we can reach the following form:

$$\dot{x}_{O_2,im} = \dot{w}_{im,air} \frac{M_{air}}{M_{O_2}}. \quad (11.10)$$

Substituting Eq. 11.8, 11.9 and 11.10 to 11.7 the intake manifold oxygen volume fraction can be written as:

$$\frac{dx_{O_2,im}}{dt} = \frac{R \cdot T_{im}}{V_{im} p_{im}} \left[\left(\frac{w_{O_2,air} M_{air}}{M_{O_2}} - x_{O_2,im}\right) \sigma_c + (x_{O_2,em} - x_{O_2,im}) \sigma_{egr}\right]. \quad (11.11)$$

11.5.2 Equations for the exhaust manifold balance volume

The exhaust pressure was defined in the isothermal form where the mass flow from the engine ($\sigma_{e_{o}}$) was defined as the sum of the engine inlet mass flow rate ($\sigma_{e_{i}}$) and the fuel flow rate $\sigma_{fuel}$ provided by the EDC.
\[
\frac{dp_{em}}{dt} = \frac{R \cdot T_{em}}{V_{em}} \cdot [\sigma_{eo} - \sigma_t - \sigma_{egr}] \quad (11.12)
\]

The exhaust temperature cannot be treated as a constant because its value depends on the engine load. Several models for the exhaust manifold temperature have been developed (see, e.g., [22]) but for our goals these models are too complex for use in a control-oriented model. Therefore, the exhaust manifold temperature was calculated from the intake gas and the fuel enthalpy.

\[
T_{em} = \frac{c_p \cdot \sigma_{ei} \cdot T_{im} + \sigma_{fuel} \cdot H_l \cdot K_{eo}}{c_p \cdot \sigma_{eo}} \quad (11.13)
\]

The EGR valve flow is assumed to be subsonic due to low pressure differences between the exhaust and the intake manifold. A hybrid mode is generated by the checkvalve in the EGR loop. If the exhaust gas flow is \( p_{em} > p_{im} \) then the EGR mass flow is calculated as [27]:

\[
\sigma_{egr} = c_{d,egr} \cdot A_{egr} \cdot \frac{p_{em}}{\sqrt{R \cdot T_{em}}} \cdot \sqrt{2 \cdot \frac{p_{im}}{p_{em}}} \cdot \left[ 1 - \left( \frac{p_{im}}{p_{em}} \right)^k_t \right], \quad (11.14)
\]

otherwise:

\[
\sigma_{egr} = 0. \quad (11.15)
\]

Turbines in turbomachinery can be approximated as orifices. Therefore the reduced mass flow rate through the turbine can be defined with the help of a simplified two-parameter model as suggested in [27]. Hence one can obtain the actual mass flow rate as follows:

\[
\sigma_t = \frac{p_{em}}{\sqrt{T_{em}}} \cdot c_t \sqrt{1 - \left( \frac{p_{em}}{p_{to}} \right)^{k_t}}. \quad (11.16)
\]

Similarly to the intake manifold the differential equation for the exhaust manifold oxygen volume fraction can be derived from the definition of the exhaust manifold air mass fraction:

\[
w_{em,air} = \frac{m_{em,air}}{m_{em}} = \frac{m_{em,air}}{m_{em,air} + m_{em,eg}} \quad (11.17)
\]

After differentiation and some manipulation the following expression can be reached:

\[
\frac{dw_{em,air}}{dt} = \frac{m_{em,air} \left( 1 - w_{em,air} \right) - w_{em,air} m_{em,eg}}{m_{em}} \quad (11.18)
\]

Using the mass conservation for the gas in the exhaust manifold (8.1, \( \frac{dm}{dt} (t) = \sigma_{in} (t) - \sigma_{out} (t) \)) and the ideal gas law (8.2 \( p (t) \cdot V = m (t) \cdot R \cdot T (t) \)) the following expression yields for the exhaust manifold air mass fraction:
\[ \frac{dw_{em,air}}{dt} = R \cdot \frac{T_{em}}{V_{em} p_{em}} (w_{eo,air} - w_{em,air}) \sigma_{eo}, \quad (11.19) \]

where the engine outflowing gas air mass fraction is:

\[ w_{eo,air} = \frac{\sigma_{ei} w_{im,air} - \sigma_f K_{L0}}{\sigma_{ei} + \sigma_f} \quad (11.20) \]

Considering the relation between the gas mass fraction and volume fraction \( x_i = \frac{w_i}{\bar{M}} \) the oxygen volume fraction of the exhaust manifold can be written as:

\[ x_{O_2,em} = w_{em,air} \frac{w_{O_2,air} \bar{M}_{air}}{M_{O_2}} \quad (11.21) \]

With differentiation we can reach the following form:

\[ \dot{x}_{O_2,em} = \dot{w}_{em,air} \frac{w_{O_2,a} \bar{M}_{air}}{M_{O_2}} \quad (11.22) \]

Substituting \( 11.22 \) to \( 11.19 \) and \( 11.20 \) the exhaust manifold oxygen volume fraction is formulated as:

\[ \frac{dx_{O_2,em}}{dt} = R \cdot \frac{T_{em}}{V_{em} p_{em}} \cdot [\sigma_{eo} (x_{O_2,e0} - x_{O_2,em})], \quad (11.23) \]

where the oxygen volume fraction of the engine outlet flow can be calculated with the following algebraic equation:

\[ x_{O_2,e0} = \frac{\sigma_{ei} x_{O_2,im} - \sigma_f K_{L0} w_{O_2,air} \bar{M}_{air}}{\sigma_{ei} + \sigma_f \bar{M}_{O_2}}. \quad (11.24) \]

### 11.5.3 Equation for the balance volume between the turbine and the exhaust throttle

The pressure state equation for the third balance volume is

\[ \frac{dp_{to}}{dt} = R \cdot \frac{T_{to}}{V_{to}} \cdot [\sigma_1 - \sigma_{el}], \quad (11.25) \]

where the turbine outlet temperature is obtained as a constant fraction of the turbine inlet temperature as follows:

\[ T_{to} = K_t \cdot T_{em}. \quad (11.26) \]

The mass flow rate through the exhaust throttle remains subsonic during the EGR rate increasing operation. Sonic conditions may occur during exhaust brake operations [P5]. Therefore the mass flow rate through the exhaust throttle can be calculated with the orifice equation for subsonic conditions as follows:
\[ \sigma_{ct} = c_d \cdot A_{ct} \cdot \frac{p_{to}}{\sqrt{R} \cdot T_{to}} \cdot \sqrt{2 \cdot \frac{p_{amb}}{p_{to}} \cdot \left[ 1 - \left( \frac{p_{amb}}{p_{to}} \right) \right].} \] (11.27)

11.6 SYSTEM MODEL IN STATE SPACE FORM

A natural form of system representation for control engineering is the state space form. In such model structures, originating from first engineering principles, state variables are the set of the extensive conserved quantities in the process system. [28]

Most theories and techniques in control system design and analysis both in linear [12] and nonlinear [67] control theory, are based on state space models. Therefore, in accordance with the predefined modeling aims and requirements \( R_4 \) and to facilitate the future controller design the above defined nonlinear model will be converted into state space form.

11.6.1 Definition of model vectors

Three balance volumes are designated in the nonlinear model, and for each one of them one differential equation for the pressure state was defined. For the intake and the exhaust manifolds, an additional differential equation was added for the oxygen concentrations to allow the calculation of the dilution effect of the exhaust gas in the intake manifold. The state vector consists of the states of the intake manifold pressure, the exhaust manifold pressure, the pressure of the volume between the turbine and the exhaust throttle and the exhaust and intake manifold oxygen concentrations as follows:

\[ x = \begin{bmatrix} p_{im} & p_{em} & p_{to} & x_{O_2,im} & x_{O_2,em} \end{bmatrix}^T. \] (11.28)

The input vector contains the EGR valve and the exhaust throttle flow areas:

\[ u = \begin{bmatrix} A_{egr} & A_{et} \end{bmatrix}^T. \] (11.29)

The measurable disturbance vector includes the engine speed and the fuel mass flow respectively:

\[ d = \begin{bmatrix} n_e & \sigma_f \end{bmatrix}^T. \] (11.30)

11.6.2 The state equations

The sonic flow condition of the EGR valve and the exhaust throttle and the EGR checkvalve adds hybrid modes to the model. However,
in order to arrive at the simplest form of the model the state space equation will be given for the nominal hybrid mode. After inserting the constitutive equations into the differential conservation balances, the state-space model can be obtained in an input-affine form.

\[
\begin{bmatrix}
\dot{p}_{im} \\
\dot{p}_{em} \\
\dot{p}_{to} \\
\dot{x}_{O_2,im} \\
\dot{x}_{O_2,em}
\end{bmatrix} = \begin{bmatrix}
f_1(x, d, r) \\
f_2(x, d, r) \\
f_3(x, d, r) \\
f_4(x, d, r) \\
f_5(x, d, r)
\end{bmatrix} + \begin{bmatrix}
g_{11}(x, d, r) & 0 \\
g_{21}(x, d, r) & 0 \\
0 & g_{32}(x, d, r) \\
g_{41}(x, d, r) & 0 \\
0 & 0
\end{bmatrix} u
\]

(11.31)

where \( r : \mathbb{R}^n \rightarrow \mathbb{N} \) is a piecewise constant switching function mapping from the state space to \( \mathbb{N} \). The integer set \( \mathbb{N} \) is finite, i.e. \( \mathbb{N} = 1, 2, ..., n \), where \( n = \prod_{i=1}^{n_i} n_i \) is the total number of hybrid modes and \( n_i \) is the number of the individual hybrid modes of a subsystem \( (n = 2 \times 3 = 6) \). Let the value of the switching function mapping \( r \) be 1 for the nominal hybrid mode.

The nonlinear state functions with all constitutive relations inserted for the nominal hybrid mode. The elements of the first state equation are given as:

\[
f_1(x, d, 1) = \frac{R \cdot T_{im}}{V_{im}} \left( \frac{p_{em}}{p_{to}} - 1 \right) \eta_1 \cdot p_{em} \cdot c_t \cdot \sqrt{\frac{1 - \left( \frac{p_{em}}{p_{to}} \right)^{k_t}}{\eta_p \cdot \eta_t \cdot \frac{V_{im}}{p_{im}} \cdot n_e \cdot n_r \cdot T_{im} + T_{im + \sigma_f} \cdot H_f \cdot K_o}} \frac{\sqrt{1 - \eta_v \cdot \frac{p_{im}}{p_{amb}}}}{R \cdot T_{amb} \cdot \left( \frac{p_{im}}{p_{amb}} \right)^{\frac{k_t}{k_t - 1}} - 1} - \eta_v \cdot \frac{p_{im}}{V_{im}} \cdot \frac{V_d}{\eta_p}
\]

(11.32)

\[
g_{11}(x, d, 1) = \frac{T_{im} \cdot c_d \cdot p_{em}}{V_{im}} \left( \frac{R \cdot 2 \cdot p_{im}}{p_{em}} \cdot \left[ 1 - \left( \frac{p_{im}}{p_{em}} \right) \right] \frac{\sqrt{\frac{1 - \eta_v \cdot \frac{p_{im}}{p_{amb}}}{\eta_p \cdot \frac{V_{im}}{p_{im}} \cdot n_e \cdot n_r \cdot T_{im} + T_{im + \sigma_f} \cdot H_f \cdot K_o}}}{c_p \cdot \eta_v \cdot \frac{p_{im}}{p_{amb}} \cdot n_e \cdot n_r \cdot T_{im} + T_{im + \sigma_f} \cdot H_f \cdot K_o} \right) \frac{\sqrt{\frac{1 - \eta_v \cdot \frac{p_{im}}{p_{amb}}}{\eta_p \cdot \frac{V_{im}}{p_{im}} \cdot n_e \cdot n_r \cdot T_{im} + T_{im + \sigma_f} \cdot H_f \cdot K_o}}}{c_p \cdot \eta_v \cdot \frac{p_{im}}{p_{amb}} \cdot n_e \cdot n_r \cdot T_{im} + T_{im + \sigma_f} \cdot H_f \cdot K_o}
\]

(11.33)

The elements of the second state equation are:
The elements of the third state equation are:

\[
f_3(x, d, 1) = \frac{R \cdot K_l \cdot \frac{1}{cp} \cdot \frac{p_{im}}{\eta_0 \cdot \frac{p_{em}}{\pi_{im}} \cdot n_c \cdot \frac{V_d}{i} + \sigma_f}}{V_{Io}} \cdot \left[ \frac{1 - \left( \frac{p_{em}}{p_{Io}} \right)^{k_l}}{\left( \frac{p_{em}}{p_{Io}} \right)^{k_l}} \right] \cdot \frac{p_{em} \cdot c_t}{c_p \cdot \left( \eta_0 \cdot \frac{p_{em}}{\pi_{im}} \cdot n_c \cdot \frac{V_d}{i} + \sigma_f \right)}.
\]

(11.36)

The elements of the fourth state equation are:

\[
f_4(x, d, 1) = \frac{R \cdot \frac{T_{im}}{V_{im}} \cdot \left( w_{O_2, air} \cdot M_{air} - x_{O_2, im} \right)}{V_{im} \cdot p_{im}} \cdot \eta_t \cdot p_{em} \cdot c_t \cdot \frac{1 - \left( \frac{p_{em}}{p_{Io}} \right)^{k_l} \cdot c_p \cdot \left( \frac{p_{em} \cdot \frac{V_d}{i} + \sigma_f}{p_{Io}} \right)}{\left( \frac{p_{em}}{p_{Io}} \right)^{k_l} - 1} \cdot \frac{R \cdot \frac{T_{amb}}{x_{T_{amb}}} \cdot \left( \frac{p_{im}}{p_{amb}} \right)^{\frac{k_l}{x_{T_{amb}}}}}{c_p \cdot \left( \eta_0 \cdot \frac{p_{im}}{\pi_{im}} \cdot n_c \cdot \frac{V_d}{i} + \sigma_f \right)}.
\]

(11.38)
\[
g_{41}(x, d, 1) = \frac{T_{im}}{V_{im} \cdot p_{im}} \cdot (x_{O_2, em} - x_{O_2, im}) \cdot c_{d, egr} \cdot p_{em} \cdot \\
\frac{\frac{R}{2} \cdot \frac{p_{im}}{p_{em}} \cdot \left[ 1 - \left( \frac{p_{im}}{p_{em}} \right) \right]}{c_{p} \left( \eta_{v} \cdot \frac{p_{im}}{p_{em}} \cdot n_{e} \cdot V_{d} \right) + \sigma_{f}}. \quad (11.39)
\]

The elements of the fifth state equation are:

\[
f_{5}(x, d, 1) = \frac{R \cdot \left( \eta_{v} \cdot \frac{p_{im}}{p_{em}} \cdot n_{e} \cdot V_{d} \cdot \sigma_{f} + \sigma_{j} \right)}{V_{em} \cdot p_{em}} \cdot \left[ \left( \eta_{v} \cdot \frac{p_{im}}{R \cdot T_{im}} \cdot n_{e} \cdot \frac{V_{d}}{i} + \sigma_{f} \right) \cdot \right. \\
\left. \left( \eta_{v} \cdot \frac{p_{im}}{R \cdot T_{im}} \cdot n_{e} \cdot \frac{V_{d}}{i} \cdot x_{O_2, im} - \sigma_{f} \cdot K_{L0} \cdot \frac{\overline{\rho_{im}}}{\overline{M_{i2}}} - \\
\eta_{v} \cdot \frac{p_{im}}{R \cdot T_{im}} \cdot n_{e} \cdot \frac{V_{d}}{i} + \sigma_{f} \right) \right]. \quad (11.40)
\]

11.6.3 The measured output and the performance equations

Since the output is linear with respect to the state vector, the measured output is written as the following linear equation

\[
y = Cx + D_{11}d + D_{12}u, \quad (11.41)
\]

where the \( D_{11} \in \mathbb{R}^{5 \times 2} \) and \( D_{12} \in \mathbb{R}^{5 \times 2} \) are zero matrices. \( C \) is determined by the application circumstances of the model. For example, on the test bed used all the state variables were measured directly, therefore \( C \in \mathbb{R}^{5 \times 5} \) can be equal to the identity matrix. For vehicle applications of the proposed intake manifold oxygen controller, some sensors should be substituted by observers to provide a cost-effective solution.

The performance output is the intake manifold oxygen concentration based on the modeling aim, and it is generated from the measured output as:

\[
z = \begin{bmatrix} 0 & 0 & 0 & 1 & 0 \end{bmatrix} y. \quad (11.42)
\]

11.7 Intake Manifold Oxygen Model Parameter Identification

There are 20 parameters in the above-described model that need to be identified. A similar identification process that was presented in Section 8.4 can also be used in case of the intake manifold oxygen concentration model. There are parameters which value are defined by
specific theories or based on modeling assumption defined in Section 11.4.

- $V_c$ and $V_d$ can be found in the engine datasheet, see 5.1.

- The whole intake and exhaust system were modeled in a 3D CAD system so the volume of the intake and exhaust manifold, the volume between the turbine and the exhaust throttle could be measured directly, see 5.3.

- The material properties of the gas used as working media (air mean molar mass, air oxygen mass fraction, oxygen molar mass) can be specified on catalog data.

- The adiabatic exponent and specific gas constant can be defined as those of the air since the engine operates mostly with a lean mixture.

- The ambient condition defined as standard air conditions.

- Due to the usage of an intercooler, the intake manifold temperature can be approximated well with a constant value.

- The diesel fuel lower heating value, and stoichiometric air consumption values were defined based on catalog data.

The other parts of the parameters to be identified are unknown static parameters. These parameter values can be estimated based on measurements taken in static operation points of the engine. For this purpose, the least-square fitting method was used for expressions which are linear in parameters.

- The engine volumetric efficiency was identified from intake mass flow measurements with the least-squares method similarly to the exhaust flap discharge coefficient.

- The same method could be used for fitting the turbine throttle characteristics and efficiency.

- With a known engine air consumption model, the mass flow of the recirculated exhaust gases can be calculated so the discharge coefficient of the EGR valve can also be identified.

- The parameter value for the exhaust manifold temperature can be estimated by using 11.13 for least-squares fitting.

- Turbine mass flow model parameters were estimated with a numeric optimization (simplex) method.

The resulted parameters are listed in Table 11.1.
Table 11.1: List of parameters

<table>
<thead>
<tr>
<th>Parameter name</th>
<th>Symbol</th>
<th>Value</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>Adiabatic exponent</td>
<td>(\kappa)</td>
<td>1.4</td>
<td>-</td>
</tr>
<tr>
<td>Air mean molar mass</td>
<td>(\overline{M}_{\text{air}})</td>
<td>28.96</td>
<td>kg/kmol</td>
</tr>
<tr>
<td>Air oxygen mass fraction</td>
<td>(w_{O_2,\text{air}})</td>
<td>0.232</td>
<td>-</td>
</tr>
<tr>
<td>Ambient temperature</td>
<td>(T_{\text{amb}})</td>
<td>300</td>
<td>K</td>
</tr>
<tr>
<td>Ambient pressure</td>
<td>(p_{\text{amb}})</td>
<td>(10^5)</td>
<td>Pa</td>
</tr>
<tr>
<td>Diesel lower heating value</td>
<td>(H_l)</td>
<td>43</td>
<td>MJ/kg</td>
</tr>
<tr>
<td>Diesel stoichiometric air consumption</td>
<td>(K_{L0})</td>
<td>14.5</td>
<td>kg/kg</td>
</tr>
<tr>
<td>EGR valve discharge coefficient</td>
<td>(c_{d,\text{egr}})</td>
<td>0.195</td>
<td>-</td>
</tr>
<tr>
<td>Engine displacement</td>
<td>(V_d)</td>
<td>0.003922</td>
<td>m³</td>
</tr>
<tr>
<td>Engine volumetric efficiency</td>
<td>(\eta_v)</td>
<td>0.82</td>
<td>-</td>
</tr>
<tr>
<td>Exhaust manifold volume</td>
<td>(V_{\text{em}})</td>
<td>0.0051</td>
<td>m³</td>
</tr>
<tr>
<td>Exhaust throttle discharge coefficient</td>
<td>(c_{d,\text{et}})</td>
<td>0.501</td>
<td>-</td>
</tr>
<tr>
<td>Intake manifold temperature</td>
<td>(T_{\text{im}})</td>
<td>315</td>
<td>K</td>
</tr>
<tr>
<td>Intake manifold volume</td>
<td>(V_{\text{im}})</td>
<td>0.0133</td>
<td>m³</td>
</tr>
<tr>
<td>Number of revolutions per cycle</td>
<td>(i)</td>
<td>2</td>
<td>-</td>
</tr>
<tr>
<td>Oxygen molar mass</td>
<td>(M_{O_2})</td>
<td>32</td>
<td>kg/kmol</td>
</tr>
<tr>
<td>The parameter for engine-out temperature</td>
<td>(K_{\text{co}})</td>
<td>0.25</td>
<td>-</td>
</tr>
<tr>
<td>Specific gas constant</td>
<td>(R)</td>
<td>287</td>
<td>J/kg K</td>
</tr>
<tr>
<td>Turbocharger efficiency</td>
<td>(\eta_t)</td>
<td>0.273</td>
<td>-</td>
</tr>
<tr>
<td>Turbine mass flow model parameter 1</td>
<td>(k_t)</td>
<td>-0.48</td>
<td>-</td>
</tr>
<tr>
<td>Turbine mass flow model parameter 2</td>
<td>(c_t)</td>
<td>2.74 (\times 10^{-5})</td>
<td>-</td>
</tr>
<tr>
<td>The volume between exh. manifold and throttle</td>
<td>(V_{\text{to}})</td>
<td>0.0045</td>
<td>m³</td>
</tr>
</tbody>
</table>

11.8 Model Validation

The model verification and validation were performed in MATLAB/Simulink environment.

The model performance was evaluated in the EU legally prescribed WHTC. For comparison, a section of the cycle from 990 to 1150 seconds is shown. The engine operation points of the complete WHTC cycle for the investigated engine are depicted in Fig. 11.2. The motoring points were measured and depicted with zero loads because the applied eddy current brake cannot produce negative torques. It can be seen that engine speeds, over 1500 RPM are quite rare. In these medium engine speeds, the boost pressure is often higher than the exhaust manifold pressure without exhaust throttling. A high number of low load points can also be observed. Consequently, the test cycle provides the opportunity to take advantage of the EGR rate increase by the exhaust throttling, and the low air-fuel ratio does not limit its utilization in low load cases. The EGR valve and exhaust throttle areas were actuated based on predefined lookup tables. The EGR valve
is fully opened at low loads and gradually closes at high loads. The exhaust throttle is nearly closed at low loads and low speeds and gradually opens with increasing load and speed.

A good fit of the modeled signals can be observed. The intake and exhaust manifold signals follow the measurement with a delay. This is caused by the unmodeled dynamics of the turbocharger which is calculated by algebraic equations. The inclusion of the turbocharger dynamics in the differential equations would lead to unacceptably complex model structures and a large number of model states which is contradictory to the control-oriented application aim. Based on the fitting of the performance variable is desired. To show the accuracy of the model the deviation of the intake manifold oxygen concentration was evaluated as RMS errors in the validation cycle based on the following expression:

$$
\varepsilon_{x_{im}} = \sqrt{\frac{1}{T_{cycle}} \int_{0}^{T_{cycle}} (x_{im,m} - x_{im})^2 \, dt} \quad (11.43)
$$

The duration of the entire cycle was 160 seconds, and the suffix ‘m’ denotes the measured value. The compliance of the predefined requirements in Section 11.2 is summarized in Table 11.2.
All the predefined requirements have been met. Therefore the model can be used for the intended intake manifold oxygen concentration controller design.
12

CONTROLLER DESIGN AND MEASUREMENT
RESULTS

12.1 TEST ENGINE AND MEASUREMENT SETUP

The controller was tuned and tested on the test engine installed on a dynamometer described in Section 11.2. For EGR duct area control a position controlled mechatronic flap valve was installed on the hot side. Moreover, a position controlled mechatronic exhaust flap valve was installed downstream of the turbine. A UEGO sensor was installed in the intake manifold which signal was processed with a dSpace RapidPro unit. The pressure of the oxygen concentration was also performed by this unit. The control software was implemented on a dSpace MicroAutoBox rapid control prototyping unit. The measurement data acquisition and the control of the dynamometer were also performed by this unit. The complete layout is depicted in Fig. 12.1. The detailed instrumentation setup is described in Appendix A.

![Figure 12.1: Control system layout](image)

12.2 CONTROL AIMS AND STRATEGY

The intake manifold oxygen concentration controller presented below was published previously by the author in [Pt10]. The goal of the designed control system is to track the intake manifold oxygen concentration demand to provide low emission (especially low PM and NOx) while minimizing the fuel consumption. The choose of the performance output was defined based on Chapter 2.2.1 where the
advantages of intake manifold oxygen concentration over EGR rate based control was evaluated. The control actuators were chosen as the EGR valve and throttling downstream the turbine according to the defined optimal engine air-path setup defined in Chapter 6. The increase of the exhaust backpressure leads to an increased PMEP and consequently to higher fuel consumption. The minimal CO₂ emission of the engine need to be also ensured while controlling the cylinder gas charge.

Therefore, the following control aims were defined:

\( \mathcal{R}_1 \). The intake manifold oxygen concentration \((x_{im})\) shall track the intake manifold oxygen concentration demand \((x_{im,dem})\) signal. Too high oxygen concentration would lead to increased NO\(_x\) emission of while too low would result in PM high emission.

\( \mathcal{R}_2 \). The pumping mean effective pressure of the engine \((p_{im} - p_{em}, the difference between the intake and exhaust manifold pressure levels) need to be minimized targeting a minimal engine fuel consumption.

\( \mathcal{R}_3 \). The controller must ensure the asymptotic stability of the system.

\( \mathcal{R}_4 \). The complexity of the control algorithm should be low enough to allow its application in an embedded environment, where the clock rate of the applied single core processor is in the range of 40-50 kHz and the available memory is below 64 kbyte.

A control strategy and controller which can fulfill the above requirements are targeted.

### 12.3 Controller Design

LQ servo controllers were widely used successfully for numerous nonlinear control problems, see, e.g., [7, 69]. Moreover, their low computational demand makes them attractive for embedded implementation which means it can fulfill \( \mathcal{R}_4 \). The LQ servo controller is a full state feedback linear quadratic regulator augmented with an additional artificial integral state. Hence, it can track a reference signal. Therefore, it can satisfy \( \mathcal{R}_3 \) and \( \mathcal{R}_4 \). In case of the engine backpressure control problem a comparison was made with engine dynamometer measurement in three different test cases between the LQ servo, the H-infinity and the sliding mode controllers (please see in details in Chapter 9). This comparison covers the main fields of the control theory (optimal, robust and nonlinear control theory) with the applied controllers that could be implemented in the targeted embedded environment (see \( \mathcal{R}_4 \)). The result of this investigation shows that the performance and fulfillment of the predefined requirements were
only achievable with the LQ servo method and the robust and nonlinear controllers couldn’t show noticeable advantaged above it. Therefore, for the backpressure control problem, the LQ servo was chosen as the proposed control. As a consequence, for the intake manifold concentration control problem as a similar system to be controlled only the LQ servo controller synthesis was targeted only.

The artificial state, in this case, is the error signal of the intake manifold oxygen concentration: \( e = x_{\text{im, dem}} - x_{\text{im}} \). The LQ servo control uses full state feedback which minimizes the following cost function of the control input and states energy.

\[
J(\bar{x}, u) = \frac{1}{2} \int_0^\infty (\bar{x}^T Q \bar{x} + u^T R u) \, dt, \tag{12.1}
\]

where \( Q = Q^T \geq 0 \) and \( R = R^T > 0 \).

The weighting matrices \( R \) and \( Q \) are the tunable parameters of the LQ servo control with appropriate dimensions. Based on [13] a suitable initial choice for the elements of the weighting matrices is:

\[
Q = \begin{bmatrix} q_1 & \cdots & q_n \end{bmatrix}, \quad R = \begin{bmatrix} r_1 & \cdots & r_m \end{bmatrix}, \tag{12.2}
\]

\[
q_i = \frac{1}{t_{si} (x_{\text{imax}})}, \quad r_i = \frac{1}{(u_{\text{imax}})}, \quad \varrho > 0, \tag{12.3}
\]

where

- \( t_{si} \) is the desired settling time of \( x_i \)
- \( x_{\text{imax}} \) is a constraint on \( |x_i| \)
- \( u_{\text{imax}} \) is a constraint on \( |u_i| \)
- \( \varrho \) is chosen to trade-off regulation versus control effort.

It can be clearly seen that the actuation of the exhaust throttle could increase the PMEP of the engine. Therefore, while the appropriate dilution of the cylinder charge is possible with the opening of the EGR valve only the exhaust flap needs to hold a fully open position. The closure of the exhaust flap is only recommended when the EGR valve is saturated in its wide open position. In Section 11.6 a MIMO state space model was defined that can be the basis for the controller design. In this context, the above control strategy can be satisfied with the separation of the MIMO system into two Single Input Single Output (SISO) system: the first aims to track the intake manifold oxygen concentration demand with the control input of the EGR valve only.
with a constant fully opened exhaust flap. The other controller operates with the exhaust flap opening only with fully opened EGR valve. Hence, the state-space model for the SISO case of EGR valve control input only with fully opened exhaust throttle can be written as follows (with notations of Section 11.6):

\[
\begin{bmatrix}
\dot{p}_{im} \\
\dot{p}_{em} \\
\dot{p}_{to} \\
\dot{x}_{O_2,im} \\
\dot{x}_{O_2,em}
\end{bmatrix}
= \begin{bmatrix}
f_1(x, d) \\
f_2(x, d) \\
f_3(x, d) + g_{32}(x, d)A_{et,max} \\
f_4(x, d) \\
f_5(x, d)
\end{bmatrix}
+ \begin{bmatrix}
g_{11}(x, d) \\
g_{21}(x, d) \\
g_{31}(x, d) \\
g_{41}(x, d) \\
g_{51}(x, d)
\end{bmatrix} A_{egr}
\]

(12.4)

and for the SISO case of exhaust throttle control input with wide opened EGR valve:

\[
\begin{bmatrix}
\dot{p}_{im} \\
\dot{p}_{em} \\
\dot{p}_{to} \\
\dot{x}_{O_2,im} \\
\dot{x}_{O_2,em}
\end{bmatrix}
= \begin{bmatrix}
f_1(x, d) + g_{11}(x, d)A_{egr,max} \\
f_2(x, d) + g_{21}(x, d)A_{egr,max} \\
f_3(x, d) \\
f_4(x, d) + g_{41}(x, d)A_{egr,max} \\
f_5(x, d)
\end{bmatrix}
+ \begin{bmatrix}
0 \\
0 \\
0 \\
0 \\
0
\end{bmatrix} A_{et}.
\]

(12.5)

Between the two controllers, the following switching logic was implemented: initially the controller with the control input of the EGR valve is actuated. If its EGR flap valve position demand is reached the 90% position, then, the switching logic actuates the other controller which operates with the exhaust throttling while holding a constant wide open EGR valve position. Switching back is occurring when the position control demand of the exhaust flap drops to 50%. Rooting from the nonlinear behavior of the system the exhaust flap valve has an only marginal effect below the 70% position. Control system stability after controller switching is targeted to be investigated with dyno measurements that are presented below.

The state-space model 12.4 and 12.5 were Jacobian linearized respectively around the operation point of 1250 RPM engine speed and moderate load equilibrium point. With the above-defined weighting matrices and linear SISO state-space systems, the CARE was solved, and the controllers were synthesized in discrete time. The two controllers were implemented in MATLAB/Simulink environment with a sampling time of 1 ms. All the states were measured directly (see Appendix A for sensor configuration).
After implementation of the above control structure in MATLAB/Simulink environment, it was run on a dSpace MicroAutoBox rapid prototyping hardware. The communication both with the actuators, engine dynam system, and sensors was performed via CAN.

To cover the engine operation field relevant to the WHTC cycle (see in Fig. 5.15) there were three stationary measurements taken at 1000 RPM, 1250 RPM, and 1500 RPM. The engine load was 100 Nm at 1000 RPM, 200 Nm at 1250 RPM and 300 Nm at 1500 RPM. Higher loads are not typically suitable for the operation of the intake manifold oxygen concentration controller since the low level of the cylinder charge oxygen content would not allow high fuel injection or at least cause reduced air-to-fuel ratios and consequently high particulate matter emission. All the three measurements start with 21 % cylinder charge oxygen concentration demand as a reference. During this initial period, the EGR valve is fully closed, and the exhaust flap is wide opened. After it, steps were performed with -1 percent increments down until 17% oxygen content.

Fig. 12.2 depicts the measurement results at 1000 RPM with 100 Nm load. The first subplot shows the in-cylinder gas oxygen concentration demand and the actual measured value. The second and the third subplot shows the actual exhaust flap and EGR valve positions, respectively in percentages. 100 % means the fully closed position in case of the exhaust throttle and fully opened position in case of the EGR valve. As it can be seen, the 18 % oxygen concentration is achievable with the opening of the EGR valve only with the position of approx. 80 %. Therefore, the exhaust flap remained wide open. However, the further opening of the EGR valve makes not possible the realization of the 17 % demand, the EGR valve saturates. Switching takes place between the to above designed LQ servo controller and the closure of exhaust throttle ensure the further increase of the exhaust gas backflow. The system remained stable after the switching. The demand is followed accurately during the whole test case. As a result, the raw nitrogen-oxide emission decreased from about 1350 ppm to 350 ppm which means a 64 % reduction. Please note that in step from 18 % to 17 % the NO\textsubscript{x} decreased from 550 ppm to 350 ppm (approx. 37 %) thanks to the actuation of the exhaust throttle. There is no noticeable change in fuel consumption during the whole cycle. During the reference no-EGR period the engine pumping work is negative (the exhaust manifold pressure is higher than the intake). The opening of the EGR duct equalizes the pressures between the manifold and as a consequence, the EGR reduces the engine pumping work and consequently increases the engine effective efficiency. Even the lowest engine out oxygen content is high enough (9.5 %), so
it does not lower the engine efficiency. Moreover, at such high air-to-fuel ratios the formation of PM is probably low.

Fig. 12.2: Cylinder charge control at 1000RPM and 100 Nm

Fig. 12.3 demonstrates the controller performance in the same step response test demand cycle but the engine speed was increased to 1250 RPM and the load to 200 Nm. The intake manifold oxygen concentration was decreased in 1 % steps to 17 % and finally settles again on 21 %. It can be seen that the 18 % oxygen concentration can be achieved by opening the EGR valve only similarly to the above operation point. Although, to reach the 17 % oxygen concentration the naturally evolved differential pressure conditions between the intake and exhaust manifold are not enough. Hence, the switching to the controller with the exhaust throttle was done while the control system remained stable. Meanwhile, the EGR valve was held continuously on a fully opened position. It reveals that the actual intake manifold concentration tracks within a 0.1% range the demand. Please note that
caused by the nonlinear behavior of the system the even lower oxygen concentration levels can be reached an even smaller modification of the exhaust throttle position. In the fourth subplot line the raw $NO_x$ emission can be seen measured by a $NO_x$ sensor. The designed control structure decreases the exhaust gas $NO_x$ concentration from the initial 1716 ppm to 346 ppm, as expected, correlating with the cylinder charge gas composition. Notably, the application of the exhaust throttling resulted in a decrease from 580 ppm to 346 ppm. The exhaust backpressure decreased in this case too. However, the boost pressure is also less than in the reference no-EGR case. Please note that the engine backpressure is further reduced as a consequence of the exhaust throttling which is caused by the reduction of the turbine power. At this higher load, a slight increase (from 1.69 g/s to 1.75 g/s which means 3.5 %) can be observed due to the reduction of the air-to-fuel ratio.
In Fig. 12.4 the result of the test cycle is depicted at the highest engine speed (1500 RPM) and load (300 Nm). As above the opening of the EGR valve only is enough to reach 18% cylinder charge oxygen content but for the further dilution, the support of the exhaust flap is needed. Again, the control system remained stable during the switching. The behavior of the engine system is similar to the above cases: the pressures difference over the intake and exhaust sides equalizes (although the boost pressure decreases) and the NO\textsubscript{x} emission decreasing significantly (from 482 ppm to 90 ppm). The rise of the fuel consumption is here the highest amongst the test cases (from 3.15 g/s to 3.85 g/s, which means approx. 22%). It occurred due to the severe decrease of the combustion air-to-fuel ratio that reaches even fuel rich region. It resulted in low engine efficiency. It reveals the rec-
ommended operating range of the proposed cylinder charge control structure: at low loads where the combustion air-to-fuel ratios are high the dilution of the cylinder charge reduces the $NO_x$ emission significantly but not affects the fuel consumption. Moreover, at such high air-to-fuel ratio there is probably no significant $PM$ emission occurs.

Figure 12.4: Intake manifold oxygen control test result at 1500RPM and 300 Nm
This part of the thesis deals with the development of the cylinder charge control function as it was targeted in Section 2.2. As a first step towards this aim, a novel performance output was chosen which is the intake manifold oxygen concentration Chapter 2.2.1. State of the art control solutions usually uses the EGR rate tracking as a target. It was revealed that the intake manifold oxygen concentration gives a more proper measure about the gas composition in the cylinder charge and, therefore, it correlates tighter with the engine emission. Hence, the use of the intake manifold oxygen concentration as a performance output is more beneficial.

After it, a physics-based, nonlinear, control-oriented model of the whole engine air-path was worked out as a basis for the controller design. The modeled engine setup uses an HP-EGR valve and a throttle flap as it was suggested in Chapter 6, as an optimal engine air-path actuator setup. Three balance volumes were chosen for the definition of the conservation equations, and consequently, the model has five state variables: the pressure of the intake and exhaust manifold and those of the volume between the turbine and the exhaust flap, moreover the oxygen concentration of the intake and exhaust manifold. The performance output was defined as the intake manifold oxygen concentration as deduced above. The model was validated with measurements with an engine dyno which showed an accuracy below of 10%. Therefore, it was accepted for further work. The model was converted into an input-affine state space form to serve as a basis of the controller design in Chapter 12.

Finally, an LQ servo control structure was developed which can ensure the tracking of the intake manifold oxygen concentration demand signal with the intervention of the HP-EGR valve and the exhaust throttle downstream the turbine. The controller design method was chosen based on the predefined control aims (see Section 12.2) and the consequences of the backpressure controller comparison tests in Section 9.4. The control structure needs to minimize the pumping losses to ensure maximum engine efficiency. It can be guaranteed only if the exhaust flap starts to close after the EGR valve saturation. For this purpose, the control model was converted into two separate SISO form: one in which only the EGR valve opening is the control input and an other where the exhaust flap. The models where linearized around a typical equilibrium point and two LQ servo controller was synthesized, and a switching logic was implemented between them.
In the end, the controller performance was demonstrated with engine dyno measurements in a test sequence where cylinder charge oxygen concentration was decreased in 1% steps to 17% and finally settled again on 21%. The analysis was done at three different engine speeds and engine loads (at 1000 RPM and 100 Nm, 1250 RPM and 200 Nm, 1500 RPM and 300 Nm) to cover the relevant engine operation range in the WHTC emission measurement cycle (see in Fig. 5.15). The proposed control structure ensured a stable and accurate tracking of the demand in each of the test cases. As a result, the reduction of the NO\textsubscript{x} emission was significant. The opening of the EGR duct equalized the pressure difference between the intake and exhaust sides the engine, thus lowered the engine pumping losses. Moreover, the boost pressure decreased with higher exhaust gas backflow rates. At the 100 Nm load case there was no rise observable in fuel consumption. This can be justified by the fact that the air-to-fuel ratio is high enough at 17% intake manifold oxygen concentrations and the opening of the EGR valve reduces the engine pumping losses. At higher engine loads the adverse effect on the fuel consumption increases while the mixture is getting richer. Although, the NO\textsubscript{x} reducing the potential of the proposed novel cylinder charge controller remains appealing.

From the above test results, it can be concluded that the operation of the control structure developed in this part of the thesis is advantageous and hence recommended in the below 200 Nm engine load range for this particular test engine. It shows good cooperation with aftertreatment systems: at low engine loads while the exhaust gas temperature is low and the efficiency of the SCR catalyst is low the proposed controller can be used for the emission limitation. At high loads and high exhaust gas temperatures, the aftertreatment system can work with good efficiency.
THESSES

The main contributions and the proposed theses of this work are summarized below. The relevant chapter of the thesis and the labels of the related publications (enumerated in Chapter Publications) are indicated in parenthesis.

THESES 1 Definition of new engine functions achievable with a flow area control in the engine air-path and optimal diesel engine air-path flap actuator setup for the realization (Part i and ii), [P1, P2, P3, P4].

New commercial vehicle diesel engine functions were described which could be achieved by suitable actuation of flap valves in different locations of the engine air-path system. These include brake blending, automated manual transmission support, exhaust gas thermomangement and cylinder charge composition control. The effect of flow area control at different locations in the engine intake-, exhaust- and EGR path on the engine performance were analyzed with a validated detailed engine model. As a result of this study, the optimal engine air-path system for the target functions was defined with throttle valves realized in the form of a HP-EGR valve and exhaust flap downstream the turbine. It was concluded that all of the targeted functions are achievable by backpressure and intake manifold oxygen concentration control with the above defined optimal air-path actuator setup.

THESES 2 Simplified, control-oriented, nonlinear, dynamic models of the engine air-path system for exhaust backpressure and cylinder charge composition control (Part iv and Chapter 8), [P5, P6, P7, P8, P9].

The simplified, control-oriented, nonlinear, dynamic models of the engine air-path system with a throttle valve installed downstream of the turbine, and high-pressure exhaust recirculation path considered as a mixed thermodynamical, mechanical system were built and verified. It was shown that the model exhibits the following unique properties:

R1. The dynamic models of the engine air-path system with a throttle valve installed downstream of the turbine, and an HP-EGR valve is given by a set of nonlinear differential-algebraic equations. The differential equations are balance equations for the mass and internal energy of the gas in the intake manifold, exhaust manifold and the volume between the turbine and the exhaust throttle as balance volumes.
R2. The model for the cylinder charge composition control uses a performance output of the intake manifold oxygen concentration.

R3. It has been shown that the two state equations of the backpressure model and the five state equations of the cylinder charge composition model can be rewritten into standard input affine form.

\[
\frac{dx}{dt} = f(x) + g(x)u, \quad y = h(x).
\]

R4. The coordinate functions of the nonlinear models have the following properties:

a) The coordinate function \( f(x) \) depends also on the disturbance vector \( d \) and includes hybrid modes: \( f(x) = f(x, d, r) \), where \( r: \mathbb{R}^n \rightarrow \mathbb{N} \) is a piecewise constant switching function mapping from the state space to the finite integer set \( \mathbb{N} = \{1, 2\} \) for the backpressure model and \( \mathbb{N} = \{1, 2, 3, 4, 5, 6\} \) for the cylinder charge composition model. \( x \) is the state vector.

b) The coordinate function \( g(x) \) is affine with respect to the state vector, i.e., \( g(x) = Bx + b \) with \( B \) being a constant matrix and \( b \) a constant vector.

c) The output equation has the following form: \( h(x) = Cx + e(d) \), where \( C \) is a constant matrix and \( e \) is a nonlinear function of the disturbance vector \( d \).

THESIS 3 Specification of the backpressure control problem and design of the pressure controller for the exhaust manifold pressure (Chapter 9), [P5, P6, P7].

R1. The exhaust manifold pressure tracking control problem is given using a position controlled exhaust throttle installed downstream the turbine with respect to the exhaust manifold pressure oscillation caused by the exhaust processes of the cylinders which is a non-modeled dynamics of the system.

R2. Control aims and requirements were defined in five points which need to fulfill by the controller.

R3. With the above assumptions, four different controllers were designed and tuned. The properties of the closed loop-systems were investigated and compared in different engine dynamometer measurement cycles (covering brake blending, thermomanagement, and EGR support operations) which lead to the following observations:

a) The LQ servo controller shows an accurate tracking but cannot fulfill at the same time the closed-loop response time requirement.
b) The H-infinity controller shows quick response times but with high control activity and computational demand.

c) The sliding-mode controller shows a fast dynamics, but its steady state error exceeds the predefined limit in some cases.

d) The LQ servo controller with the inclusion of a model based feedforward can fulfill both the response time and low control accuracy challenge parallel.

\( \mathcal{R}_4 \). Distilled from the above comparison results the LQ servo with the model inversion based feedforward controller structure was chosen as the proposed controller. It fulfills all the predefined requirements, namely the pressure overshoot, tracking accuracy, control activity, computational demand and easy calibration.

\( \text{THESIS 4} \) Specification of the cylinder charge composition control problem and design of the intake manifold oxygen concentration controller with high-pressure exhaust gas recirculation and exhaust throttling (Part iv), [P9, P10, P11].

\( \mathcal{R}_1 \). The cylinder charge gas composition tracking control problem using a position controlled exhaust throttle installed downstream of the turbine, and HP-EGR valve was defined.

\( \mathcal{R}_2 \). A novel performance output was chosen as the intake manifold oxygen concentration after highlighting disadvantages of the commonly used EGR rate based control.

\( \mathcal{R}_3 \). With the above assumptions, an LQ servo controller has been designed and tuned. The properties of the closed-loop system were investigated by engine dynamometer measurements which lead to the following observations:

a) The controller fulfills the requirement on tracking accuracy.

b) The controller fulfills the requirement on minimizing the engine pumping losses.
The results presented above in this thesis can be extended in numerous aspects, of course. In particular, the following research sub-topics can be considered in future work.

\( R_1 \). The intake manifold oxygen concentration control with HP-EGR and exhaust throttling is proved to be a useful tool for the appropriate cylinder charge gas composition adjustment. The novel choice of the performance variable (namely the intake manifold oxygen concentration) eliminates the drawbacks of the control for EGR-rate and reflects for the pollutant formation which is in the real interest. However, the set-point generation for the controller that would result in minimal fuel consumption while agreeing on the legislative emission limits is an other important issue towards the commercialization of the proposed engine air-management system that planned to carry out in the future.

\( R_2 \). The effect of the intake manifold oxygen concentration control on the particulate matter formation was not measured and evaluated in this work, but it has a significant impact on the applicability. Therefore, it is targeted in the future. Moreover, the proper set-point map generation and its optimization couldn’t be probably performed without the development of an emission model that can predict the pollutant formation from the cylinder charge gas composition and injection parameters. As a result with a proper emission model, the direct emission control of the engine could be possible.

\( R_3 \). The implementation of other controller types with even significantly more computational demand (MPC, gain-scheduling LPV, etc.) could result in an interesting comparison how control system performance could increase with the use of a more expensive microcontroller hardware platform.

\( R_4 \). The controller tests were carried out with a rapid prototyping module software implementation. The serial production of the proposed diesel engine air-management system is only possible with an embedded microcontroller software implementation. Therefore, it would be a logical next step to implement and test the system accordingly and evaluate the computational demand for the microcontroller.

\( R_5 \). The intake manifold oxygen concentration control is based on the measurement of all the five state variable. However, it in-
creases the system cost significantly. For this reason with the design of a state estimator (e.g., Kalman filter and Linear Quadratic Gaussian Control (LQG) control) the total number of the needed sensors in the engine air-path system could be reduced that can improve the cost-benefit of the proposed emission control system over other possible solutions (e.g., aftertreatment).

R6. Reduction of engine emission always has a cost which means partially the price of the newly installed equipment (e.g., SCR catalyst, flap valves, etc.) or the increase of the fuel consumption. An economic comparison of state of the art emission handling techniques (e.g., aftertreatment, VGT turbine, improved injection methods, etc.) and the proposed air-management system should be carried out for the evaluation of the customer (commercial vehicle Original Equipment Manufacturer (OEM) or fleet owners) benefit and the emission reduction potential. The synergies of different methods also should be taken into account.

R7. The precise adjustment of the cylinder charge composition can be beneficial in realization and control of different advanced combustion methods (e.g., LTC, Homogeneous Charge Compression Ignition (HCCI), etc.). The evaluation of how effectively can the proposed air-management control method be used in synergy with an advanced combustion system can be considered in the future.

R8. However, the proposed air-management methods were demonstrated with transient dynamometer measurements before a serial production a field test in real life conditions should be considered. In this way, the calibration effort performance and behavior on different engines (robustness) could be precisely analyzed.
SUMMARY AND CONCLUSION

Commercial vehicle diesel engines which are in the focus of this research need to fulfill several requirements traced from different sources, e.g., legislation, economical operation, safety, etc. The legal framework for commercial vehicle diesel engines forces engine developers to constantly utilize new tools to reduce engine emissions and fuel consumption. Engine air-path management with flap valves installed in the intake and exhaust system can be an effective tool for the fulfillment of these requirements. In the first part of this thesis possible new functions feasible by flap valves in the air-path were listed as goals to be achieved. These include brake blending, AMT support, exhaust gas thermomanagement, and cylinder charge composition control. To realize the above goal function the design of a backpressure controller and a cylinder charge composition controller was targeted.

In an air-path system of a turbocharged diesel engine, there are several possible locations for the installation of the flap valves: upstream and downstream the compressor and upstream and downstream the turbine. However, the effect of throttling at different places is not necessarily identical on the engine performance measures (e.g., fuel consumption, emission, etc.). Aiming to identify the optimal flap actuator location layout a comparative analysis was carried out in Part ii. For this purpose, a detailed model of the test engine was designed and validated. The EGR mass flow was supported with throttling at different locations of the intake and exhaust system. It resulted in the most advantageous actuator layout as an HP-EGR valve in combination with throttling at downstream the turbine evaluated from engine performance and actuator physical realization aspects. Hence, this actuator layout was chosen as the proposed optimal setup, and it was used in the further work for the realization of the target control functions.

In Part iii the backpressure control function was realized. This work was started with the description, identification, and validation of a first engineering principle-based, mean-value, nonlinear engine model. After it, four controllers were designed based on the from different fields of control theory for the sake of optimal solution. It included and LQ servo, and LQ servo with model-inversion based feedforward, an H-infinity and a sliding-mode. The control performance was compared with engine dyno measurements in different test cycles covering the differently targeted usecases. As a result, the LQ servo controller augmented with the mode-inversion feedforward
was chosen as the proposed control structure that can fulfill the predefined requirements.

In Part iv a similar workflow was applied for the cylinder charge composition control development. The EGR-rate gives inappropriate characterization about the cylinder charge gas composition. Therefore, the novel performance variable was chosen as the intake manifold oxygen concentration. After it, a physics-based, nonlinear, control-oriented model of the whole engine air-path was worked out, identified and validated as a basis for the controller design. As a consequence of the comparative study of different controllers for the back-pressure control problem in the previous part, the LQ servo design method resulted as the most advantageous controller. Hence, it was chosen for the solution of the intake manifold oxygen control problem. The control structure needs to minimize the pumping losses to ensure maximum engine efficiency. It can be guaranteed only if the exhaust flap starts to close after the EGR valve saturation. For this purpose, the control model was converted into two separate SISO form: one in which only the EGR valve opening is the control input and an other where just the exhaust flap. The models where linearized around a typical equilibrium point and two LQ servo controller was synthesized, and a switching logic was implemented between them. Finally, the controller performance was tested with engine dyno measurements in at three different engine speeds and engine loads. It revealed that the proposed control function could effectively reduce NOx emission at low loads while doesn’t affect the fuel consumption. At higher loads, the rise of the fuel consumption was observable, so the use of the intake manifold oxygen concentration control is recommended here only with limitations. It shows good cooperation with aftertreatment systems: at low engine loads while the exhaust gas temperature is low and the efficiency of the SCR catalyst is low the proposed controller can be used for the emission limitation. At high loads and high exhaust gas temperatures, the aftertreatment system can work with good efficiency.

In Part v the contribution of this work was summarized in four thesis points. The first covers the definition of possible new functions achievable by flap valves in the engine air-path and control strategies and optimal actuator setup for the realization. The second thesis point covers the developed models that can be used for the controller design to realize the targeted new functions. Thesis 3 includes the backpressure controller design, and Thesis 4 summarizes the achievements in connection with the intake manifold oxygen control problem. Finally, directions for future research were outlined. As a contribution of this Ph.D. research work the practical realization of a brake blending, AMT, exhaust gas thermomanagement and intake manifold oxygen concentration control has become possible.
The results of this thesis have been presented at conferences and published or accepted in journals as follows (in parenthesis the relevant Thesis is indicated):


Part VI

APPENDIX
TEST BENCH SETUP

A.1 ENGINE DYNOMETER INSTRUMENTATION

The overall layout of the engine dyno test cell used in this work for the model identification, validation measurements, and controller tests is depicted in the figure below.

Figure A.1: Overall engine test cell layout
The list of measurement devices, sensors, actuators and control devices are listed in the following table with the numbering shown in Fig. A.1.

<table>
<thead>
<tr>
<th>Nr.</th>
<th>Description</th>
<th>Type</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Engine speed sensor</td>
<td>IFM RO6345</td>
</tr>
<tr>
<td>2</td>
<td>Erőmérő cella</td>
<td>Acell TC 30kg 462892</td>
</tr>
<tr>
<td>3</td>
<td>Ambient temperature sensor</td>
<td>Delta OHM HD 2008 TC1-1</td>
</tr>
<tr>
<td>4</td>
<td>Gravimetric fuel consumption meter</td>
<td>Energotest BDF700</td>
</tr>
<tr>
<td>5</td>
<td>Humidity sensor</td>
<td>Delta OHM HD 2008 TC1-1</td>
</tr>
<tr>
<td>6</td>
<td>Barometric pressure sensor</td>
<td>Delta OHM HD9408T Baro</td>
</tr>
<tr>
<td>7</td>
<td>Exhaust manifold pressure sensor</td>
<td>Gems 2200SG</td>
</tr>
<tr>
<td>8</td>
<td>Exhaust manifold temperature sensor</td>
<td>Ni-Cr-Ni</td>
</tr>
<tr>
<td>9</td>
<td>Cylinder pressure sensor (in each cylinders)</td>
<td>AVL GH-14D</td>
</tr>
<tr>
<td>10</td>
<td>Engine speed sensor for combustion analysis (optical, high accuracy)</td>
<td>AVL 365C</td>
</tr>
<tr>
<td>11</td>
<td>Exhaust manifold low-pressure indicating sensor</td>
<td>AVL LP11DA</td>
</tr>
<tr>
<td>12</td>
<td>Lambda sensor</td>
<td>Bosch LSU 4.9D</td>
</tr>
<tr>
<td>13</td>
<td>Intake manifold temperature sensor</td>
<td>PT-100</td>
</tr>
<tr>
<td>14</td>
<td>Exhaust manifold temperature sensor</td>
<td>Gems 2200SG</td>
</tr>
<tr>
<td>15</td>
<td>Intake manifold low pressure indicating sensor</td>
<td>AVL LP11DA</td>
</tr>
<tr>
<td>16</td>
<td>Electromechanical position controlled hot side EGR valve</td>
<td>Knorr-Bremse A sample</td>
</tr>
<tr>
<td>17</td>
<td>EGR temperature sensor</td>
<td>PT-100</td>
</tr>
<tr>
<td>18</td>
<td>Compressor outlet temperature sensor</td>
<td>PT-100</td>
</tr>
<tr>
<td>19</td>
<td>Compressor outlet pressure sensor</td>
<td>Gems 2200SG</td>
</tr>
</tbody>
</table>
In this thesis work, the engine-out nitrogen oxide raw concentration was measured since it can be effectively adjusted by the exhaust gas recirculation. For this purpose, a Continental UniNOx sensor was installed in the exhaust system directly downstream the turbine which gives fast feedback even in transient operation. The sensor was depicted with Nr. 29 in the above section about the sensor layout. The smart NO\textsubscript{x}-Sensor consists of the sensing element (material: Zirconia multilayer ceramics in metal housing) and the electronic control unit, combined by an approx. 600mm (24 inches) cable. Similar to a wide-range linear lambda sensor, electro-chemical pumps adjust
the oxygen concentration in the cavities of the sensing element. The $NO_x$ concentration in the exhaust gas is proportional to the electrical current controlling the pumps. Based upon the physical measurement, the electronic control unit generates three output signals ($NO_x$, binary, linear). The signals are transmitted via Controller Area Network (CAN) bus.

![Continental UniNOx sensor](image)

**Figure A.2: Continental UniNOx sensor used for raw nitric oxide concentration measurement**

<table>
<thead>
<tr>
<th>Description</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>ZrO2-based multilayer sensor with 3 oxygen pumps</td>
<td></td>
</tr>
<tr>
<td>Triple output signal ($NO_x$, linear $\lambda$, binary $\lambda$)</td>
<td></td>
</tr>
<tr>
<td>Supply voltage:</td>
<td>24V</td>
</tr>
<tr>
<td>Measuring range:</td>
<td></td>
</tr>
<tr>
<td>$NO_x$: 0 - 500 ppm or 0-1500 ppm</td>
<td></td>
</tr>
<tr>
<td>lin. $\lambda$: 0,75 to air</td>
<td></td>
</tr>
<tr>
<td>bin. $\lambda$: $&gt;0,75$V at $\lambda=0,9$; $&lt;0,2$V at $\lambda=1,1$</td>
<td></td>
</tr>
<tr>
<td>Accuracy:</td>
<td></td>
</tr>
<tr>
<td>$NO_x$: at 100ppm and 500ppm: ± 10%; at 0ppm: ± 10ppm</td>
<td></td>
</tr>
<tr>
<td>lin. $\lambda$: at $\lambda=0$ ±6 (1000/$\lambda$) fresh</td>
<td></td>
</tr>
<tr>
<td>bin. $\lambda$: 1,002 ± 0,008</td>
<td></td>
</tr>
</tbody>
</table>

**Table A.2: Technical specification of the used Continental UniNOx sensor**

Source: Continental UniNOx datasheet
The measurement of the oxygen concentration in the intake manifold is a challenging task. However, it cannot be avoided for the model validation and of course can make the control design easier and more robust if the measurement for the state variables used as feedback guaranteed. The use of a wide band UEGO seems to be an obvious choice although the temperature in the intake manifold is considerably lower than it is in the exhaust manifold (in its originally intended location). Clearly, a certain temperature range of the sensor material needs to be ensured to ensure the Nernst reaction. Modern lambda sensors are equipped with electrical heater elements. Therefore, it can be ensured by the appropriate control of the heater despite the extensive cooling effect of the cylinder charge gases. From the other side, the intake manifold pressure changes in a wide range due to the turbocharging and the reaction mechanism of the UEGO sensor is highly sensitive on the pressure. For this reason, a correction should be applied to the measured oxygen concentration based on the pumping current. The deviation of the pumping current due to the pressure change can be seen in the figure below.

![Figure A.3: Bosch LSU 4.9D lambda sensor signal pressure dependence](image)

Source: Bosch LSU 4.9D datasheet

Due to the above reasons the whole control and signal processing of the lambda sensor needed to be realised by a custom made system. For the heating a dSpace RapidPro Power Unit was used as a power stage while the control logic was implemented in a dSpace MicroAutoBox. The pump current sensing control and oxygen concentration correction was carried out with a Bosch CJ125 IC installed in a dSpace RapidPro Signal Conditioning Unit in connection with a MicroAutoBox.
The undersigned, Ádám Bárdos declares that this Ph.D. thesis has been prepared by himself as well as that the indicated sources have been used only. All parts that have been taken over literally or by content are cited unambiguously.

Alulírott Bárdos Ádám kijelentem, hogy ezt a doktori értekezést magam készítettem és abban csak a megadott forrásokat használtam fel. Minden olyan részt, amelyet szó szerint, vagy azonos tartalomban, de átfogalmazva más forrásból átvettem, egyértelműen, a forrás megadásával megjelöltem.

Budapest, January 2019

Bárdos Ádám