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Indirect Boosting for Turbocharged SI Engines

Master Thesis

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Supervision

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Preface

The author would like to thank his friends and family for their support during the master thesis. Especially he would like to thank Christoph Voser for the numerous and inspiring discussions throughout the thesis.
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Abstract

The demand for mobility is steadily increasing. The automobile is very popular for transportation purposes. Increasing fuel prices on one hand and also the increasing awareness of environmental problems associated with CO₂ emissions on the other hand lead to the necessity of more efficient drivetrains and engines. One possibility to improve the overall efficiency of engines is the approach called "downsizing". Using this approach, the engine displacement is reduced and a turbocharging device is installed to preserve the maximum power output.

In transient operation, the maximum torque is only available when the turbocharger has accelerated. The time that passes until the turbocharger has accelerated is referred to as the turbo lag. This leads to a decrease in drivability, which means that the potential of downsized engines cannot be exploited. The smaller engines get, the more increases their overall efficiency but the bigger the turbo lag gets.

One system that can be used to overcome these drawbacks are Hybrid Pneumatic Engines (HPE), where the possibility of in-cylinder boosting (direct boosting) exists. In this work, the effectiveness of an alternative system to eliminate the turbo lag is shown and discussed: indirect boosting. In this approach, air is injected into the intake manifold andtreaches the cylinder on an indirect path. Using a model based approach, control strategies are developed. Their feasibility is shown and the results of the different strategies are compared. Indirect boosting is realized on the test bench to demonstrate the feasibility and the transient performance. Lastly, the characteristics of direct boosting and indirect boosting are compared.
## Nomenclature

### Symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
<tr>
<td>(e)</td>
<td>Willans Efficiency Coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>(P)</td>
<td>Power</td>
<td>[kW]</td>
</tr>
<tr>
<td>(T)</td>
<td>Torque</td>
<td>[Nm]</td>
</tr>
<tr>
<td>(V)</td>
<td>Volume</td>
<td>[m³]</td>
</tr>
<tr>
<td>(B)</td>
<td>Bore</td>
<td>[m]</td>
</tr>
<tr>
<td>(S)</td>
<td>Stroke</td>
<td>[m]</td>
</tr>
<tr>
<td>(l_p)</td>
<td>Connecting Rod Length</td>
<td>[-]</td>
</tr>
<tr>
<td>(\epsilon)</td>
<td>Compression Ratio</td>
<td>[-]</td>
</tr>
<tr>
<td>(N)</td>
<td>Number of Cylinders</td>
<td>[-]</td>
</tr>
<tr>
<td>(\omega)</td>
<td>Rotational Speed</td>
<td>[rpm, rad/s]</td>
</tr>
<tr>
<td>(H_l)</td>
<td>Lower Heating Value</td>
<td>[J/kg]</td>
</tr>
<tr>
<td>(\sigma_0)</td>
<td>Stoichiometric Coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>(\lambda)</td>
<td>Air-to-Fuel Ratio</td>
<td>[-]</td>
</tr>
<tr>
<td>(\lambda_l)</td>
<td>Volumetric Efficiency</td>
<td>[-]</td>
</tr>
<tr>
<td>(\kappa)</td>
<td>Ratio of Specific Heats</td>
<td>[-]</td>
</tr>
<tr>
<td>(A)</td>
<td>Aarea</td>
<td>[m²]</td>
</tr>
<tr>
<td>(R)</td>
<td>Specific Gas Constant</td>
<td>[J/kg·K]</td>
</tr>
<tr>
<td>(\theta)</td>
<td>Temperature</td>
<td>[K]</td>
</tr>
<tr>
<td>(m)</td>
<td>Mass</td>
<td>[kg]</td>
</tr>
<tr>
<td>(\dot{m})</td>
<td>Mass Flow</td>
<td>[kg/s]</td>
</tr>
<tr>
<td>(t_{90})</td>
<td>Rise Time</td>
<td>[s]</td>
</tr>
<tr>
<td>(\nu_{ob})</td>
<td>Overboosting Factor</td>
<td>[-]</td>
</tr>
<tr>
<td>(c_p)</td>
<td>Isobaric Heat Capacity</td>
<td>[J/kg·K]</td>
</tr>
<tr>
<td>(c_d)</td>
<td>Discharge Coefficient</td>
<td>[-]</td>
</tr>
<tr>
<td>(t)</td>
<td>Time</td>
<td>[s]</td>
</tr>
<tr>
<td>(\tau)</td>
<td>Time Constant</td>
<td>[s]</td>
</tr>
<tr>
<td>(CA)</td>
<td>Crank Angle</td>
<td>[°CA]</td>
</tr>
<tr>
<td>(\Delta\Theta_{90})</td>
<td>Combustion Duration</td>
<td>[°CA]</td>
</tr>
<tr>
<td>(\dot{q}'')</td>
<td>Heat Flux</td>
<td>[W/m²]</td>
</tr>
<tr>
<td>(h)</td>
<td>Heat Transfer Coefficient</td>
<td>[W/m²·K]</td>
</tr>
</tbody>
</table>
Indicies

a  Ambient
air  Air
thr  Throttle
bv  Boost Valve
b  Boost
osv  Overstream Valve
im  Intake Manifold
em  Exhaust Manifold
eg  Exhaust Gas
t  Pressure Tank
crit  Critical
in  In
out  Out
e  Engine
ss  Steady State
max  Maximum
me  Mean Effective
mi  Mean Indicated
m  Mixture
φ  Fuel
d  Displacement
c  Compressor
ω  Rotational Speed
f  Friction
des  Desired
eff  Effective
β  Engine
α  Throttle
ζ  Ignition
TC  Turbo Charger
t  Turbine
off  Off
end  End of a Boosting Process
cyl  Cylinder
g  Gas Exchange
hp  High Pressure
IPS  Induction-to-Power-Stroke
**Acronyms and Abbreviations**

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>ETH</td>
<td>Eidgenössische Technische Hochschule</td>
</tr>
<tr>
<td>IDSC</td>
<td>Institute for Dynamic Systems and Control</td>
</tr>
<tr>
<td>MVM</td>
<td>Mean Value Model</td>
</tr>
<tr>
<td>HPE</td>
<td>Hybrid Pneumatic Engine</td>
</tr>
<tr>
<td>DB</td>
<td>Direct Boosting</td>
</tr>
<tr>
<td>IB</td>
<td>Indirect Boosting</td>
</tr>
<tr>
<td>MBT</td>
<td>Maximum Brake Torque</td>
</tr>
<tr>
<td>HFM</td>
<td>Air Flow Sensor</td>
</tr>
<tr>
<td>PWM</td>
<td>Pulse Width Modulation</td>
</tr>
<tr>
<td>SI</td>
<td>Spark Ignited</td>
</tr>
<tr>
<td>OSV</td>
<td>Overstream Valve</td>
</tr>
<tr>
<td>LHV</td>
<td>Lower Heating Value</td>
</tr>
</tbody>
</table>
Chapter 1

Introduction

1.1 Downsizing and Turbocharging

One major disadvantage of spark ignited (SI) combustion engines is that the load is changed by varying the quantity of the mixture. This means that in part load operation the intake pressure has to be reduced significantly which results in pumping losses and a decrease in overall efficiency. In order to avoid high pumping losses, the average operating point of the engine needs to be shifted towards higher loads. Assuming the use of the same gearbox, this can be achieved by reducing the displacement of the engine. However, when doing so the maximum power output of the engine also decreases. One way to maintain the same power output of the engine at a reduced displacement is the use of supercharging or turbocharging devices.

These devices increase the pressure in the intake manifold of the engine and thereby lead to a higher engine mass flow. By this means higher mean effective pressures ($p_{me}$) than in naturally aspirated engine operation can be achieved.

The engine used in this project is a turbocharged engine. The basic parameters of the engine are listed in table 1.1.

<table>
<thead>
<tr>
<th>Table 1.1: Basic Data of the Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>engine type</td>
</tr>
<tr>
<td>injection</td>
</tr>
<tr>
<td>turbocharger</td>
</tr>
<tr>
<td>rated power</td>
</tr>
<tr>
<td>rated torque</td>
</tr>
<tr>
<td>displacement</td>
</tr>
<tr>
<td>bore, stroke</td>
</tr>
<tr>
<td>connecting rod</td>
</tr>
<tr>
<td>compression ratio</td>
</tr>
<tr>
<td>number of cylinders</td>
</tr>
</tbody>
</table>
1.2 Turbo Lag and Drivability

Adding a turbocharger significantly alters the transient behaviour of an engine. At low loads the turbocharger is only rotating at low speeds. In order to provide high boost pressure to enable high loads however, the rotational speed of the turbocharger has to be significantly higher. Only at high rotational speeds, high mass flows and at the same time high intake pressures can be provided. Due to the inertia of the compressor and the turbine, it takes some time until the turbocharger has accelerated and the desired torque is reached. This time is generally referred to as the "turbo lag".

The presence of this turbo lag changes the drivability of a vehicle. If the driver desires the full load torque at a tip in event, the torque response of the engine is much slower. The more downsized an engine is, i.e. the higher the mean effective pressures get, the bigger the turbo lag gets. There are several approaches on how the turbolag can be avoided:

- adding a second turbocharger
- combining a turbocharger and a supercharger
- adding an electric machine to the drivetrain, e.g. Hybrid Electric Vehicles
- adding an electric machine to the turbocharger itself
- injection of air into the cylinder, e.g. Hybrid Pneumatic Engines (HPE)
- ...

All of these concepts improve the response behaviour of a vehicle.

1.3 Hybrid Pneumatic Engines

On HPEs the cylinder is connected to a pressure tank by a valve (the "charge valve"). Several operating modes that are not possible in a conventional engine can be realized on HPEs. These operating modes will be introduced briefly here.

- Pneumatic Pump Mode: This operating mode can be engaged, when no fuel is injected. The charge valve is opened during the compression stroke and air is transported from the cylinder into the pressure tank. By this means energy in the form of pressurized air can be stored in the tank.

- Pneumatic Motor Mode: In this operating mode the charge valve is opened during the expansion stroke. The air from the tank flows into the cylinder and hence the cylinder pressure is increased. Thereby the piston is pushed downwards and the engine is driven by air alone.

- Boost Mode: This operating mode can be activated during conventional operation with combustion. If the charge valve is opened in the compression stroke shortly after the intake valve closes, the pressure in the cylinder is still lower than the pressure in the tank. Air can flow from the tank into the cylinder. The in-cylinder air mass is thereby increased and more fuel can be burned per cycle. This results in an immediate increase in torque on one hand but also in a higher exhaust gas mass flow on the other hand.
Figure 1.1: System structure of a Hybrid Pneumatic Engine. The pressure tank is directly connected to the cylinder.

that helps accelerate the turbocharger. In transient engine operation both these effects are beneficial. In this text, this operating mode is referred to as “direct boosting”.

- Pneumatic Start: With a HPE it is also possible to start the engine solely with air. This helps prevent the idling losses without an additional start stop system.

As described in [2] the downsizing effect is by far the most important effect when it comes to fuel saving. Using the boost mode of a HPE, the turbo lag can effectively be overcome. The HPE’s major advantage compared to a conventional turbocharged engine is that it enables strong downsizing without compromising drivability. Further details about HPEs can be found in [9] and in [10].

1.4 Indirect Boosting

The existance of the turbo lag is only due to the lack of air while the turbocharger is accelerating. Instead of injecting air directly into the cylinder to overcome the turbo lag, one other possibility is to supply additional air at a location upstream of the intake valves during the phase the turbocharger accelerates. If a higher intake manifold pressure can be reached like this the engine mass flow would increase. One possibility to realize that is the use a valve that
injects air into the path of the fresh air. This valve will be referred to as the "boost valve". That means, that the supplied air will reach the combustion chamber only indirectly and hence such a configuration is referred to as "indirect boosting". Several locations for the injection of air are possible. In air is injected upstream of the compressor. In this work however, the air is injected into the intake manifold as can be seen in figure 1.2.

Figure 1.2: System structure of an engine with indirect boosting. The pressure tank is connected to the intake manifold.

In contrast to a HPE, no modifications to the cylinder head are necessary with such a configuration. The established configuration of todays turbocharged engines can remain almost unchanged. The biggest advantages of HPEs, namely enabling strong downsizing while offering the ability to overcome the turbo lag, are expected to be realizable with this concept.

1.5 Motivation and Goals of the Project

Modifications to the cylinder head can be very costly, which is a disadvantage of HPEs. The biggest advantage of these engines is to enable strong downsizing without compromising drivability by using the boost mode. In this thesis indirect boosting is investigated since it is expected to show similar performance when it comes to eliminating the turbo lag.

Guidelines are given for the dimensioning of an indirect boosting system. Several control strategies are developed to realize indirect boosting. The most important aspects of this concept are the transient performance since the system is intended
to overcome the turbo lag and the air mass consumption required for that. The air mass consumption is especially important since it will give information on how the packaging of a vehicle will be influenced by an indirect boosting system, i.e. how large the pressure tank has to be.

The goals of this thesis are:

- carry out load steps with indirect boosting on the testbench to demonstrate the feasibility of indirect boosting
- compare direct boosting (DB) and indirect boosting (IB) with respect to the transient performance and the air mass consumption

1.6 Structure of the Thesis

The work is divided into the following parts which had to be carried out successively in order to achieve the goals:

- Theoretical Part
  - modifications to the existing mean value model
  - dimensioning of the boost valve
  - problems with compressor surge in the simulations
  - development of control strategies

- Experimental Part
  - modifications to the engine
  - identification of the boost valve
  - modifications to the engine control
  - attain experimental results
  - comparison of measurements and simulations

- Comparison of direct boosting and indirect boosting
1.6. Structure of the Thesis
Chapter 2

Simulation Model for Indirect Boosting

No information on whether indirect boosting has been carried out before in the way presented here on a SI engine can be found. In addition, only few information is found on indirect boosting of Diesel engines (see [5] and [6]). Hence, it cannot be known how fast the dynamics of the single components such as the throttle or the boost valve need to be in order to fulfill the requirements that may arise when applying indirect boosting. Due to this fact and also because the first part of the theoretical study is intended to be a potential study, ideal actuators are considered here. That means that they are infinitely fast and have no leakage.

2.1 Modifications to the Existing Mean Value Model

Due to the numerous works on HPEs that have already been carried out at the Institute for Dynamic Systems and Control (IDSC) a Mean Value Model (MVM) of the engine to be used already exists. Only few modification had to be made for the implementation of indirect boosting.

The intake manifold receiver gets an additional inflow which is the mass flow of the boost valve. The boost valve itself is modeled as a compressible flow restriction. In this part of the thesis the boost valve is connected to an infinitely large pressure tank, i.e. the pressure of the tank is constant.

In order to design a controller for the mass flow of the boost valve a torque inversion model has to be developed first. The purpose of the torque inversion model is to calculate the desired engine air mass flow $\dot{m}_\beta$ that is necessary for a desired engine torque depending on the current values of the intake pressure, exhaust pressure and engine speed. In the MVM present, a Willans approximation is used to calculate the engine torque:

$$p_{me} = e(m_\omega, \omega_e, \lambda, \zeta, \ldots) \cdot p_{me0}(\omega_e, \vartheta_e, \ldots)$$

The thermodynamic efficiency of the combustion in the Willans approximation is represented by $e$. The effective mean pressure $p_{me}$ is a normalized value for
the engine torque $T_e$. The fuel mass flow can also be normalized and then is represented by $p_{m\varphi}$. They are defined as written in equation 2.2.

$$p_{me} = \frac{T_e \cdot 4\pi}{V_d}, \quad p_{m\varphi} = \frac{H_l \cdot m_\varphi}{V_d}$$

(2.2)

The term $p_{me0}$ includes the mechanical friction losses as well as the pumping losses of the engine.

$$p_{me0}(\omega_e, \dot{m}_\beta, \dot{\vartheta}_e, \ldots) = p_{me0,f}(\omega_e, \dot{\vartheta}_e, \ldots) + (p_{em} - p_{im})$$

(2.3)

Details of the models used to represent the thermodynamic efficiency $e$ and the mechanical engine friction $p_{me0,f}$ can be found in [3] as well as a more detailed derivation of the Willans approximation. The parameters to adapt these models have been identified in previous work (see [11] and [7]). Some simplifications can be made to these general formulations in case of this work:

- the engine is always operated at $\lambda = 1$
- no external recirculation of exhaust gas is present
- all measurements are carried out using the warmed up engine
- in the inversion of the torque model, an ignition efficiency of $e_\zeta(\zeta) = 1$ is assumed

With these simplifications, the effective mean pressure of the engine is represented by:

$$p_{me} = e_\omega(\omega_e) \cdot \frac{LHV \cdot m_\beta}{V_d \cdot \sigma_0} - p_{me0,f}(\omega_e) - (p_{em} - p_{im})$$

(2.4)

The inversion of equation 2.4 finally leads to the equation for the desired engine air mass $m_{\beta,des}$:

$$m_{\beta,des} = \frac{V_d \cdot \sigma_0}{LHV \cdot e_\omega(\omega_e)} \cdot [p_{me,des} + (p_{em} - p_{im}) + p_{me0,f}(\omega_e)]$$

(2.5)

where $p_{me,des}$ represents the desired effective mean pressure.

### 2.2 Model based analysis

The pressure dynamics of the intake manifold can be calculated analytically. The equation for the intake manifold pressure dynamics is:

$$\frac{d}{dt} \dot{p}_{im}(t) = \frac{\kappa \cdot R}{V_{im}} \cdot [\dot{m}_{im} \cdot \dot{\vartheta}_i - \dot{m}_\beta \cdot \dot{\vartheta}_{im}(t)]$$

(2.6)

The outflow of the intake manifold is the engine mass flow. In an MVM the engine is modeled as a volumetric pump. So in each engine cycle a flow is forced that is approximately equal to the displacement of the engine:

$$\dot{m}_\beta(t) = \dot{\rho}_{im}(t) \cdot \lambda_1(p_{im}, \omega_e) \cdot \frac{V_d \cdot \omega_e(t)}{4\pi} = \frac{p_{im}}{R \cdot \dot{\vartheta}_{im}} \cdot \lambda_1(p_{im}, \omega_e) \cdot \frac{V_d \cdot \omega_e(t)}{4\pi}$$

(2.7)
The volumetric efficiency $\lambda_l$ describes what fraction of the engine’s displacement is filled with fresh air in each cycle. It is modeled according to \[3\]. However, it does not vary significantly over the engine operating range and is treated as a constant here. A value of $\lambda_l = 0.84$ is chosen, which is a good approximation for this engine at high load for all engine speeds.

The inflow into the intake manifold in the case of indirect boosting can come from the throttle and from the boost valve. To clarify the results that will be obtained with this analysis, the throttle mass flow is assumed to be zero and there is only a mass flow from the boost valve.

Inserting equation 2.7 into equation 2.6 and assuming a constant engine speed $\omega_e$ results in a linear first order differential equation with constant coefficients.

\[
\frac{d}{dt} p_{im}(t) = -k_1 \cdot \omega_e \cdot p + k_2 \cdot \dot{m}_{bv}, \quad p_{ss} = \frac{k_2 \cdot \dot{m}_{bv}}{k_1 \cdot \omega_e} \quad (2.8)
\]

$p_{ss}$ is the highest intake pressure that can be reached at a certain engine speed and boost valve mass flow. The coefficients are:

\[
k_1 = \kappa \cdot \lambda_l \cdot \frac{V_d}{V_{im} \cdot 4\pi}, \quad k_2 = \frac{\kappa \cdot R \cdot \vartheta_t}{V_{im}}
\]

Note that the time constant of this system is $\tau_{im} = \frac{1}{k_1 \cdot \omega_e}$. The solution to equation 2.8 is

\[
p_{im}(t) = p_{ss} - (p_{ss} - p_0) \cdot e^{-\frac{t}{\tau_{im}}} \quad (2.9)
\]

with $p_0$ being the intake manifold pressure when the boosting event starts.

From equation 2.8 two very important insights become apparent. Firstly of all the steady state pressure $p_{ss}$ that can be reached is inversly proportional to the engine speed: $p_{ss} \sim \frac{1}{\omega_e}$. This means that at lower engine speeds, where the turbo lag is more pronounced, a higher intake manifold pressure can be reached. Secondly, the intake manifold time constant $\tau_{im}$ gets shorter with increasing engine speed. Both these effects must be taken into account when comparing the pressure increase at different engine speeds and equal boost valve mass flows. Assuming a step in the boost valve mass flow at $t = 0$, the pressure gradient at that time calculates to:

\[
\frac{d}{dt} p_{im}(t = 0) = k_2 \cdot \dot{m}_{bv} - k_1 \cdot \omega_e \cdot p_0 \quad (2.10)
\]

In 2.10 it becomes apparent that the pressure increase is faster at lower engine speeds. Since the torque depends mainly on the intake pressure, the torque build-up will be faster at lower engine speeds where the turbo lag is most pronounced, which is advantageous.

### 2.3 Compressor properties

Centrifugal compressors, as the one used the present engine, have several operating limits. The most important ones can be seen in figure 2.1, namely they are:
- the blocking limit: at zero speed the compressor represents a blocking orifice
- the mechanical limit: there is a maximum rotational speed above which the compressor can get damaged due to the large centrifugal forces
- the choke limit: the maximum possible mass flow is reached, when there are sonic conditions in the narrowest section between two compressor blades
- the surge limit: at every compressor mass flow there is a maximum pressure ratio that should not be exceeded. If it is exceeded however, fluid-dynamic instabilities destroy the regular flow pattern inside the compressor (3). This leads to backflow through the compressor and can lead to oscillations of the air column at the compressor inlet which can damage the compressor.

![Compressor Operating Map](image)

Figure 2.1: Compressor Operating Map

The variable $\mu_c$ denotes a normalized value for the compressor mass flow and $\Pi_c$ is the ratio of the pressure upstream and downstream of the compressor. These operating limits, especially the surge limit, must be respected during indirect boosting.

### 2.4 Dimensioning of the Boost Valve

When it comes to the dimensioning of indirect boosting, some pondering of the implications of opening the boost valve must be done first. At the beginning of a boosting process the engine is operated in low part load and the intake manifold pressure is very low. If the tank pressure is sufficiently high, there will be sonic conditions during the whole boosting process, even when high intake manifold pressures are reached. Therefore a certain mass flow will definitely enter the intake manifold. If the throttle was kept open during a boosting process and a high mass flow was injected by the boost valve the pressure would increase in both the intake manifold and also in the receiver
between the compressor and the throttle. Since at very low compressor speeds, as they are present in low partload operating points of an engine, the pressure ratio over the compressor can only be very small, a too high boost valve mass flow would drive the compressor into surge.

From this consideration two things can be concluded. Firstly, the throttle should be closed during a boosting process as a first premise to avoid surge. Secondly, under this assumption the boost valve must supply the complete engine mass flow during the boosting process.

Due to the fact that with indirect boosting the turbo lag shall be eliminated, only low and medium engine speeds must be considered, since at high engine speeds the turbo lag gets smaller. In particular load steps to full load are of interest, since there the turbo lag is most pronounced.

In this thesis, the maximum boost valve mass flow is chosen to be equal the engine mass flow at the operating point 4000 rpm and 17 bar effective mean pressure which is 0.0441 kg/s.

For this work a tank pressure of 10 bar is assumed. A result of the chosen tank pressure is that the pressure ratio over the boost valve is always supercritical. For supercritical pressure ratios the mass flow equation for isentropic flow is:

\[ \dot{m}_{bv} = \frac{A \cdot c_d \cdot p_t \Psi_{crit}}{\sqrt{R \cdot T_t}} \]  

The outcome of this is a required effective area \( A_{eff} = A \cdot c_d \) of the boost valve of approximately \( 1.9 \cdot 10^{-5} \) m\(^2\).

### 2.5 Compressor Surge Problem

During a boosting process, the throttle has to be closed as described in 2.4. This means that the compressor releases its complete massflow into a receiver that has just one inflow and no outflow. The pressure in that receiver will therefore increase and at some point the surgeline may be reached.

In order to avoid surge, an overstream valve (OSV) is included in the model. The overstream valve connects the receivers downstream and upstream of the compressor. To avoid surge, the valve can be opened and thereby air is recirculated to the compressor entry. This measure reduces the boost pressure and hence the mass flow of the compressor increases and the operating point of the compressor moves away from the surge line.

It is one major challenge of indirect boosting to avoid surge. In all control strategies this must be respected.
Chapter 3

Controller Structure

The desired torque is the reference value for the boost valve controller and the ignition controller. With the torque inversion model the torque is converted to a desired air mass. Feedback is supplied by the air mass entering the engine which is calculated in a MVM as described by formula 2.7. In the engine control on the testbench the engine mass flow is determined by an observer. An overview of the controller structure used in the model can be seen in 3.1.

![Controller Structure](image)

The OSV controller and the ignition controller are always active. The boost valve controller only is used during fast engine transients. The throttle controller is used in slow transients and during steady state engine operation.

3.1 Throttle Controller

The throttle controller is implemented as a PI-controller with a feedforward path. During conventional operation it is used to control the air mass necessary for the demanded torque.


### 3.2 Boost Valve Controller

The systems that need to be controlled here are the boost valve and the intake manifold. A PI-controller with a feedforward path is developed for this purpose. The boost valve controller determines the boost valve mass flow for the desired torque during transients. As previously stated the boost valve is modeled without dynamics for now. This means that the boost valve mass flow can change arbitrarily. There are also no dynamics present in the engine mass flow so the only dynamics that have to be considered are the dynamics of the intake manifold. Equation 2.8 already takes into account all those effects and can have the engine mass flow as an output. It can be rewritten considering that $A = -k_1 \cdot \omega_e + \frac{1}{\tau_{im}}$, $b = k_2$ and $c = k_3 \cdot \omega_e$ with $k_3 = \frac{\lambda V_e}{\pi^3 \cdot \rho_d \cdot \theta_{im} \cdot 4}$. The intake manifold temperature is assumed to be constant and equal to the tank temperature. The input $u(t)$ is the boost valve mass flow $\dot{m}_{bv}$. This leads to the state space system description in the time domain:

$$\frac{dx(t)}{dt} = -k_1 \cdot \omega_e \cdot x(t) + k_2 \cdot u(t) = A \cdot x(t) + b \cdot u(t) \quad (3.1)$$

$$y(t) = k_3 \cdot \omega_e \cdot x(t) = c \cdot x(t) \quad (3.2)$$

Transforming this into the input-output description in the frequency domain results in the following system description:

$$\Sigma(s) = \frac{1}{\tau_{im} \cdot s + 1} \quad (3.3)$$

When a value of $T_i = \tau_{im}$ is chosen for the PI-controller the phase margin is $90^\circ$ for all frequencies. Since $\tau_{im}$ is a function of the engine speed only, since all other variables $\tau_{im}$ consists of are constant, the controller has to be gain scheduled with the engine speed.

Due to the limitation of the maximum boost valve mass flow, a saturation and an anti-reset-windup are added to the controller. Also, since the controller needs to be very fast, a feedforward path is added. In the feedforward path the desired mass flow $\dot{m}_{\beta,des}$ calculated by the torque inversion model is added.

### 3.3 Overstream Valve Controller

The function of the overstream valve controller is to keep the operating point of the compressor as close as possible to the surgeline, but to avoid surge. From measurement data of the compressor, that was also used to identify the compressor model, the surgeline was extracted. The surgeline is approximated with a third order polynomial in the region of low mass flows. For higher mass flows the surgeline is approximated with a linear function. The input to the overstream valve controller is the difference between the current and the surge compressor pressure ratio $\varepsilon = \Pi_c - 0.97 \cdot \Pi_{c,surge}$ and a PI-controller is used. The factor of 0.97 is necessary to allow small overshoots while still avoiding surge. It is arbitrarily chosen and was found to show good performance.
3.4 Ignition Controller in the Model

In an MVM, there is no ignition angle present. However, the effect of changing torques due to alterations in the ignition angle on a real engine is simulated by the use of an ignition efficiency $e_\zeta$. Since the efficiency of the combustion is constant at every engine speed in a Willans approximation, the ignition efficiency calculates as the ratio of desired air mass and the actual air mass:

$$e_{\zeta, \text{des}} = \min \left( \frac{m_{\beta, \text{des}}}{m_\beta}, 1 \right)$$  \hspace{1cm} (3.4)

When too much air enters the engine but the produced torque shall not overcome a certain value this can be adjusted by reducing the ignition efficiency. When the ignition efficiency is reduced, the exhaust gas temperature increases. According to [3] the increase in exhaust gas temperature this causes calculates to

$$\Delta \vartheta_{\text{eg}}(e_\zeta) \approx k_e \cdot \frac{LHV_m}{c_{p, \text{eg}}} \cdot c_{\omega, e} (\omega_{e}) \cdot (1 - e_\zeta)$$  \hspace{1cm} (3.5)

whereas $k_e \approx 0.5$ is a factor that quantifies how much of the available heat flux is diverted to the exhaust gas. $LHV_m$ and $c_{p, \text{eg}}$ are the lower heating value of the mixture and the specific heat capacity of the exhaust gas respectively.

In the present MVM, there is an ignition controller implemented to do that. In this work the ignition controller is used for torque shaping on one hand and for limiting the maximum output torque on the other hand.

The relation between a desired efficiency reduction and the required delay of the ignition angle for the real engine are explained in section 5.3.1.
Chapter 4

Different Strategies for Indirect Boosting

Four different strategies of how indirect boosting can be realised were developed and investigated. In all these simulations, the wastegate was closed at all times. At first each strategy is presented and discussed at one load step and in the complete operating area. After that the results of the various strategies are compared.

4.1 Strategy I

In the first strategy that was developed, the throttle is closed during the load step and the boost valve supplies no more than the air mass flow calculated by the torque inversion model. The throttle is opened again once the boost pressure (i.e. the pressure between compressor and throttle) has reached the value of the intake manifold pressure. A load step at 2000 rpm from 1.7 bar to 17 bar $p_{me}$ is simulated. This load step is the reference for comparing the different strategies in this thesis and hence it is sometimes referred to as "reference load step".

4.1.1 Result for the Reference Load Step

As can be seen in the upper plot of figure 4.1, the desired $p_{me}$ increases very quickly, after 137 ms 90% of the desired $p_{me}$ is reached. However, the whole boosting process takes much longer than that, approximately 750 ms. The air mass consumed during this boosting process is 18.35 g. The boost valve opens completely at the beginning of the load step until the intake pressure is sufficiently high. Then the boost valve closes so that no more than the engine mass flow necessary for the desired $p_{me}$ is delivered which can be seen in the third plot. The fourth plot shows that the overstream valve opens during the boosting process.
Figure 4.1: Turbo lag compensation with indirect boosting Strategy I for a load step from 1.7 bar to 17 bar \( p_{me} \) at \( \omega_e = 2000 \text{ rpm} \).
4.1.2 Discussion

The air mass consumed from the pressure tank by a boosting process will be referred to as $m_b$. The rise time of the $p_{me}$, until 90% of the $p_{me}$ difference is overcome, will be referred to as the $t_{90}$-time. For the following discussions the boosting process will be divided into three phases:

Phase I: From the step time (which is always at $t = 0$ s) until 90% of the desired $p_{me}$ is reached

Phase II: From the end of Phase I until the end of the boosting process when the throttle opens again

Phase III: After the boosting process is over, i.e. when the throttle is open again and the throttle controller is used again to control the engine air mass flow

The reason that the boosting process takes much longer than just the $t_{90}$-time can be seen in the second plot. The turbocharger has not fully accelerated when $p_{me,des}$ is reached. Since the turbocharger only accelerates slowly, the boost pressure $p_b$ cannot build up quickly. The throttle however can only be opened when the boost pressure has reached the level of the intake manifold pressure. Otherwise, air would flow out of the intake manifold and the intake manifold pressure and hence $p_{me}$ would collapse. The fact that the overstream valve is opened means that this is necessary for this strategy to avoid surge. The operating point of the compressor follows the surgeline due to the OSV controller (see figure 4.8). Without the OSV controller surge would occur.

On one hand, the value of $m_b$ results from the desired $p_{me}$ since the complete engine mass flow has to be supplied by the boost valve and on the other hand from the time until the turbocharger has accelerated. Equation 4.1 shows this correlation.

$$m_b = \int_0^{t_{end}} \dot{m}_{bv} \, dt$$

The rise time is very short which may be disadvantageous for the drivetrain since oscillations of elastic components in the drivetrain might occur. Mechanical stress caused by these oscillations could cause damage to the engine.

4.1.3 Decreasing the Rise Time

Opening the boost valve slowly at the beginning of the load step will lead to a less rapid increase of $p_{me}$ but will also extend the duration of Phase I and hence increase the air mass consumption (see figure 4.3). Hence, an other possibility was investigated to allow slower $t_{90}$-times: ignition efficiency reduction. By reducing the ignition efficiency, the produced $p_{me}$ decreases. So by providing a ramp for the desired air mass at the beginning of the boosting process, the engine torque development can be adjusted in a wide range. Figure 4.2 shows the result for a desired $t_{90}$-time of 350 ms. Since a very late ignition may lead to ignition failures, the minimum possible ignition efficiency is limited to a value of $\epsilon_{\zeta, min} = 50\%$ for this engine. An implication of delaying the ignition angle is a lower required air mass $m_b$. This is due to an increased speed-up of the turbocharger. A faster turbocharger
strategy I

\[ t_{90} = 350 \text{ ms} \]
\[ t_{90} = 137 \text{ ms} \]

\[ p_{me} \]
\[ \omega_{TC} \]
\[ p_{me,max} \]
\[ \omega_{TC,max} \]
\[ 1 - e^{-\zeta} \]

\[ t [s] \]

Figure 4.2: Load step with retardation of the ignition angle to increase the rise time for a load step from 1.7 bar to 17 bar \( p_{me} \) at \( \omega_e = 2000 \text{ rpm} \).

Speed-up leads to a faster increase of the boost pressure and hence the boost valve can be closed earlier. For the reference load step the rise time was decreased once by reducing the boost valve opening at the beginning of the load step and once by reducing the ignition efficiency. The resulting air mass consumption \( m_b \) for several slower \( t_{90} \)-times can be seen in figure 4.3.

Figure 4.3: Air mass consumption \( m_b \) for increased rise times \( t_{90} \) for a load step from 1.7 bar to 17 bar \( p_{me} \) at \( \omega_e = 2000 \text{ rpm} \).

4.1.4 Varying the Volume of Receiver Two

Receiver two is the receiver between the compressor and the throttle. The pressure in this receiver \( (p_b) \) determines the operating point of the compressor. The compressor speed together with \( p_b \) determines the compressor mass flow. Since the volume if this receiver determines the operating point of the compressor it also must have an influence on the surge tendency. The surge tendency is high, when the operating point of the compressor gets close to the surgeline during a boosting process.

In figure 4.4 it can be seen that the volume of that receiver has a significant
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Figure 4.4: Operating point of the compressor during a load step from 1.7 bar to 17 bar $p_{me}$ at $\omega_c = 2000$ rpm for various volumes of the receiver downstream of the compressor.

Influence on the trajectory of the compressor during the load step. The basic value of the volume of this receiver is 5 l. Directly after the load step the mass flow decreases for all receiver volumes. When the turbocharger accelerates, for an increasing volume the operating point moves more towards higher mass flows directly after it had reduced. When the volume is reduced, the trajectory reaches the surgeline very quickly. All trajectories eventually follow the surge line. This means that the OSV controller is necessary in all cases.

In all cases the trajectory of the turbocharger speed $\omega_{TC}(t)$ in Phase I is almost the same. The reason that, after shortly decreasing to zero, the operating point moves towards higher mass flows with an increasing receiver volume is that due to the low compressor mass flow at the beginning of a load step the pressure increases slower for bigger receiver volumes. After the same time the operating points for various receiver volumes are approximately on the same compressor iso speedline. At equal turbocharger speeds, the mass flow is lower for lower pressure ratios due to the negative slope of the compressor iso speedlines. Since the pressure is lower in case of the bigger receiver volume, a higher mass flow adjusts and the operating point moves away from the surgeine.

The result of this is that with a bigger receiver volume the osv is only opened during a shorter time since the operating point only reaches the surgeine later. In case of the smallest receiver volume an air mass of 4.9 g is recirculated over the OSV while for the biggest volume only 2.5 g of air are recirculated.

During Phase II of the boosting process the turbocharger speed and also $p_b$ increase slower for larger receiver volumes. This has an implication on the air mass consumption from the pressure tank. Since a slower increase of the boost pressure leads to a longer time until the boost valve closes, the air mass
consumption $m_b$ increases for bigger receiver volumes. In case of the smallest receiver volume an air mass of $m_b = 17.7 \text{ g}$ is needed while for the biggest volume $19.1 \text{ g}$ are necessary to carry out the load step.

These results show that the surge tendency is increased when the volume between the compressor and the receiver is small since the surgleine is reached quicker in that case. A larger receiver volume leads to an increase of the air mass consumption but helps to move the operating point of the compressor away from the surgleine.

### 4.1.5 Results for the Entire Operating Range

With every strategy presented, load steps were simulated over the whole operating range of indirect boosting. Of special interest are the air mass consumption $m_b$, the rise time $t_{90}$ and whether the OSV needs to be controlled or can remain closed.

![Graphs showing consumption and rise time](image)

In figure [image] the results for Strategy I are shown. The consumed air mass $m_b$ is highest at low engine speed and for steps to high engine loads. For every engine speed, $m_b$ increases with the engine load. This increase with engine load can be explained by the fact, that the turbocharger speed increases with the engine load. Since during this time the complete engine mass flow has to be supplied and the engine mass flow increases with the load, more air mass $m_b$ is needed. For every engine load, $m_b$ decreases with increasing engine speed. This decrease can be explained by the turbocharger characteristics. At low engine loads the turbocharger speed is much higher for high engine speeds. At very high engine loads, this is also the case. However, the difference in turbocharger speed at high loads is small for different engine speeds. Due to this, the difference in turbocharger speed before and after the step that has to be overcome during the boosting process is smaller at higher engine speeds. Also due to the engine working as a volumetric pump, the exhaust gas massflow is higher at higher engine speeds. This means that the power the turbine can extract from the
exhaust gas is higher at higher engine speeds. As a result of these two effects the duration of the boosting process decreases with increasing engine speed and the air mass consumption $m_b$ decreases, too.

The $t_{90}$-times are very fast over the complete operating range with the tendency to get slower for higher engine speeds and steps to higher engine loads. The reason for the increase with the engine load is that for higher loads higher intake manifold pressures are necessary. Reaching a higher pressure requires more time. The increase in the $t_{90}$-time with increasing engine speed can be explained intuitively. At higher engine speeds, the outflow of the intake manifold, i.e. $\dot{m}_\beta$, increases since the engine is working as a volumetric pump. The inflow, the maximum boost valve mass flow $\dot{m}_{bv,max}$, however remains constant for different engine speeds. This must lead to a slower increase of the intake manifold pressure. Another way to describe this was already presented in section 2.4: the gradient $\frac{dp_{im}}{dt}$ decreases for higher engine speeds. When the pressure increases slower, the engine mass flow $\dot{m}_\beta$ builds up slower as well and it takes more time until a certain $p_{me}$ is reached.

When the question on the necessity of a controlled overstream valve arises, the air masses that stream from downstream to upstream of the compressor during a boosting process is consulted. The results for Strategy I is shown in figure 4.6. To give a feeling for these values, the air mass in the receiver downstream of the compressor will be estimated. Assuming an average temperature of 350 K, an average pressure of 1.5 bar during the boosting process and using the identified volume of 5 l for that receiver, the ideal gas law yields an air mass inside that receiver of 7.5 g. With this strategy up to 3.9 g are recirculated to the compressor entry, that is equal to 52% of the mass inside the receiver. This amount of additional air would lead to a significant increase in the boost pressure if it was not recirculated. With this strategy, the overstream valve is necessary to avoid surge.

Since especially at low engine speeds and load steps to high engine loads the air
mass consumption is very high (see figure 4.5) a second strategy is developed to improve that. At low engine speeds it is possible to deliver much higher boost valve mass flows than required to produce the desired $p_{mc}$. Since a retardation of the ignition efficiency has proven to be beneficial for the air mass consumption, the second strategy involves overboosting.

4.2 Strategy II

For this strategy, the desired air mass from the torque inversion model is multiplied with an "overboosting factor $\nu_{ob}$", $\nu_{ob} > 1$. Like this it can be ensured, that more air than necessary to produce the desired $p_{mc}$ will be delivered by the boost valve. At the same time, the ignition controller is used to prevent the $p_{mc}$ from overshooting the desired value. This higher engine mass flow can only be caused by a higher intake manifold pressure. The intake manifold pressure during the boosting process increases to much higher values than the steady state pressure after the load step. This means that the end of the boosting process can no longer be determined by the criterion that the boost pressure has increased to the intake manifold pressure. For this strategy, the boost valve is closed once a certain turbocharger speed is reached. Then the intake manifold pressure decreases and the throttle is opened when the intake manifold pressure has decreased to the value of the boost pressure. For the following simulations, a value of $\nu_{ob} = 1.5$ was chosen.

4.2.1 Result for the Reference Load Step

The desired $p_{mc}$ is reached after 126 ms and the air mass consumed during this boosting process is 12.45 g. In the second plot it can be seen that the intake manifold pressure increases much above the steady state pressure after the load step. Also, the exhaust gas backpressure is higher than in steady state at the end of the boosting process. The boost valve remains completely open until the desired mass flow which is 50% higher than the steady state value is reached. As can be seen in the second plot of figure 4.7 the boost valve is closed before the end of Phase II (for the result shown at 0.30 s). From this point until the end of Phase II (at 0.37 s), air gets sucked out of the intake manifold and the intake manifold pressure decreases to the steady state value. In that same time the boost pressure and the turbocharger speed increase to their steady state values. The throttle is opened once the intake manifold pressure has reduced to the value of the boost pressure. In the bottom plot the efficiency reduction $1 - \epsilon_c$ by the ignition controller can be seen. As soon as the engine air mass flow gets bigger than necessary to produce the desired $p_{mc}$, the efficiency is reduced to prevent the $p_{mc}$ from overshooting. At the beginning of Phase III, the ignition efficiency is still slightly reduced.

4.2.2 Discussion

The speed-up of the turbocharger is very fast with this strategy. This is due to the fact that the power the turbine extracts from the exhaust gas mass flow is very high with this strategy. The turbine power $P_t$ is given by:

$$P_t = \dot{m}_t \cdot c_p \cdot \bar{\theta}_{enm} \cdot \left[ 1 - \Pi_t^{\frac{1}{\kappa_t}} \right] \cdot \eta_t$$  \hspace{1cm} (4.2)
From equation 4.2 it can be directly seen that the turbine power in Strategy II gets much higher in Phase II, due to the increased exhaust gas mass flow since $\nu_{ob} = 1.5$ is used. In addition to that, the exhaust gas temperature $\vartheta_t$ is higher when delaying the ignition angle as can be seen in formula 3.5. Due to the fact that turbines can be modeled as orifices according to [3], it becomes clear that also the exhaust gas backpressure and hence the turbine pressure ratio $\Pi_t$ must increase due to the higher turbine mass flow.
In conventional turbocharged engines (without indirect boosting), reducing the ignition efficiency is not an effective measure to improve the torque response. On a conventional turbocharged engine only the mass flow of the naturally aspirated full load operation can be reached quickly. If the ignition efficiency is reduced in that case, the torque that can be reached quickly is even lower than the full load torque of the naturally aspirated operation. However, for an engine with indirect boosting this strategy works very well since in a very short time an engine mass flow even higher than the full load mass flow can be delivered. There is sufficient air to deliver the full load torque even at a very much reduced ignition efficiency.

One disadvantage of the overboosting strategy is that additional fuel is needed. Since up to 50% more air is aspirated, also up to 50% more fuel is required. However, the duration of this phase when more fuel is needed is only very short, \( \approx 300 \text{ ms} \) for this load step. For the reference load step this strategy results in an additional fuel consumption of 111 mg. This fuel is not wasted, one way to interpret the additional fuel consumption is that it is used to accelerate the turbocharger, since the speed-up is very fast. However, this additional fuel consumption is rather small; assuming a driving profile where the reference load step is carried out every 300 m an additional fuel consumption of only \( 0.05 \frac{\text{l}}{100 \text{km}} \) is necessary.

Once the boost valve is closed at the end of Phase II, the intake manifold gets sucked empty until the throttle opens. The duration of this transition period could not be determined analytically. It is implemented that the boost valve closes once a certain turbocharger speed \( \omega_{TC, \text{off}} \) is reached. This value is calculated as a fraction of the steady state turbocharger speeds before the load step, \( \omega_{TC, ss, 1} \) and after the load step \( \omega_{TC, ss, 2} \):

\[
\omega_{TC, \text{off}}(\omega_e, p_{me}) = \omega_{TC, ss, 1} + f_{\text{off}}(\omega_e, p_{me}) \cdot [\omega_{TC, ss, 2} - \omega_{TC, ss, 1}] \quad (4.3)
\]

(The dependencies of \( \omega_{TC, ss, 1} \) and \( \omega_{TC, ss, 2} \) on the engine speed and the engine load have been omitted due to reasons of space)

A low value of \( f_{\text{off}}(\omega_e, p_{me}) \) will lead to a too early end of the boosting process, meaning the boost valve will close early and the intake pressure will decrease to a value lower than in steady state. In this case the \( p_{me} \) cannot be kept at the desired level and will collapse for a short period of time. If the value of \( f_{\text{off}}(\omega_e, p_{me}) \) is high, the boosting process will take slightly longer and the air consumption will increase. However this will not cause a collapse of the \( p_{me} \), since like this more air than necessary is available. Therefore the ignition will still be retarded in Phase III. In the fourth plot, this slight retardation of the ignition angle at the beginning of Phase III can be seen. The value for \( f_{\text{off}}(\omega_e, p_{me}) \) had to be tuned for several operating points across the engine operating map. For the other operating points the value can be interpolated.

### 4.2.3 Surge Tendency for Strategy II

What is remarkable about Strategy II is that the OSV can remain closed. This means that the operating point of the turbocharger does not get to the surge-line with this strategy. The reason for this is the extreme acceleration of the turbocharger. This leads to a shorter duration of the entire boosting process.
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4.2.4 Various Overboosting Factors

For the reference load step the results for the air mass consumption $m_b$ for various values of $\nu_{ob}$ are shown in figure 4.9. It can be seen that the higher the overboosting factor $\nu_{ob}$ is chosen the less air mass from the tank is required for the load step. However, the curve $m_b$ over $\nu_{ob}$ is steep for low overboosting factors and flattens out the higher $\nu_{ob}$ is chosen.

In figure 4.10 it is shown how the intake manifold pressure, the turbocharger speed, the consumed air mass and the efficiency reduction develop over time for the different values of $\nu_{ob}$. The higher the overboosting factor is, the higher the intake manifold pressure gets during Phase II. The turbocharger accelerates
4.2. Strategy II

Figure 4.9: Air mass consumption $m_b$ for various overboosting factors $\nu_{\text{ob}}$ for a load step from 1.7 bar to 17 bar $p_{\text{me}}$ at $\omega_e = 2000$ rpm.

The curves for the consumed air mass are normalized with the air mass $m_{b,\text{max}}$ that is consumed for the load step with an overboosting factor of $\nu_{\text{ob}} = 1.0$. It can be seen that with a higher overboosting factor more air is consumed in a shorter time, but the time until the boost valve closes is shorter. At the beginning of the load step all curves are on top of each other. The lower the overboosting factor is the earlier the boost valve mass flow is reduced which can be seen as a reduction of the gradient of the curves for the consumed air mass.

The end of Phase II is when the boost pressure and the intake pressure are equal and the turbocharger has reached its steady state speed. This indicates the duration of the boosting process. Figure 4.10 shows that a short duration of Phase II is advantageous for the air mass consumption. Supplying more air faster for higher overboosting factors and the efficiency reduction increases.
mass in a shorter time leads to a much higher intake manifold pressure and hence engine mass flow. Reducing the ignition efficiency to prevent an overshoot of the $p_{me}$ leads to more exhaust gas enthalpy and hence to a faster acceleration of the turbocharger. Even for low overboosting factors significant savings in $m_b$ can be achieved. The reason that the curve $m_b(\nu_{ob})$ flattens out can be seen in plot four of figure 4.7. The period of time during which the engine mass flow actually is equal to $\nu_{ob} \cdot m_{\beta,des}$ is only short. The higher $\nu_{ob}$ is chosen, the shorter this period of time is and hence the effect of increasing $\nu_{ob}$ gets smaller for higher values of $\nu_{ob}$. Since the acceleration of the turbocharger and the duration of the boosting process have an influence on the surge tendency, it should be mentioned that surge can be avoided when a value $\nu_{ob} \geq 1.3$ is chosen.

4.2.5 Results for the Entire Operating Range

For the discussion of the overall results with this strategy, a remark on the overboosting factor $\nu_{ob}$ is helpful. Since the maximum boost valve mass flow is designed to match the engine mass flow at 4000 rpm and 17 bar $p_{me}$, at that operating point no higher mass flow can be delivered which means that $\nu_{ob} = 1$ for that load step. So even though all the simulations were carried out with $\nu_{ob} = 1.5$ this is not possible at all load steps and $\nu_{ob}$ can therefore be seen as a maximum overboosting factor. The lower the engine speed gets, the more the ratio $m_{b, max} / m_{b, des}$ increases. From this it can be concluded that the possibility to conduct overboosting increases for lower engine speeds.

Keeping this in mind it can be explained that the lower the engine speed is, the more air mass $m_b$ can be saved. The reason is that with $\nu_{ob} > 1$ less time is needed for the acceleration of the turbocharger as explained in section 4.2.2. At high engine speeds and rather low loads, e.g. at 4000 rpm and 12 bar $p_{me}$ no advantage can be achieved. In that load step the ignition efficiency is never reduced. This is due to the fact that once the engine mass flow gets high
enough so the ignition controller would have to reduce the ignition efficiency the boosting process is already over. The overall result for $m_b$ is the combination of these effects and consequently, the consumed air mass increases with the engine load but remains rather constant with the engine speed.

In case of Strategy II $m_{osv}$ remains below 0.17 g, which corresponds to 2.3% of the air mass in the receiver downstream of the compressor. According to the ideal gas law, an additional air mass of 0.17 g leads to an increase in boost pressure of only 34 mbar or 2.3% when compared to 1.5 bar. So if the OSV had remained closed during this boosting process, the boost pressure would be about this much higher. These numbers show the order of magnitude that is up for discussion here. Whether such a small increase in pressure would lead to surge cannot be answered with the simple models. Firstly of all because the MVM is not as precise as 2.3% and secondly because the surgeline cannot be determined precisely and may also be slightly different in transient operation and steads state measurements. So the statement, that the overstream valve can or cannot remain closed must be backed up by test bench results.

The additional fuel consumption is highest for the reference load step. It decreases with increasing engine speed, since the possibility to conduct overboosting also decreases with the engine speed. As previously stated the additional fuel consumption is very low.

### 4.3 Strategy III

Since load steps often start from low part load operating points, the intake manifold pressure is very low, i.e. $p_{im} \ll 1$ bar. When the throttle is opened at the beginning of a load step the intake manifold pressure quickly increases to ambient pressure independently from the speed of the turbocharger. The throttle closes and the boost valve opens once the intake manifold pressure has reached
ambient pressure. The previously discussed Strategy II that involves overboosting has lead to the best results so far considering the air mass consumption. Strategy III is therefore only regarded in combination with overboosting.

4.3.1 Result for the Reference Load Step

Figure 4.13 shows the result for the reference load step. The throttle opens for $\approx 90$ ms. In this time, the intake manifold pressure increases to ambient conditions and the $p_{me}$ builds up to 7 bar. The turbocharger speed doubles during that short period of time. Then the throttle closes and the boost valve opens. The intake manifold pressure increases until a 50% higher engine air mass flow than necessary for $p_{me, des}$ is reached. From the time the boost valve opens the $p_{me}$ increases quickly to the desired value. The ignition efficiency then is reduced to prevent an overshoot of $p_{me}$. With this strategy, the OSV is opened shortly after the throttle has closed. The $t_{90}$-time is 172 ms and the air mass consumption is 9.78 g.

4.3.2 Discussion

The $p_{me}$ build-up could be affected with the ignition controller as shown in section 4.1.3. With this method, the $p_{me}$ build-up could be smoothed out. The air mass that can be saved in comparison to Strategy II can easily be estimated. Assuming an intake manifold temperature of 300 K, a pressure of 0.4 bar and using the value for the intake manifold volume of 2.4 l, the ideal gas law yields:

$$\Delta m_b = \frac{V_{im}}{R \cdot T_{im}} \cdot \Delta p_{im} = 1.67 \text{ g}$$  \hspace{1cm} (4.4)

This air mass does not have to be supplied by the boost valve from the pressure tank. Even more air than estimated is saved with this strategy. The reason for this is that the turbocharger speed increases during the time the throttle is open. From that increased speed less time is required until the turbocharger has accelerated to its final speed. In that shorter time, less mass flows over the boost valve. However, due to the modeling uncertainties of the turbocharger especially in the operating region of low pressure ratios and low mass flows this must be regarded with suspicion. The reason that the OSV opens with this strategy, is that in the moment the throttle closes the compressor mass flow has already increased do to the increased turbocharger speed. When a higher mass flow enters the receiver downstream of the compressor the pressure there will increase faster and this gives a surge tendency. When the turbocharger is accelerated afterwards, the operating point of the compressor moves away from the surgeline. The mass flow over the OSV is very small during the short time it is opened.

4.3.3 Results for the Entire Operating Range

Due to the fact that overboosting is also used with this strategy, the air mass consumption increases with the load and stays rather constant with the engine speed as shown in figure 4.14. The rise time also increases with the load. It is slowest for load steps to high engine loads at high engine speeds.
With this strategy $m_{\text{osv}}$ remains below 0.06 g which corresponds to 0.8%. Following the estimations made in 4.1.5, not opening the OSV would lead to an increase of 12 mbar or 0.8% when compared to 1.5 bar. Also in this case it is not clear if surge would occur due to model uncertainties.

The extra fuel consumption for this strategy also decreases for higher engine speeds. It is highest for load steps to high engine loads at low engine speeds.
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Figure 4.14: The consumed air mass $m_b$ and the rise time $t_{90}$ for Strategy III for load steps from 1.7 bar to $p_{me}$.

Figure 4.15: The air mass $m_{osv}$ that streams from downstream to upstream of the compressor during a boosting process and the additional fuel consumption for Strategy III.

4.4 Strategy IV

In order to quantify the statement that a small additional mass flow during a load step while the throttle is open will only improve the performance slightly, yet another strategy is investigated. Here the throttle opens completely during the load steps just as in conventional turbocharged operation. The boost valve controller is modified to supply a mass flow small enough to avoid surge, but to move the operating point of the compressor close to the surge line. Input to this modified boost valve controller is the difference between the compressor pressure ratio at the surge line $\Pi_{c,surge}(\dot{m}_c)$ and the compressor pressure ratio $\Pi_c(\dot{m}_c)$. With this strategy the boosting process is over and the boost valve
is closed once the desired $p_{me}$ is reached. The conventional throttle controller does not have to be modified for this strategy.

4.4.1 Result for the Reference Load Step

A can be seen in Figure 4.16 the rise time is 1372 ms and the air mass consumption is 15.9 g. The $p_{me}$ increases quickly to 7 bar and then increases linearly to the
desired value. The boost pressure and the intake manifold pressure are equal during the time the throttle is completely opened. The OSV is not opened with this strategy. In the third plot of figure 4.16 it can be seen that the boost valve mass flow is always smaller than the engine mass flow. The moment the boost valve closes, the mass flow over the throttle increases to the steady state value. The ignition controller is not required with this strategy.

In the first 50 ms after the load step the mass flow over the throttle is very high, then decreases again rapidly and stays at a rather low value. Also the boost valve mass flow is very high directly after the load step.

The gray curve in figure 4.17 shows the trajectory of the compressor operating point for this strategy.

4.4.2 Discussion

This load step is basically like a load step with a conventional turbocharged engine. The trajectories of basically all signals are essentially similar to the ones without boosting. The only difference is that everything happens faster now, the gradient of the effective mean pressure $\frac{dp_{me}}{dt}$, the intake manifold pressure $\frac{dp_{im}}{dt}$ and the turbocharger speed $\frac{d\omega_{TC}}{dt}$ are steeper.

The reason for the high mass flows directly after the load step is as follows. There are no dynamics in the model and the throttle opens completely from one timestep to the next. Due to the relatively large pressure ratio over the throttle in that moment and the high flow cross-section of the completely opened throttle a high mass flow is induced. However, since it only takes a very short time until the boost pressure is equal to the intake manifold pressure, the throttle mass flow $\dot{m}_a$ reduces again very quickly. At the beginning of the load step the boost pressure reduces until the pressure balance is reached with the intake manifold pressure. The operating point in the compressor map is then shifted towards higher mass flows for a short time as can be seen in figure 4.17. Since at a higher mass flow a higher surge pressure ratio is present, the boost valve opens completely for this short period of time. Note that these very high values only occur since the actuators have no dynamics in the model. If throttle dynamics were included the throttle mass flow would only increase slowly. In that case the operating point of the compressor would not move to higher mass flows for a short time and the boost valve also would not open completely at the beginning of the load step.

The reason that an improvement compared to the conventional turbocharged operation can be achieved is the following: when the boost valve is opened and brings the compressor operating point to the surgeline, it moves there approximately along an iso speedline of the compressor since the compressor speed can only change much slower than the pressure ratio. Since the slope of the iso speedlines is negative, a higher pressure ratio is possible at the surgeline than at an operating point on the same compressor speedline with a higher mass flow. This leads to a slight increase of the boost pressure and intake manifold pressure and more air is aspirated by the engine. This increased air mass leads to a higher $p_{me}$ and hence more exhaust gas enthalpy. This on the other hand accelerates the turbocharger faster. At higher turbocharger speeds, the compressor pressure ratio ratio at the surgeline is higher which again leads to a higher engine mass flow. When the boost valve closes once the desired engine mass flow
is reached the complete engine mass flow is supplied by the compressor again. This can be seen in the bottom plot of figure 4.16 at 1.6 s. The boost valve mass flow $\dot{m}_{bv}$ decreases to zero and the throttle mass flow $\dot{m}_{\alpha}$ increases and is equal to the engine mass flow $\dot{m}_{\beta}$. The compressor mass flow then is equal to the throttle mass flow.

Regarding the high air mass consumption that results from this strategy the formulation an "additional mass flow is supplied" may be misleading. When following a constant speed line of the compressor to the surge line the pressure ratio increases slightly while the compressor mass flow decreases significantly. This means that to enable the slightly higher pressure ratio, a significant part of the engine mass flow must be supplied by the boost valve, since the compressor mass flow is reduced by this measure. Since the "additional mass" also must be supplied for a relatively long time, the air mass consumption is high with this strategy.

![Figure 4.17: Operating point of the compressor during load steps from 1.7 bar to 17 bar $p_{me}$. Case a) is at 2000 rpm, case b) is at 4000 rpm, both without boosting. The gray line (Strategy IV) is at 2000 rpm with turbo lag compensation. The dotted black line represents the threshold of 0.97 $\Pi_{c, surge}$.](image)

It should be noted that this strategy is only possible if the slope of the compressor iso speedlines $\frac{d\Pi_{c}}{d\dot{m}_{c}}$ is negative. If the slope was zero, the pressure ratio could not be increased by moving the operating point of the compressor to the surge line and no beneficial effects could be achieved. For some compressors the slope of the iso speedlines is even positive close to the surge line. In that case boosting with this strategy would even have a negative effect.

### 4.4.3 Results for the Entire Operating Range

In figure 4.18 the air mass consumption $m_b$ and rise time $t_{90}$ are shown for this strategy. The air mass consumption is rather high with this strategy, it reduces with engine speed and engine load. The increase for higher engine loads can be explained with the constant gradient of the effective mean pressure. It takes a longer time to reach a higher effective mean pressure so the time that the boost
valve must keep the operating point of the compressor close to the surgeline hence is longer at higher loads. Therefore, the required air mass $m_b$ increases. The decrease in air mass consumption for higher engine speeds can be explained similarly. A higher boost valve mass flow can be supplied at higher engine speeds because the operating point of the compressor is further away from the surgeline as can be seen in figure 4.17. However this effect gets overcompensated by the much shorter boost duration. The boost duration is slightly longer than the rise time, since the boosting process is over when the desired torque is reached and not when 90% are reached as it is the case for the $t_{90}$-time.

With this strategy the rise times get shorter for higher engine speeds. This can be explained by the fact that now the boost valve rather supplies an additional mass flow. Now the system behaves basically like a conventional turbocharged engine and the rise times get shorter at higher engine speeds. The increase of the rise time with the engine load can be explained by the fact that the torque increases linearly.

With this strategy, the OSV can remain closed and no additional fuel is needed.

### 4.5 Comparison of the Various Strategies

At the beginning of this section a short summary of the various strategies is given and the major differences in the control strategies are discussed. After that the results are compared for the reference load step as well as for the entire operating range.
Strategy I: At the beginning of the load step, the throttle is closed and the boost valve supplies the desired engine mass flow. The boosting process is over once the boost pressure has reached the level of the intake manifold pressure.

Strategy II: Also here the throttle is closed at the beginning of the load step. The boost valve supplies a mass flow higher than the desired engine mass flow and an intake manifold pressure higher than the steady state pressure after the load step is reached. To prevent an overshoot of the $p_{\text{me}}$ the ignition efficiency is reduced. Once the turbocharger has reached a certain speed the boost valve is closed. Then the intake manifold pressure reduces and the throttle opens when the boost pressure is equal to the intake manifold pressure.

Strategy III: The throttle is opened at the beginning of the load step until the intake manifold pressure reaches ambient pressure. Then the throttle is closed and the boost valve is opened. From then on, this strategy is equal to strategy II.

Strategy IV: The throttle is opened throughout the entire loadstep. The boost valve supplies an additional mass flow.

The major differences are the following: With Strategy I and II the throttle is closed throughout the entire boosting process. With Strategy III it is open for a short time at the beginning of the boosting process and with Strategy IV it is open throughout the entire boosting process. Strategy I and IV have the advantage that the determination of the moment the boost valve can close is straightforward. In Strategy I the boost valve is closed at the end of the boosting process which is when the boost pressure is equal to the intake manifold pressure. In Strategy IV the boost valve is closed when the desired engine air mass flow $\dot{m}_\beta$ is reached. For Strategy II and III this is not straightforward. A threshold $f_{\text{off}}(\omega_e, p_{\text{me}})$ has to be determined for several operating points. This threshold could not be determined analytically.

In the following the results will be compared. At first, the results of the reference load step with the different strategies are compared. After that, a comparison of the results obtained over the complete operating range follows.

### 4.5.1 Comparison of the Results for the Reference Load Step

Table 4.1 shows the numerical results for the rise time $t_{90}$ and the air mass consumption $m_b$. Also an estimate for the surge tendency is given. The figures of the reference loads step can be found in the corresponding chapters on page 16 for Strategy I, page 23 for Strategy II, page 30 for Strategy III and on page 32 for Strategy IV.

The whole idea of indirect boosting is to overcome the turbo lag. So the criterion to quantify that effect is the rise time.

The slowest rise time is achieved with Strategy IV. It is twice as fast as the rise time without boosting, however $\approx 10$ times slower than with the other strategies. Strategy I and Strategy II are basically equally fast. The small advantage of
Table 4.1: Results for the Reference Load Step for all Strategies

<table>
<thead>
<tr>
<th>strategy</th>
<th>$t_{90}$</th>
<th>$m_b$</th>
<th>surge tendency</th>
</tr>
</thead>
<tbody>
<tr>
<td>Strategy I</td>
<td>137 ms</td>
<td>18.35 g</td>
<td>high</td>
</tr>
<tr>
<td>Strategy II</td>
<td>126 ms</td>
<td>12.45 g</td>
<td>none</td>
</tr>
<tr>
<td>Strategy III</td>
<td>172 ms</td>
<td>9.78 g</td>
<td>low</td>
</tr>
<tr>
<td>Strategy IV</td>
<td>1372 ms</td>
<td>15.90 g</td>
<td>none</td>
</tr>
</tbody>
</table>

Strategy II is due to the fact that there the $p_{mc}$ rises linearly until the desired $p_{mc}$ is reached while with Strategy I the gradient $\frac{dp_{mc}}{dt}$ decreases when the desired $p_{mc}$ is almost reached. Strategy 3 is slightly slower than Strategy I and Strategy II. The opening of the throttle at the beginning of the loadstep leads to a slower increase of the $p_{mc}$ than the opening of the boost valve with the throttle closed.

The air mass consumption is highest for Strategy I. This is due to the fact that the turbocharger needs a rather long time to accelerate. When comparing Strategy I and strategy II, the time until the boost valve closes again is more than twice as long for Strategy I. With Strategy II a higher mass flow is supplied during this time due to the overboosting but this is overcompensated by the much shorter time the boost valve has to remain open. Strategy III has the lowest air consumption since it basically is equal to Strategy II with two additional benefits. First of all the air mass to increase the intake manifold pressure to ambient pressure must not be supplied by the tank, secondly because the turbocharger accelerates during this time and only a smaller difference in turbocharger speed must be overcome during the boost valve is open. One major difference Strategy IV has compared to the other strategies is that not the complete engine mass flow must be supplied by the boost valve. The time the boost valve is open here is approximately twice as long as compared to Strategy I. However, less air is required since only a fraction of the engine mass flow is supplied by the boost valve.

The major advantage Strategy IV has over all other strategies is that with this strategy no surge has to be expected. The boost valve controller in this strategy is especially designed to avoid surge. With Strategy I the surge tendency is the highest since the boosting process takes very long compared to Strategy II and III. Even with a low compressor mass flow the compressor discharges more air mass in total during this long time. Without opening the OSV, this would lead to a very high boost pressure and hence drive the compressor into surge. Using Strategy II the overstream valve remains closed. Due to the very high acceleration of the turbocharger and the short duration of the entire boosting process the operating point of the compressor moves away from the surgeline. This is basically also the case for Strategy III, however due to the increased turbocharger speed when the throttle is closed the operating point of the compressor moves to the left of the surge line for a short period of time. The OSV opens, however the mass recirculated is very low so it is unclear if surge would occur.

Reducing the ignition efficiency is necessary for Strategy II and III due to the overboosting, since with a higher engine air mass flow the $p_{mc}$ shall not overshoot.
the desired value. This is disadvantageous regarding fuel consumption, since a part of the fuel would not be needed to produce the desired $p_{mec}$ but is rather used to increase the exhaust gas temperature. This measure on the other hand significantly reduces the required air mass. The amount of extra fuel needed is estimated for a very ambitious operational profile in section 4.2 and even then only leads to a very low extra fuel consumption of 0.05 l/100 km. Strategy I and Strategy IV do not need an efficiency reduction since $p_{mec}$ does not overshoot the desired value.

4.5.2 Comparison of the Results for the Entire Operating Range

The figures that show the results for the complete operating range of the different strategies can be found on page 29 for Strategy I, page 27 for Strategy II, page 31 for Strategy III and on page 35 for Strategy IV.

Also when regarding the whole operating range Strategy IV still has the slowest rise times. The rise times are significantly improved when compared to the conventional turbocharged operation, they are $\approx 24\%$ to $58\%$ shorter. Only small improvements are possible at low engine speeds and load steps to rather low engine loads. At higher engine speeds and loads a greater improvement can be achieved. At 4000 rpm the rise times are roughly equal to the ones achieved with Strategy I and II, while at low engine speeds they are about a factor of 10 slower. Only with Strategy IV the rise times get shorter with increasing engine speed. For all other strategies this correlation is inverse: the rise time increases when the engine speed increases. The rise times of Strategy II are the shortest, however only very little improvement can be achieved compared to Strategy I.

The reason for this is that for Strategy I the gradient $\frac{dp_{mec}}{dt}$ decreases when the desired $p_{mec}$ is almost reached. The difference is smaller at higher engine speeds. Strategy II is slower than Strategy III. At 2000 rpm the rise time is $\approx 50\text{ ms}$ longer while at 4000 rpm the rise time is $\approx 10\text{ ms}$ longer. This is due to the fact that the intake manifold dynamics get faster with increasing engine speed and it takes less time for the intake manifold pressure to increase to ambient pressure at higher engine speed. Since this pressure increase always happens at the beginning of a load step, at any engine speed the difference in the rise time is the same for load steps to all engine loads in the considered range. For all strategies load steps to higher engine loads require more time.

Regarding higher engine speeds than 4000 rpm indirect boosting cannot be carried out with strategies 1, 2 and 3 because the boost valve massflow is too low. However it can be expected, that with Strategy IV, an additional mass flow can be supplied even at higher engine speeds. Even though the rise times are rather fast at high engine speeds they could still be improved with Strategy IV.

In both Strategy I and Strategy IV, the required air mass increases with the engine load and also with the engine speed. In both Strategy II and Strategy III the air mass consumption depends on the engine load but is rather independent from the engine speed. With Strategy II a significant improvement can be achieved compared to Strategy I: up to $35\%$ less air is needed at low engine speeds. At high engine speeds, no improvement can be attained. With Strategy III an average of $\Delta m_b = 2.8\text{ g per loadstep}$ can be saved compared to Strategy II.
Strategy IV is the only strategy that completely avoids surge over the whole operating range. With Strategy I at low engine speeds only by opening the OSV surge can be avoided. With Strategy II and Strategy III almost over the whole operating area the OSV is opened for a short period of time but the air mass recirculated is always very small.

In strategies I and IV the ignition efficiency is never reduced. Therefore no additional fuel is needed. In strategies II and III where overboosting is carried out the reduction of the ignition efficiency leads to a very low additional fuel consumption.
4.5. Comparison of the Various Strategies
Chapter 5

Indirect Boosting on the Test Bench

In the experimental part of the thesis indirect boosting was also investigated on the test bench. In this section all the necessary prerequisites to carry out load steps and evaluate the results are described and discussed.

5.1 Boost Valve Implementation

For the implementation on the test bench a solenoid proportional valve was chosen. It can deliver the required mass flow and withstand a tank pressure of up to 12 bar. The boost valve was installed on top of the intake manifold and connected to the pressure tank. Air being injected by the boost valve impacts the area right between the two aspiration ports. The actuation of the boost valve occurs with a PWM signal.

5.1.1 Test Bench Setup during the Boost Valve Identification

For the determination of the boost valve parameters, the configuration of the test bench was changed. The engine was stopped at a position where both intake valves were closed. During the measurements the engine was at standstill. Air that is injected by the boost valve now enters the intake manifold and leaves through the throttle in the opposite direction as during conventional engine operation. Downstream of the throttle, a long pipe was attached, which had a sensor (HFM) to measure the mass flow rate installed in its middle (see figure 5.1). With this setup, it is possible to directly measure the boost valve mass flow. Since no information of how the inside of the valve looks, it is not possible to calculate the area $A$ and the discharge coefficient $c_d$ separately but only the product of the two $A_{eff} = A \cdot c_d$. The idea was to be able to compute the effective area at various PWM signals, therefore measurements are required for the tank pressure, the tank temperature, the intake manifold pressure and the mass flow. $A_{eff}$ can then be determined by rearranging equation (5.1). It must be mentioned that the tank pressure that is used here is the measured pressure in the pressure
5.1. Boost Valve Implementation

Figure 5.1: Engine setup for the boost valve identification measurements.

tank, not the measured pressure directly upstream of the boost valve since the equations for isentropic flow call for the total pressure. Due to the rather thin pipe, high flow velocities upstream of the boost valve must be expected and the measured (static) pressure at that location will be much lower than the pressure measured in the tank.

5.1.2 Results of the Boost Valve Identification

The complete boost valve identification can be found in appendix A. This section gives an overview over the most important results. When the boost valve is opened or closed completely in the measurement setup, the mass flow measured with the HFM increases or decreases like a first order element. The time constant of the boost valve \( \tau_{bv} \) was found to be \( \tau_{bv} = 15 \text{ ms} \). Also a small time delay is present. This could not be measured with this test bench setup but was identified with the first load steps. The delay of the boost valve is 5 ms.

The steady state measurements for various boost valve openings only showed disappointing results. The operating line of this valve can not be used for feedback control, since not every valve opening \( A_{eff} \) can be realized because the operating line contains several steps. Since the boost valve is very fast another attempt was chosen to achieve various steady state mass flows: fast opening and closing of the boost valve, which corresponds to a very low PWM frequency, was found to deliver a strongly oscillating massflow whose average increases linearly with the PWM-signal. The best result was obtained with a PWM-frequency of 10 Hz. Higher PWM-frequencies lead to nonlinear operating
lines. An other object that was found is that the operating line alters for various intake manifold pressures. So the operating line had to be extended to an operating map for various intake manifold pressures, see figure 5.2. What was measured is the pulsating mass flow, the effective area is calculated with the mean value of this mass flow. The intake pressure also oscillates when the boost valve is operated like this, for example at an intake pressure of 2 bar the amplitude of the oscillations is on the order of 0.2 bar.

![Operating map of the boost valve for quick opening and closing.](image)

Figure 5.2: Operating map of the boost valve for quick opening and closing. \( A_{eff} \) is calculated from the mean value of the measured pulsating mass flow.

### 5.2 Engine Mass Flow Observer and Predictor

The engine mass flow can only be measured directly in stationary operation. Then the mass flow \( \dot{m}_\beta \) is equal to the mass flow measured with the HFM \( \dot{m}_{HFM} \). In transient engine operation however, this value is not accurate since the HFM is located upstream of the compressor. To estimate the engine mass flow better in transient engine operation an observer and a predictor are included in the engine control at the testbench. Given that especially during indirect boosting, \( \dot{m}_\beta \) and \( \dot{m}_{HFM} \) are completely uncoupled from each other the observer and predictor are modified for this work.

Purpose of the observer is to estimate how much air is being aspirated by the intake valves in the current engine cycle. For the observer an intake pressure filter is implemented as described in [1]. Based on a model for the throttle mass flow \( \dot{m}_\alpha \) and the measurement of the intake manifold pressure the intake pressure filter reduces the pumping noise and also calculates a correction state \( m_{air,corr} \).
The engine mass flow is then calculated by subtracting the correction state from the value calculated with the volumetric efficiency model and the filtered intake manifold pressure as described by equation 2.7.

In a port fuel injection engine the fuel must be injected before the intake valves are opened. That means that during engine transients when the intake manifold pressure changes rapidly, it must be known in advance how much air will be aspirated in the next engine cycle to calculate the amount of fuel that needs to be injected. Purpose of the predictor hence is to give an estimate on how much air will be aspirated in the next engine cycle in order to keep deviations in the air to fuel ratio at a minimum. Since the gradient of the intake manifold pressure is also calculated by the observer, this estimate is done by assuming a constant pressure gradient until the end of the prediction time. The pressure gradient is corrected by an estimate on how big the inflow into the intake manifold will be at the end of the prediction horizon. With this corrected pressure gradient the intake manifold pressure at the prediction time is calculated.

Both observer and predictor require models for the mass flows into and out of the intake manifold. The outflow can be calculated as ever, however the inflow into the intake manifold does not come from the throttle anymore. Since the throttle is closed, the inflow comes from the boost valve.

To clarify why the observer was modified in the way described subsequently, first a chain of causality is presented:

- From the output of the boost valve controller, the desired effective area $A_{\text{eff,des}}$ that is necessary to deliver the desired boost valve mass flow is calculated.

- This signal then is processed by the boost valve operating map. Here, two things are important. On one hand the value for the maximum possible effective boost valve area $A_{\text{eff,max}}(p_{\text{in}})$ at the current intake manifold pressure is provided. On the other hand, the necessary PWM-signal for the corresponding $A_{\text{eff,des}}$ is allocated.

- In the pulse generator, the PWM-signal is processed to either send the "open" or "close" command to the power electronics of the boost valve.

- The observer gets the information on the current value of $A_{\text{eff,max}}$ and if the "open" or the "close" signal is sent to the valve by the power electronics.

With these information the boost mass flow can easily be estimated. Since the boost valve is identified to behave like a first order system, the input to that system when the boost valve is "open" is the value of $A_{\text{eff}} = A_{\text{eff,max}}$. When the boost valve is supposed to close, the value of $A_{\text{eff}} = 0$ is input to the first order system. The output of this first order system is the current value of $A_{\text{eff}}$ of the boost valve. This value is then processed with the formulas for isentropic flow to provide the value for the mass flow that currently streams through the valve. In this fashion the pulsating mass flow can be reproduced. Figure 5.3 illustrates this behaviour.

The mass flow that is output from the first order element then input to the estimator, since it requires all the mass flows into the intake manifold to calculate the outflow into the engine $\dot{m}_g$. 
The predictor also gets the same signals as the observer, however the predictor is not a first order element but a lag element. Since it is the task of the predictor to give information on how the mass flow several time steps ahead will be an additional input to the predictor is the prediction horizon. The prediction horizon constantly changes during engine operation. Output of the lag element is $A_{eff}$ of the boost valve at the end of the prediction horizon. This is used to calculate the boost valve mass flow at the end of the prediction horizon.

5.3 Ignition Controller on the Test Bench

Due to the major importance of the ignition controller for the strategies developed, special care must be taken when implementing it at the testbench.

5.3.1 Implementation of the Ignition Controller

The ignition timing has a significant influence on the engine torque produced. Following [3], for every engine operating point, there is an optimal (maximum brake torque, MBT) ignition angle denoted as $\zeta_{MBT}(\omega_e, p_{me})$. This ignition angle varies strongly, especially with the engine load $p_{me}$. The ignition angle that is realized on the engine is the MBT ignition angle except when engine knock is present. This ignition angle is typically saved in an ignition angle map $\zeta_{map}(\omega_e, p_{me})$. When deviations from the MBT ignition angle are present, the produced torque drops parabolically, independent of the operating point:

$$e_{\zeta}(\zeta) = 1 - k_{\zeta} \cdot (\zeta - \zeta_{MBT}(\omega_e, p_{me}))^2$$  \hspace{1cm} (5.1)

The constant $k_{\zeta}$ must be determined on the testbench so that the model can predict by how many degrees the ignition has to be retarded in order to realize a desired efficiency reduction. A series of measurements was carried out at 2000 rpm at an intake manifold pressure of 1 bar to determine the value of $k_{\zeta}$.
Therefore the ignition angle of both cylinders is adjusted to be the same angle and then varied to later and earlier timings than the value in the ignition angle map. The measured points are interpolated with a second order function and it was found that \( k_\zeta = 4.1 \cdot 10^{-4} \text{[}/\text{CA}^2\text{]} \). So at the testbench the ignition controller first calculates the desired efficiency reduction \( 1 - e_\zeta \) and then the amount of degrees that the ignition must be retarded to achieve this reduction by inverting equation 5.1. The ignition angle calculated to:

\[
\zeta = \zeta_{\text{map}} + \Delta \zeta = \zeta_{\text{map}} + \sqrt{\frac{1 - e_\zeta}{k_\zeta}} \tag{5.2}
\]

Just as in the model, at the testbench the correct value for the mass flow must be input to the ignition controller. Since the combustion, which is caused by the ignition, takes place roughly one revolution after the intake center (when the mass has been aspirated by the engine), the ignition controller gets the mass estimate from the observer delayed by the induction-to-power-stroke delay \( \tau_{\text{IPS}} \approx \frac{2\pi}{\omega_e} \).

### 5.3.2 Optimal Ignition Angles

At low engine speeds and high loads the MBT ignition angle can not be attained due to knock limitations. The ignition angle of the engine then has to be retarded. In order to know by how many degrees the ignition angle has to be delayed to achieve a desired efficiency reduction \( \Delta e_\zeta \), the offset between the MBT ignition angle and the ignition angle in the ignition angle map \( \zeta^*(\omega_e, p_{me}) \) must be known.

The ignition controller needs two operating maps. One map that contains the base ignition angles \( \zeta_{\text{map}}(\omega_e, p_{me}) \) and one that contains the offset from the ignition angle in the map to the MBT ignition angle \( \zeta^*(\omega_e, p_{me}) \).

![Figure 5.4: Consequences of neglecting the offset from the MBT ignition angle. Example is for \( k_\zeta = 4.1 \cdot 10^{-4} \) and \( \Delta e_\zeta = 0.15 \).](image-url)
Figure 5.4 illustrates how great the difference in \( \Delta \varepsilon \) can be when neglecting the offset \( \varepsilon^\ast (\omega_e, p_{me}) \) from the ignition angle in the map to the MBT ignition angle. In this example the mapped ignition angle has an offset of \( \varepsilon^\ast = 10^\circ \text{CA} \) to the MBT ignition angle. If an efficiency reduction of 15% is demanded and the offset is not considered, the ignition angle is delayed by:

\[
\Delta \varepsilon = \sqrt{\frac{1 - \varepsilon^\ast}{k}} = \sqrt{\frac{0.15}{4.1 \cdot 10^{-2}}} = 19.1^\circ \text{CA}
\]

This would however lead to an efficiency reduction of

\[
\Delta \varepsilon_{\text{MBT}} = 0.96 - \left(1 - k \cdot (\Delta \varepsilon)^2\right) = 0.96 - \left(1 - 4.1 \cdot 10^{-4} \cdot (19.1 + 10)^2\right) = 0.31
\]

since the second order fit for the efficiency reduction is only valid when \( \Delta \varepsilon \) is taken from the MBT ignition angle. The value 0.96 thereby corresponds to the ignition efficiency at the ignition angle from the map. This is not equal to one due to the offset of \( \varepsilon^\ast = 10^\circ \text{CA} \). To correct for this, the ignition angle retardation must be calculated by

\[
\Delta \varepsilon = -\Delta \varepsilon^\ast + \sqrt{(\Delta \varepsilon^\ast)^2 + \frac{1 - \Delta \varepsilon^\ast}{k}} 
\]

(5.3)

For this example, the ignition has to be retarded by 11.6\(^\circ\)CA from the ignition angle in the ignition angle map instead of 19.1\(^\circ\)CA, which then leads to the desired efficiency reduction of \( \Delta \varepsilon_{\text{MBT}} = 15\% \).

To find this offset and create the second ignition angle map \( \varepsilon^\ast (\omega_e, p_{me}) \) a series of measurements has been conducted. Thereby the ignition angle is varied from the ignition angle in the map to later and earlier angles for each cylinder individually. The ignition angle for the other cylinder was kept constant. In case of very high engine loads, the ignition was only delayed from the ignition angle in the map due to the knocking restriction. In those cases, the MBT ignition angle was attained by fitting a second order function to the measured values and extrapolating the curve to earlier ignition angles.

One thing that must be taken care of when conducting these measurements is that especially at high engine loads a reduction in torque is not the only consequence of delaying the ignition angle in a turbocharged engine. If the ignition occurs later, the exhaust gas temperature increases, that means that there is a higher enthalpy flow in the exhaust gas. As a consequence the turbine will extract more energy from the exhaust gas which then will lead to a higher turbocharger speed. This will on the other hand increase the boost pressure, so the throttle must be controlled in these measurements in order to keep the air mass aspirated in each cycle constant.

This was carried out for several engine operating points. Since the offset from the MBT ignition angle is a consequence of knocking this was mainly done at high engine loads and low engine speeds since knocking is mostly a problem there.

In figure 5.5 the results for 2000 rpm and 2500 rpm are shown. At higher engine speeds no such measurements were conducted. However, the trend can be seen that towards higher engine speeds the offset from the MBT ignition angle gets small and hence can be neglected. At higher engine speeds the knocking problem is not present anymore.
5.4 Further Preliminary Investigations

There are several other objects that must be taken care of in order to carry out and evaluate load steps successfully. This section deals with the throttle leakage and the determination of the tank size, which will become important for the simulation model to compare the measurements with simulations. Also the determination of the air mass $m_{\beta,cyl}$ in the measurements is depicted here. Lastly it is shown what implications the operation of the engine with the boost valve has.

5.4.1 Throttle Leakage

When it comes to the throttle, one parameter that has to be identified for indirect boosting is the throttle leakage. Since during indirect boosting the throttle is completely closed and the intake pressure $p_{im}$ can be significantly higher than the boost pressure $p_b$, backflow can occur from the intake manifold towards the compressor. In order to measure the intake manifold leakage, the measurement setup as described in 5.1.1 was helpful, too. Various pressures were adjusted in the pressure tank. Then the throttle was completely closed and the boost valve was completely opened. The same pressure now is present in the pressure tank and in the intake manifold. Since this volume is large compared to the throttle leakage mass flow, this pressure remains constant for a long enough period of time. The effective area of the throttle when completely closed was found to be $8.5 \cdot 10^{-6} \text{ m}^2$. This value is implemented in the simulation model for the comparison of the measurements and simulations.

5.4.2 Determination of the Pressure Tank Volume

The volume of the pressure tank was measured in the following way: the pressure tank was connected to the air supply and filled to a pressure of 9 bar. Once the tank temperature had reached steady state conditions the boost valve was completely opened and the air was released into the atmosphere. Since the effective flow area of the boost valve is known, the tank volume can be calculated.

---

Figure 5.5: Offset $\Delta \zeta^*$ from the MBT ignition angle at various engine speeds and loads, i.e. in cylinder air masses $m_{\beta,cyl}$.
from the pressure trace of the tank pressure. A model was set up, that consists of the boost valve and the pressure tank. The pressure tank was assumed to be adiabatic, which means that this is valid only for a short time after the boost valve opens, when no significant amount of heat has been transferred from the walls of the tank to the air inside. The tank volume in the model was modified by a numerical optimization routine until the measurement and the simulation showed well agreement.

### 5.4.3 Determination of $m_b$ in the experiments

The air mass consumed during a load step cannot directly be measured but it can be calculated from the temperature and pressure of the air in the tank before and after the load step with the ideal gas law.

$$
\Delta m_b = m_{t,1} - m_{t,2} = \frac{V_t}{R_{air}} \cdot \left(\frac{p_{t,1}}{\vartheta_{t,1}} - \frac{p_{t,2}}{\vartheta_{t,2}}\right)
$$

(5.4)

When air streams out of the tank, the pressure and hence the temperature decrease. The temperature of the air in the tank is measured, however, the temperature sensor used is very slow. So this means that the temperature after a load step cannot be measured. In order to calculate the air mass in the tank after the load step one could wait until the steady state temperature is reached again. In that case, however, the measurements would be very time-consuming and also an error would be made due to the small leakage of the pressure tank. Another possibility to obtain the temperature after the load step is to assume an isentropic expansion from the starting pressure to the final pressure. Then the final temperature $T_{t,2}$ can be calculated as:

$$
\vartheta_{t,2} = \vartheta_{t,1} \cdot \left(\frac{p_{t,2}}{p_{t,1}}\right)^{\frac{1}{\kappa}}
$$

(5.5)

Since this is a very strong assumption and also the measured air mass is of great importance the validity of this approach will be strengthened with a simplified example. The temperature after an isentropic expansion $T_{t,2,s}$ is the lowest temperature that can be achieved. In the presence of heat transfer this temperature would be higher. The heat transfer from the inner walls of the tank to the air is proportional to the temperature difference: $\dot{q}'' \sim \Delta \vartheta$. So the temperature difference due to heat transfer that the mass $m_{t,2}$ experiences calculates to:

$$
\Delta \vartheta = \frac{1}{m_{t,2} \cdot c_v} \cdot A_t \cdot h \cdot (\vartheta_{1} - \vartheta_{t,2,s}) \cdot t_b
$$

(5.6)

This simplified calculation can show the order of magnitude of the change in tank temperature during the boosting process due to heat transfer. Assuming the temperature difference to be equal to the maximum possible temperature difference during the whole process this estimation is conservative. Inserting the values $\vartheta_{t,1} = 300 \text{ K}$, $p_{t,1} = 4 \text{ bar}$, $p_{t,2} = 3.5 \text{ bar}$, $t_b = 0.5 \text{ s}$, $A_t = 0.7 \text{ m}^2$, $V_t = 32 \text{ l}$ and $h = 10^3 \text{ W/m}^2 \text{ K}$ into equation (5.5) and (5.6), the result is $\Delta \vartheta = 0.4 \text{ K}$. Since the tank temperature is on the order of $300 \text{ K}$ and also these assumptions are conservative the error made when approximating the temperature after the load step with an adiabatic expansion is negligible.
5.5 Implications of Operating the Engine in Boost Mode

The test bench setup as described in section 5.1.1 allows to operate the engine in an exceptional way. Suppose the throttle is fully opened and the boost valve is fully opened as well with the engine running. The intake manifold pressure will be \( \approx 1 \) bar and the mass entering the intake manifold through the boost valve will leave the intake manifold partly through the throttle and partly through the engine. Suppose the throttle is closed gradually, this will lead to an increased intake manifold pressure. Then the fraction that leaves through the throttle will gradually decrease while the fraction entering the engine will gradually increase.

If the wastegate of the turbocharger is opened during these measurements (also it has to be open to prevent overspeeding) the exhaust backpressure remains at a low level, while high intake manifold pressures can be achieved. This way to operate the engine will be referred to as "boost mode". With this operation mode it is possible to compare the conventional engine operation with the engine operation as it will be during indirect boosting.

Steady state measurements were made using the boost mode where the intake manifold pressure was between 1 bar and 1.7 bar. Later on, measurements were taken under conventional operation with the same intake manifold pressures. These measurements were all conducted at 2000 rpm with the \( \lambda \)-controller activated to ensure that after the turbine the air-to-fuel ratio is always at \( \lambda = 1 \).

One effect that might occur when operating the engine in the boost mode is scavenging. Since the intake manifold pressure is higher than the exhaust gas backpressure and there is a valve overlap of \( \approx 48^\circ \text{CA} \) some of the air aspirate could directly escape the combustion chamber over the exhaust valves. If scavenging occurs the engine mass flow will be higher in the boost mode compared to the conventional operation when compared at equal intake manifold pressures. In conventional operation, the engine mass flow \( \dot{m}_g \) is equal to the mass flow measured by the sensor \( \dot{m}_{HF M} \). In the boost mode the engine mass flow cannot be directly measured but calculated since the boost valve has been identified. The engine mass flow in that case is \( \dot{m}_g = \dot{m}_{HF M} - \dot{m}_{HF M} \).

The results show the opposite of what was expected: a lower mass flow in the boost mode (see figure 5.7). So this means that instead of scavenging the volumetric efficiency is reduced in the boost mode. The reason for this can be seen in figure 5.6. When injecting air directly between the aspiration ports the pressure waves that propagate in the aspiration ports are altered. Since these pressure waves are very important for the volumetric efficiency (see [8] for details) it is clear that a change in the pressure waves can lead to a change in the volumetric efficiency. Also, since now air is injected continuously right to the intake valves the air motion at the intake valve may also be different than during conventional operation. For this engine and boost valve location, these changes lead to a slight decrease in volumetric efficiency.

Also another result can be obtained from these measurements when comparing measurements from boost mode and conventional operation at equal mass flows: the efficiency of the combustion increases in the boost mode. Figure 5.8 shows the high pressure part of the mean indicated pressure of the engine cycle \( p_{mi, hp} \). The indicated mean pressure of the gas exchange cycle \( p_{mi, ge} = p_m - p_{em} \).
has therefore been subtracted. In case of the boost mode the efficiency of the combustion process is higher since a higher $p_{m, hp}$ can be achieved with the same air and fuel mass flow respectively. When carrying out a process simulation, the $\Delta \Theta_{90}$-value that describes how long it takes until 90% of the fuel are burned is significantly lower in case of the boost mode. This $\Delta \Theta_{90}$-value is determined by fitting the parameters for the heat release rate using a Vibe combustion model. The combustion duration decreases from $\approx 58^\circ$CA for the conventional mode to $\approx 49^\circ$CA for the boost mode.

The reason for this could be an increased turbulence inside the cylinder during the combustion. An increased turbulence during the combustion leads to higher turbulent flame speeds and hence the combustion process only requires a shorter time. The reason for a higher turbulence during the combustion could be a higher turbulence at the intake valves during the time they are open is present. This might occur, since now air is injected continuously to the location upstream of the intake valves. There is a high level of turbulence in the injected air since the tank pressure is very high and sonic conditions are reached in the narrowest cross-section of the boost valve. However, these details can only be revealed with
5.5. Implications of Operating the Engine in Boost Mode

![Figure 5.8: indicated mean pressure of the high pressure process $p_{mi,hp}$ for the conventional operation and for the boost mode for various engine mass flows $\dot{m}_\beta$.](image)

It is assumed that the combustion efficiency increases similarly at other engine speeds. The Willans efficiency of the combustion while the boost valve is open experiences an increase of 2.3%. This must be implemented in the torque inversion model of the simulation model and at the test bench.
Chapter 6

Test Bench Results

As all necessary modifications were implemented at the test bench load steps were carried out in order to measure the transient performance on a real engine. Several load steps were carried out at different engine speed to prove the feasibility of the developed strategy and to validate the model over a wide range. The load steps are at 2000 rpm and at 2500 rpm from 1.7 bar $p_{me}$ to 13.4 bar and to 16.7 bar respectively. All the measurements were conducted using Strategy II, even though Strategy III yields better results. The reason for this is that the throttle at the test bench is rather slow which is unfavourable for this strategy.

6.1 Results for the Load Steps

Due to the highly nonlinear operating line of the boost valve at first load steps were carried out at a rather low tank pressure. In combination with a high overboosting factor it can be assured that the boost valve remains completely open throughout the whole boosting process and the load is controlled by the ignition angle controller alone. The Boost valve will open completely at the beginning of the load step and close again once the threshold turbocharger speed is reached. Like this the functioning of the ignition controller can be evaluated independently from the functioning of the boost valve controller. All load steps were carried out without using the $\lambda$-controller.

The OSV controller was not implemented on the testbench since it could be shown that with Strategy II it is not necessary. Only a very small mass flow or no mass at all is recirculated with the OSV in the simulations. So for all load steps shown the OSV was closed.

6.1.1 Load Step from 1.7 bar to 16.7 bar $p_{me}$ at $\omega_e = 2000 \text{ rpm}$

Figure 6.1 shows the result for a load step from 1.7 bar to 16.7 bar $p_{me}$ at $\omega_e = 2000 \text{ rpm}$. In the upper plot it can be seen the a rise time of $\approx 120 \text{ ms}$ is reached. After the desired $p_{me}$ is reached, the cycle averaged value of $p_{me, meas}$ follows the desired value well. The intake manifold pressure shown in the second plot increases to a value higher than the steady state pressure after the load step. An overboosting factor of $\nu_{ob} = 1.5$ is used for this load step. After the boost valve closes at $t = 343 \text{ ms}$ the intake pressure reduces. The throttle starts to
Figure 6.1: Comparison of measurement and simulation for a load step from 1.7 bar to 16.7 bar $p_{mc}$ at $\omega_e = 2000$ rpm. The measurement results are averaged over one engine revolution.
open at $t = 421$ ms. In the second plot also the signals of the boost valve and the throttle are shown. In case of the throttle the desired value is shown with a dotted grey line and the actual throttle position is shown with a dashed grey line. The feedback from the throttle position however reveals that the throttle opens only slowly. In the measurements the throttle opens when the intake manifold pressure is less than 500 mbar higher than the boost pressure. The fourth plot shows the tank pressure and the air mass consumption. For this load step 13.2 g of air are consumed. The bottom plot shows the measurement of $\lambda$ after the turbine. The deviation of the $\lambda$-value remains below 6%.

The fact that the load step is realized means that the control strategy presented in section 4 is suitable for indirect boosting.

For the comparison of measurements and simulations the signal from the boost valve controller from the measurements was used as an input for the model. In addition to that the time the throttle was opened also is taken from the measurement and used as an input for the simulation. Also the values for ambient conditions and the tank pressure and temperature before the load step were taken from the measurement and used in the simulation.

The simulation reproduces very well the behaviour of the real engine. All major signals are reproduced within 5% deviation. Only in the phase where the intake manifold pressure reduces after the boost valve is closed the deviation is higher.

The fact that the air mass consumption is higher than in the simulations made in section 4 (which is 12.45 g) has several reasons. One reason is the intake manifold volume. When the MVM was tuned to match the measurements it was found that an intake manifold volume of 2.61 l yields best results whereas in the model used in section 4 a value of $V_{im} = 2.41$ l is used. Increasing the pressure of a larger receiver requires more time so the duration of Phase I of the boosting process increases. Due to this longer time the boost valve must supply the engine mass flow for a longer time which requires more air. An other reason is that the throttle now does not close infinitely fast and also has a leakage. According to the simulations $\approx 0.2$ g of air enter the intake manifold until the intake manifold pressure reaches 1 bar and $\approx 1.2$ g of air escape from the intake manifold while the boost valve is open and the throttle is closed. As soon as the intake manifold pressure becomes bigger than the boost pressure air streams out of the intake manifold. The net air mass that the boost valve would not have to supply if the throttle had no leakage is $\approx 1$ g less. This matches well the simulation results obtained in section 4 for Strategy II.

The feed forward path of the fuel injection is very important when the load changes quickly. Within $\approx 180$ ms, which corresponds to only six ignition events, the intake manifold pressure increases by a factor of five. The modified predictor shows a good performance since the $\lambda$-value stays close to one.

### 6.1.2 Load Step from 1.7 bar to 16.7 bar $p_{me}$ at $\omega_e = 2500$ rpm

Figure 6.2 shows the result for a load step from 1.7 bar to 16.7 bar $p_{me}$ at $\omega_e = 2500$ rpm. Also at 2500 rpm the measurement shows good results. In this measurement the $p_{me}$ collapses at $\approx t = 480$ ms. The intake manifold pressure also collapses but at $\approx t = 450$ ms. The air mass consumption for this load step is 15 g. A low overboosting factor of $\nu_{ob} = 1.2$ is used here.
Figure 6.2: Comparison of measurement and simulation for a load step from 1.7 bar to 16.7 bar $p_{mc}$ at $\omega_e = 2500$ rpm. The measurement results are averaged over one engine revolution.
The reason that the $p_{me}$ slightly collapses is compressor surge, see figure 6.5. The boost pressure $p_b$ and the intake manifold pressure $p_{im}$ are equal since the throttle has opened sufficiently when the pressures collapse at $t \approx 450$ ms. However, the compressor mass flow is negative for a short period of time after the throttle has opened. It becomes positive again at $t = 425$ ms. Shortly after that when there is a higher compressor mass flow again the pressures $p_b$ and $p_{im}$ increase again. Surge does occur due to the low overboosting factor. This is also the reason that the air mass consumption is higher than in the measurement of the same load step at 2000 rpm even though it was shown in section 4 that with this strategy approximately the same air mass consumption must be expected.

For this load step the measurement and the simulation also show well agreement. The only time there is a greater deviation is directly after the boost valve closes as for the previous load step shown.

In appendix [B] two load steps from 1.7 bar to 13.4 bar $p_{me}$ at $\omega_e = 2000$ rpm and $\omega_e = 2500$ rpm, respectively are shown.

### 6.1.3 Load Step with the Boost Valve open/close Operation

Finally, also a result for the open/close operation of the boost valve is shown in figure 6.3. For this load step the pressure of the pressure tank is increased and the effective area of the boost valve is limited. Like this it can be assured that the boost valve in this load step does not stay completely open but opens and closes as described in appendix [A] to provide on average the desired effective area.

Due to the increased pressure the gradient of the intake manifold pressure at the beginning of the load step is very steep as can be seen in the second plot of figure 6.3. This faster increase of the intake manifold pressure is followed by a rapid increase of the $p_{me,meas}$. The cycle averaged value for $p_{me}$ oscillates strongly. In the second plot it can be seen that the intake manifold pressure also oscillates. The $\lambda$ trajectory is not as good as in the first load step, however it remains in a range of $\pm 12\%$. The air mass consumption for this load step is 13.3 g.

The very fast increase of the $p_{me}$ is one reason for the oscillations of the $p_{me}$ that can be seen in the upper plot. The oscillations of the elastic shaft are excited very much with this operation. The oscillation of the intake manifold pressure had to be expected as mentioned in appendix [A] the amplitude is approximately as high as shown in figure 6.5. This is the second reason for the strong oscillations of the $p_{me}$: the frequency of the PWM signal is 10 Hz, which is approximately equal to the eigenfrequency of the elastic shaft. The ignition controller cannot keep the indicated mean pressure $p_{mi}$ constant as can be seen in the upper plot of figure 6.3. Since the $p_{mi}$ oscillates at the same frequency as the elastic shaft the eigenmode of the shaft gets excited even more. This is the reason that the measured value of the $p_{me}$ oscillates so much. At $t = 500$ ms when the boosting process is over and the intake manifold pressure does not oscillate anymore the oscillations in the measured $p_{me}$ quickly reduce.
6.1. Results for the Load Steps

Figure 6.3: Comparison of measurement and simulation for a load step from 1.7 bar to 16.7 bar $p_{me}$ at $\omega_e = 2000$ rpm for the open/close operation of the boost valve. The simulation results are averaged over one engine revolution.
The fact that the $\lambda$-value can be kept in a 12\% range even for such strong oscillations of the intake manifold pressure shows that the modifications that were applied to the observer and predictor also work fairly well when the boost valve is used in the open/close operation. The model reproduces this behaviour very well, the oscillations of the intake manifold pressure and exhaust gas backpressure are simulated very accurately. Also the simulated value of the air mass consumption is very close to the measured air mass consumption.

This result shows that it is basically possible to realize indirect boosting with an open/close valve instead of a proportional valve. It would be advantageous if an open/close valve that is faster was used since that will lead to smaller oscillations of the intake manifold pressure. Like this the $\lambda$-trajectory could be improved.

### 6.2 Surge on the Test Bench

For the following discussion the compressor mass flow is necessary. It is not measured on the test bench. However, the air mass measurement with the HFM is just upstream of the compressor. It is assumed that the difference between the mass flow measured with the HFM and the compressor mass flow is only small also in transient engine operation. Consequently the mass flow measured with the HFM is used to generate the plots of the operating point in the compressor map.

For the load step shown in figure 6.1 the air mass sensor upstream of the compressor shows the collapse of the mass flow at the beginning of the load step, this means in the bottom left of the compressor operating map (figure 6.4). The mass flow reduces shortly after the throttle opens. After the throttle opens the compressor operating points are beyond the surgeline. However the pressure keeps increasing and the mass flow does not decrease to zero.

For the load step shown in figure 6.2 this is somewhat different. Here also at the beginning of the load step the mass flow collapses. After the throttle is opened in this load step the compressor runs into surge since the mass flow becomes negative and also the pressure decreases during that time.

The collapse of the mass flow right at the beginning of the load step happens because of the low compressor speed. Only a minimal pressure ratio is possible, so instead of an increasing boost pressure a decreasing compressor mass flow results. The surge tendency is strongly increased due to the throttle leakage. So in addition to the mass flow from the compressor a mass flow on the order of 1 g streams into the receiver downstream of the compressor. Following the approximations made in section 4.1.5 this 1 g of air is approximately 13\% of the total mass in that receiver. If the operating points of the compressor would cross the surgeline even if the throttle had no leakage can not be determined.

For the load step at 2000 rpm that is conducted with a high overboosting factor $\nu_{ob} = 1.5$, the compressor mass flow does not get negative.

In the measurement at 2500 rpm shown in figure 6.2 surge occurs. A low overboosting factor $\nu_{ob} = 1.2$ is used in this measurement to show that surge occurs when the overboosting factor is not chosen high enough. For this load step the
6.2. Surge on the Test Bench

The simulation model shows a backflow of air out of the intake manifold over the throttle when the throttle is opened. This also increases the surge tendency. However, since the simulation model does not match very well the measurement in the phase when the throttle opens, this result must be regarded with suspicion. The tendency this shows is correct: when the throttle must be opened at a rather high pressure difference between the intake manifold pressure and the boost pressure because the throttle is slow, backflow over the throttle might occur. This also increases the surge tendency. However, the only effect that this has on the load step is a minimal reduction of the $p_{me}$ for a timespan of less than 100 ms. The pressure downstream of the compressor does not oscillate.
Chapter 6. Test Bench Results

The question whether surge as it occurs when conducting indirect boosting is harmful to the compressor could not be answered with the measurements.

6.3 Reduced Rise Times

What was also implemented on the test bench is the possibility to retard the ignition in order to achieve slower $t_{90}$-times.

In figure 6.6 a result for this is shown for a load step from 1.7 bar to 13.4 bar $p_{me}$ at $\omega_e = 2000$ rpm. In the trace of the $p_{me}$, even when looking at the cycle averaged values, there is a strong oscillation with a frequency of about 10 Hz for the measurement with the decreased rise time.

The oscillation at 10 Hz correspond to the eigenfrequency of the elastic shaft. The oscillations of the shaft get excited very much with the slower $t_{90}$-time. The performance of the ignition controller can be better evaluated when consulting the mean indicated pressure $p_{mi}$. Figure 6.6 shows the $p_{mi}$ for the decreased rise time. Except for unavoidable cycle-to-cycle fluctuations it increases in good approximation linearly.

Figure 6.6: Increased rise times $t_{90}$ for a load step from 1.7 bar to 13.4 bar $p_{me}$ at $\omega_e = 2000$ rpm.

6.4 Exhaust Gas Temperatures

What could not be measured on the test bench is the exhaust gas temperature during the transients. The sensors used are too slow. The exhaust gas temperature is much higher than in steady state due to the delayed ignition angles. According to the approximation presented in equation 3.5 a reduction of the ignition efficiency of 30% leads to an increase in the exhaust gas temperature of 300 K.

It would be interesting to know how high the exhaust gas temperature gets, since a too high temperature can damage the turbine. However there are two factors that indicate that the turbine will not get damaged. First of all overboosting is only possible at low engine speeds. Only there it is possible for the exhaust
gas to reach temperatures that are significantly higher than in steady state. At high engine speeds the steady state full load exhaust gas temperature is higher than at low engine speeds. For the engine on the testbench this difference is about 150 K between 2000 rpm and 4000 rpm. Secondly, the time in which the exhaust gas temperature is higher is only a few tenths of a second. Since the temperature is only increased for such a short period of time and also because the temperature of the turbine at the beginning of a load step is lower than the full load steady state temperature, it is not expected that the turbine wheel gets damaged.

6.5 Summary of the Test Bench Results and Recommendations

On the testbench the throttle has to be opened already at a pressure difference of $p_{\text{im}} - p_{\text{b}} < 500 \text{ mbar}$. Since the throttle gives a feedback about the current throttle position it can be seen that the throttle needs about 60 ms to open 50%. Apparently a faster throttle would be better suited for indirect boosting, especially when Strategy III shall be implemented where the throttle quickly opens and then closes again at the beginning of a load step. Since no other throttle was tested only estimates can be given on how fast the throttle should be. A throttle with a time constant of $\approx 25 \text{ ms}$ is assumed to be fast enough (the throttle is assumed to behave like a first order system). This has to be taken into account, when designing an engine with indirect boosting. The time constant of the boost valve is very low for the valve used. However, the time constant does not need to be this much lower than the one of the throttle. Assuming the throttle opens at the same moment the boost valve closes, given similar time constants of throttle and boost valve, a smooth transition can be expected.

One object that was not considered so far is the wastegate. In modern turbocharged engines typically a "wastegate open" strategy is used in order to keep the exhaust gas backpressure low and to save fuel. In such a case the wastegate needs to be closed at the load step and it should be closed once the exhaust gas backpressure starts to increase in order to accelerate the turbocharger as quickly as possible. For the load step shown in 6.1 at 2000 rpm this time is $\approx 80 \text{ ms}$. At double the engine speed a time of $\approx 40 \text{ ms}$ can be expected. So with the use of indirect boosting the wastegate should be fast enough to fully close at the highest engine speed where boosting shall be conducted during the time until the exhaust backpressure starts to increase.

It could be shown that surge can be avoided when a sufficiently high overboosting factor is used. The operating points can move slightly beyond the surglene without causing a collapse of the compressor mass flow and the compressor pressure ratio.

Even with the occurrence of surge, i.e. the compressor mass flow and the compressor pressure ratio do collapse, a load step can be carried out almost uninfluenced by the occurrence of surge. There is only a short, minor reduction of the $p_{\text{me}}$. In the measurement shown where surge occurs the mass flow collapses just once and does not start to oscillate. The pressure upstream and downstream of
the compressor also do not oscillate during this time. Due to that the question arises what damage surge, when it appears in this way, can cause.
6.5. Summary of the Test Bench Results and Recommendations
Chapter 7

Comparison of Direct Boosting and Indirect Boosting

Before comparing direct boosting and indirect boosting, the major difference between the two systems has to be illustrated. With indirect boosting, the complete mass flow is supplied into the intake manifold. The throttle is closed and the torque buildup is completely decoupled from the boost pressure and the turbocharger speed during the boosting process. With direct boosting, an additional mass flow is supplied directly into the cylinder. This means that the torque buildup depends on the air entering the cylinder through the intake valves and also on the air entering the cylinder through the charge valves. The torque buildup here is still coupled to the turbocharger speed. This additional mass introduced by the charge valves allows a sudden increase of the torque beyond the naturally aspirated torque. Due to the fact that the exhaust gas mass flow is also higher like this, the turbocharger speed-up is faster. This leads to a steeper gradient of the effective mean pressure $\frac{dp_{me}}{dt}$ when compared to a conventional turbocharged engine.

Figure 7.1 illustrates the torque trajectories for a loadstep at 2000 rpm from 1.7 bar to 16.7 bar $p_{me}$. The trajectories are for the operation without boosting, for direct boosting and for indirect boosting using Strategy II is shown. For the sake of completeness the trajectory of Strategy IV is also included. Even though it could be shown that supplying an additional mass flow with indirect boosting is basically also possible (Strategy IV), this possibility has extremely limited performance at low engine speeds. Since especially at low engine speeds the torque build-up is still slow, this strategy is ignored here.

One other difference between direct boosting and indirect boosting is the design point. Indirect boosting, as described in section 2.4, is designed such that the complete engine mass flow at the highest torque and highest engine speed where boosting shall be conducted can be delivered. With direct boosting however, the design point is the lowest engine speed where boosting shall be conducted and the highest load. The charge valves are designed so that this load step can be realized within a certain time.
No boosting

Indirect boosting

Direct boosting

With this in mind, the comparison is always somewhat unfair for one of the two systems since they cannot be designed following the same criteria. However, each system is designed in a way that leads to satisfying results over the whole operating map, so that it seems appropriate to compare the two systems anyway.

Figure 7.2 shows the simulation results for the air mass consumption $m_b$ over the whole operating range for indirect boosting and direct boosting. In case of direct boosting the results are obtained with a model that includes actuator dynamics. For indirect boosting a model without actuator dynamics is used. So this means that the results for indirect boosting might get slightly worse with a model that includes actuator dynamics. Since this is a potential study the values for indirect boosting are the ones obtained with Strategy III. That means that in this comparison the best possible strategy for indirect boosting without actuator dynamics is compared to a realistic model for direct boosting with actuator dynamics.
The comparison of the air mass that is needed with direct boosting and indirect boosting show that indirect boosting needs \( \approx 2\ldots 3 \) times as much air. The reason for this is, as mentioned before, that direct boosting only supplies an additional air mass while indirect boosting must supply the complete engine mass flow. There are two reasons that this can become a problem. First of all the pressure tank should be as small as possible since it must fit in the vehicle and packaging problems should be avoided. This means that there is only a limited air mass that can be used and the lower the air mass consumption is, the more load steps can be carried out without filling the tank. Secondly, the filling of the tank requires energy. In case of direct boosting, the tank can be filled during braking phases with the engine operating in pneumatic pump mode. The amount of air that can be gathered during braking phases may be enough so that a lack of air in the pressure tank can be avoided. With indirect boosting however, an extra air compressor needs to be used to refill the tank. There are several possibilities how this compressor could be used. One possibility is to couple it mechanically with a clutch to the drivetrain and charge the air tank in the braking phases. Like this no additional energy would be needed. However it is not clear if the braking phases would be sufficient to fill the tank due to the higher air mass consumption. Another possibility is to drive this compressor electrically. Then the compressor could run continuously and it could be designed so there will never be a lack of air. In that case, the compressor would need energy from the battery and hence this would be disadvantageous regarding fuel consumption. This question is open and must be investigated in order to overlook all impacts of indirect boosting.

The rise times are fast, both with direct and indirect boosting as figure 7.3 shows. With indirect boosting they are even faster, especially at low engine speeds. However, this must not necessarily be an advantage. Considering oscillations in the drivetrain or mechanical loads of the components involved, the rise times of the indirect boosting might have to be reduced. One major advantage that indirect boosting can offer is that the torque build-up can be shaped freely and
### Table 7.1: Comparison of Direct and Indirect Boosting

<table>
<thead>
<tr>
<th>criterion</th>
<th>direct boosting</th>
<th>indirect boosting</th>
</tr>
</thead>
<tbody>
<tr>
<td>hardware modifications</td>
<td>• charge valve in cylinder head</td>
<td>• boost valve in intake manifold</td>
</tr>
<tr>
<td></td>
<td>• charge valve actuation (camshaft driven, electrohydraulic...)</td>
<td>• boost valve actuation (PWM)</td>
</tr>
<tr>
<td></td>
<td>• pressure tank</td>
<td>• pressure tank</td>
</tr>
<tr>
<td>throttle</td>
<td>no modifications</td>
<td>a fast throttle with low leakage is advantageous</td>
</tr>
<tr>
<td>additional operating modes of the engine</td>
<td>• boost mode</td>
<td>• boost mode</td>
</tr>
<tr>
<td></td>
<td>• pneumatic motor mode</td>
<td></td>
</tr>
<tr>
<td></td>
<td>• pneumatic pump mode</td>
<td></td>
</tr>
<tr>
<td></td>
<td>• pneumatic start</td>
<td></td>
</tr>
<tr>
<td>pressure tank filled</td>
<td>by the engine itself using the pneumatic pump mode</td>
<td>by an external compressor</td>
</tr>
<tr>
<td>rise time $t_{90}$</td>
<td>$[-]$ (reference)</td>
<td>faster</td>
</tr>
<tr>
<td>air mass consumption $m_b$</td>
<td>$[-]$ (reference)</td>
<td>higher</td>
</tr>
<tr>
<td>surge problems</td>
<td>no</td>
<td>can be avoided with overboosting</td>
</tr>
</tbody>
</table>

In a very simple way. So realizing slower $t_{90}$-times is hence not a problem. If direct boosting is realized as a fixed camshaft concept, i.e. that the charge valves can either be used or not used and cannot be adjusted continuously, there is an other advantage that indirect boosting can offer: it can compensate for different tank pressures. Since the boost valve mass flow depends on the effective flow area and on the tank pressure, a lower tank pressure can be compensated with a higher valve opening. Like this, the same $t_{90}$-time can be realized at different tank pressures. With a fixed camshaft direct boosting application, the valve lift profile cannot be changed and hence the mass that enters the cylinder depends on the tank pressure. With such a system, the $t_{90}$-time also depends on the tank pressure.

Besides the pros and cons for these two criteria, there are several other implications that must be consireded. Table 7.1 gives an overview of the two systems.
Chapter 8

Summary and Conclusion

In this thesis, a design process is presented on how to implement indirect boosting for SI internal combustion engines. It is shown how the boost valve must be dimensioned in order to be able to carry out load steps over a desired operating range of the engine. Control strategies are presented that lead to a low as possible consumption of pressurized air. Also with the strategies presented it could be shown that surge can be avoided in the simulations without the use of an overstream valve.

To demonstrate the realizability of indirect boosting and to show the transient performance of the strategies developed, one strategy is realized on the test bench.

It is shown how the cylinder air mass estimation must be modified and also it is shown what must be considered when designing an ignition controller to accurately reduce the engine load just as much as desired. Due to the high accuracy with which these modifications were implemented, the desired torque can be reached very accurately during load steps and also the deviations in the air-to-fuel ratio remain below 6% without λ-control. Due to the problems with the operating line of the boost valve it could eventually be shown that an open/close valve is sufficient to facilitate indirect boosting.

All goals of the thesis have been achieved. Most importantly, it could be shown that with indirect boosting, which only requires few changes to the engine, the turbo lag can effectively be overcome. This result shows that a good drivability can be achieved even for heavily downsized engines with indirect boosting.

8.1 Outlook

Several tasks are still open or could be analyzed in more detail. With use of Strategy II the criterion when the boost valve can be closed was not found analytically. This criterion might be derived or could be found by the use of model predictive control.

The fact that the efficiency of the combustion and the volumetric efficiency are different during the boosting process can have an influence on the knocking behaviour. This is due to the fact that the combustion process is faster, which has an impact on the optimal ignition timing. These correlations could be investigated in more detail.
All load steps have been carried out with a closed overstream valve. Since surge can be an issue on the testbench an overstream valve controller could be designed or other measures to avoid surge could be analyzed. Also experiments could be carried out using a throttle that has a lower leakage to determine if surge still occurs in that case. However since the load steps still showed good results even with surge, an other possibility would be to quantify how bad surge is for the compressor and the turbocharger. Like this, conclusions could be drawn on whether there is even a necessity to avoid surge.

At high engine speeds the operating point of the compressor during transients is not close to the surgeline anymore. This means that air injected into the intake manifold with an open throttle will not drive the compressor into surge. This might be a strategy of indirect boosting at higher engine speeds, outside of the range that indirect boosting was originally designed for. These issues have not been investigated in detail and offer options for future work.
Appendix A

Boost Valve Identification

At first, measurements of the dynamic properties of the boost valve were made. Therefore, the pressure tank was connected to the air supply, filled and then disconnected from the supply. At a completely opened throttle, the boost valve was completely opened and then closed again a short time later. This was done at several tank pressures. One measurement for opening and closing can be seen in figure A.1.

Figure A.1: Opening and closing of the boost valve. In gray the measurement with the HFM, in black the approximation as a first order element. The HFM was mounted in the middle of a long pipe for this measurement.

In this measurement, two things can be observed: Firstly, the boost valve opening and closing behaves like a first order element. The time constant was determined to be 15 ms. Secondly, we can see strong oscillations in the measured mass flow when the boost valve is closed. This could either mean that the boost valve does not close properly but rather bounces in its valve seat or that an eigenmode of pressure wave oscillations inside the pipe is excited.

To distinguish, which of the two possibilities it is, the same experiment was conducted with a much shorter pipe. If the valve closes properly and the oscillations come from pressure waves propagating in the pipe, with this shorter pipe, higher frequencies of the oscillations are expected. As can be seen in figure A.2, the frequency of the oscillation is much higher for the shorter pipe. From
this it can be concluded, that the valve does not bounce in its valve seat.

Figure A.2: Opening and closing of the boost valve. In gray the measurement with the HFM, in black the approximation as a 1st order element. The HFM was mounted directly downstream of the throttle for this measurement, there was no pipe attached downstream of the HFM.

From these measurements it could not be determined how big the delay of the boost valve is. This could however easily be identified later on consulting the intake manifold pressure trace at the beginning of a boosting event, see section 6, figure 6.1. The delay was increased until the pressure trace in the first moments of the simulation matches the one in the measurements. The boost valve was found to have a delay of 5 ms.

The result of the dynamic measurements is that the boost valve can be implemented as a first order element with a time constant of $\tau_{bv} = 15 \text{ ms}$. At different tank pressures the boost valve dynamics were equally fast. However, when steady state measurements are carried out to determine the operating line by simply changing the PWM signal, only disappointing results are obtained. Even though the valve is supposed to be a proportional valve, the operating line is not proportional over a wide range as can be seen for one set of measurements in figure A.3.

It is especially unfortunate that exactly the range needed for control is in the steep region of the operating line. Different PWM-frequencies (100 Hz, 200 Hz and 500 Hz were tried out) did not make a difference in that behaviour. The maximum effective area of the valve differed to a large extent from the specification in the data sheet. However this is not a problem. Since it is bigger than expected, the tank pressure can be reduced accordingly to account for that. Consequently, the equation for isentropic flow at subcritical pressure ratios must be included when calculating the mass flow from the effective valve area.

Several procedures were attempted in order to get a linear operating line, however none of them showed satisfying results.

Since the boost valve is very fast another attempt was chosen to achieve different steady state mass flows: Fast opening and closing of the boost valve, which corresponds to a very low PWM frequency, was found to deliver a strongly varying massflow whose average increases linearly with the PWM-signal. The best result was obtained with a PWM-frequency of 10 Hz. Higher PWM-frequencies lead to nonlinear operating lines. An other object that was found is that the
The operating line alters for different intake manifold pressures. So the operating line had to be extended to an operating map for different intake manifold pressures, see [A.4]. What was measured is the pulsating mass flow, the effective area is calculated with the mean value of this mass flow. The intake pressure also oscillates when the boost valve is operated like this, an example is shown in figure [A.5].

Figure A.3: Operating line of the boost valve when simply modifying the PWM signal.

Figure A.4: Operating map of the boost valve for quick opening and closing. $A_{eff}$ is calculated from the mean value of the measured pulsating mass flow.

Also it was found that a hysteresis is present, which can be seen in [A.6] for an
Figure A.5: Oscillations of the intake pressure at an average intake pressure of 2 bar and a PWM-signal of 65%.

intake manifold pressure of 2 bar. Since the control strategy of the boost valve requires it to open completely at the beginning of a load step and then to close again, the operating map for the "closing operation" was implemented.

Figure A.6: Hysteresis of the operating line at $p_{im} = 2$ bar.
Appendix B

Load Steps to 13.4 bar $p_{me}$

Two load steps are shown here to a $p_{me}$ of 13.4 bar for an engine speed of 2000 rpm and 2500 rpm, respectively.
Figure B.1: Comparison of measurement and simulation for a load step from 1.7 bar to 13.4 bar $p_{me}$ at $\omega_e = 2000$ rpm. The measurement results are averaged over one engine revolution.
Figure B.2: Comparison of measurement and simulation for a load step from 1.7 bar to 13.4 bar $p_{me}$ at $\omega_e = 2500$ rpm. The measurement results are averaged over one engine revolution.
Bibliography


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