Compressor Modeling for Control of Automotive Two Stage Turbochargers

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Abstract

There is a demand for increasing efficiency of automotive engines, and one way to achieve this is through downsizing and turbocharging. In the design compromises are made, for example the maximum power of the engine determines the size of the compressor, but since the compressor mass flow range is limited, this affects the torque for low engine speeds. A two stage system, with two different sized turbochargers, reduces this compromise, but the system complexity increases. To handle the complexity, models have come to play a central role where they aid engineers in the design. Models are used in simulation, for design optimization and also in the control synthesis. In all applications it is vital that the models have good descriptive capabilities for the entire operating range studied.

A novel control oriented compressor model is developed, with good performance in the operating regions relevant for compressors in a two stage system. In addition to the nominal operating regime, also surge, choke and operation at pressure ratios less than unity, are modeled. The model structure can be automatically parametrized using a compressor map, and is based on static functions for low computational cost. A sensitivity analysis, isolating the important characteristics that influence surge transients in an engine is performed, and the gains of a novel surge controller are quantified.

A compressor map is usually measured in a gas stand, that has different surrounding systems, compared to the application where the compressor is used. A method to automatically determine a turbo map, when the turbo is installed on an engine in an engine test stand is developed. The map can then be used to parametrize the developed compressor model, and effectively create a model parametrized for its intended application.

An experimental analysis of the applicability of the commonly used correction factors, used for estimating compressor performance when the inlet conditions deviate from nominal, is presented. Correction factors are vital, to e.g. estimate turbocharger performance for driving at high altitude or to analyze second stage compressor performance, where the variations in inlet conditions are large. The experimental campaign uses measurements from an engine test cell and from a gas stand, and shows a small, but clearly measurable trend, with decreasing compressor pressure ratio for decreasing compressor inlet pressure, for points with equal corrected shaft speed and corrected mass flow. A method is developed, enabling measurements to be analyzed with modified corrections. An adjusted shaft speed correction quantity is proposed, incorporating also the inlet pressure in the shaft speed correction. A high altitude example is used to quantify the influence of the modified correction.
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Contents

1 Introduction 3
  1.1 Contributions ............................................. 4
  1.2 Future work ............................................... 5

2 Turbo and experimental setup 7
  2.1 Main components of a turbo ......................... 7
    2.1.1 Compressor ........................................ 7
    2.1.2 Housing and bearings ............................. 9
    2.1.3 Turbine ........................................... 9
  2.2 Turbo maps ............................................. 9
    2.2.1 Compressor map ................................ 9
    2.2.2 Turbine map .................................... 11
  2.3 Experimental setup ................................. 13
    2.3.1 Engine, dynamometer and measurement systems ... 13
    2.3.2 Sensors ........................................... 14

3 Compressor modeling 17
  3.1 Model based control and mean value engine modeling ...... 17
  3.2 Different model families ............................ 18
    3.2.1 3D and 1D models ............................... 18
    3.2.2 Physical 0D compressor models ................. 18
    3.2.3 Curve fitting 0D based models ..... 20
  3.3 Choke flow and restriction modeling ..................... 21
  3.4 Surge and zero mass flow modeling .................... 21
# Introduction to combustion engine turbocharging

## 1 Time to surge concept and surge control for acceleration performance

1. **Introduction**
2. **Modeling**
   1. Surge region modeling
   2. Surge region validation
3. **Time to surge – TTS**
   1. System 1: Instantaneously zero throttle mass flow
   2. System 2: Dynamic throttle behavior
   3. System 3: Temperature dynamics in intermediate control volumes
   4. System 4: Complete 14 states MVEM
   5. Conclusions of the TTS-investigation
4. **Construction of a surge control system**
   1. Feedforward or feedback control
   2. Surge valve characteristic
   3. Formulation of control goal
   4. Control algorithm
5. **Controller evaluation**
6. **Conclusions**
7. **Nomenclature**
8. **References**

## Papers

### 2 Engine Test Bench Turbo Mapping

1. **Introduction**
2. **System description and turbo maps**
   1. System description
   2. Compressor and turbine maps
3. **Measurements**
   1. Gas stand measurements
   2. Engine test stand measurements
4. **Engine test bench imposed limits**
5. **Theoretical investigation of limits**
   1. Turbine inlet temperature
   2. Turbine mass flow
   3. Turbine mass flow measurement
   4. Compressor temperature increase
   5. Flexible pipes
   6. All Constraints Overlaid
   7. Extensions using pre-compressor throttle
6. **Turbo Mapping Method**
7. **Correction Factors for Measurements**
   1. Correction by Dimensionless Numbers
Introduction to combustion engine turbocharging
Combustion engines have for a long time been the most important prime mover for transportation globally. A combustion engine is simple in its nature; a mix of fuel and air is combusted, and work is produced in the operating cycle.

“Air, fuel, compression and a spark, and it should start.”

The amount of combusted air and fuel controls the amount of work the engine produces. The engine work has to overcome friction and pumping losses, and a smaller engine has smaller losses and is therefore more efficient. Increasing engine efficiency in this way is commonly referred to as downsizing. Downsizing has an important disadvantage; a smaller engine can not take in as much air and fuel as a larger one, and is therefore less powerful, which can lead to less customer acceptance. By increasing the charge density the smaller engine can be given the power of a larger engine, and regain customer acceptance. A number of charging systems can be used for automotive application, e.g. supercharging, pressure wave charging or turbocharging. Turbocharging has become the most commonly used charging system, since it is a reliable and robust system, that utilizes some of the energy in exhaust gas, otherwise lost to the surroundings. It is outside the scope of this thesis to give a comprehensive summary of basic engine operations and the interested reader is referred to [1, 2, 3].

There are however some drawbacks and limits of a turbo. The compressor of a single stage turbo system is sized after the maximum engine power, which is tightly coupled to the maximum mass flow. The mass flow range of a compressor is limited, which imposes limits on the pressure build up for small mass flows and thereby engine torque at low engine speed. Further, a turbo needs to spin with high rotational speed to increase air density, and due to the turbo inertia it takes time to spin up the turbo. This means that the torque response of

†Free translation from a seminar, held in Swedish by Per Gillbrand.
a turbocharged engine is slower than an equally powerful naturally aspirated engine, which also lead to less customer acceptance.

A two stage turbo system combines two different sized turbo units, where the smaller mass flow range of the smaller unit, means that pressure can be increased for smaller mass flows. Further, due to the smaller inertia of the smaller unit, it can be spun up faster and thereby speed up the torque response of the engine. The smaller unit can then be bypassed for larger mass flows, where instead the larger turbo unit is used to supply the charge density needed [4, 5]. A brief summary of the most important turbo characteristics, is found in Chapter 2, and the more interested reader is referred to [6, 7].

The use of two turbochargers adds actuators and complexity to the engine system. This illustrates that in the process of designing more efficient engines, they are made more flexible to reduce design trade-offs and enable optimization. As a side effect, this also increases the system complexity, and models are used as the foundation for concept development and implementation of control systems, that meet the increased demand for engine performance combined with increasingly stringent emission legislation. To be useful it is vital that the models have good descriptive capabilities over the relevant operating range. Development and validation of control oriented compressor models for two stage systems, is the scope of this thesis, and Chapter 3 presents related research in the compressor modeling area.

1.1 Contributions

Paper 1 [8] extends a mean value engine modeling framework, with surge description capability. A sensitivity analysis is performed, showing the important characteristics that influence surge properties in an engine. This knowledge is used in the design of a novel surge controller, that avoids surge and improves the vehicle acceleration performance.

Paper 2 [9] contributes with a method for determining turbocharger performance on engine test bench installations. An analysis of the limits that an engine installation imposes on the reachable points in the compressor map is performed, in particular it shows what these limits depend on. The novel use of a throttle before the compressor is proposed, enabling the engine system to span a larger region in corrected flow. An engine and test cell control structure, that can be used to automate and monitor the measurements by controlling the system to the desired operating points, is also proposed. Two methods that compensate for the deviation between measured and desired speed are proposed and investigated.

The contribution in Paper 3 [10] is the development of a compressor model, capable of representing mass flow and pressure characteristic for three different regions: surge, normal operation, as well as for when the compressor acts as a restriction. Both the parametrization and validation are supported by measured data. The proposed model is shown to have good agreement with measured data for all regions, without the need for extensive geometric information or data.

An analysis of the corrections used to scale compressor performance for varying inlet conditions is presented in Paper 4 [11]. A novel surge avoidance strategy is proposed, where the result is that a reduction in inlet pressure can
increase the surge margin. The method to investigate the applicability of the
strategy is straight forward and general. An experimental analysis of the correction
factors, commonly used to determine compressor performance when inlet
conditions deviate from nominal conditions, is then presented. Experimental
data from an engine test cell and a gas stand shows a small, but clearly measurable
trend, with decreasing compressor pressure ratio for decreasing compressor
inlet pressure. A method is developed, enabling measurements to be analyzed
with modified corrections. An adjusted shaft speed correction quantity is pro-
posed, incorporating also the inlet pressure in the shaft speed correction.

1.2 Future work

This section briefly presents continuations of this thesis, or research topics that
have been found along the way, but that did not gain the deserved attention
due to the time constraints.

A turbocharger consists of two main parts, a compressor and a turbine, and a
natural continuation of this thesis is therefore found in its title. To evaluate and
validate control oriented turbine models, can be motivated by the development
and increased use of twin scroll turbines, mixed flow turbines, variable geometry
or variable nozzle turbines.

To investigate and evaluate observer designs based on the developed compres-
 sor models is also an interesting continuation. To accurately know the two
stage system states, e.g. pressures, temperatures and turbo speeds, is impor-
tant for a controller. Especially to estimate the shaft speed of the bypassed high
pressure stage is an interesting scope. This is important for transients involving
a stage switch, where the charging effort is transferred from one of the stages
to the other. Further investigation of shaft friction can be motivated by experi-
mental experience from the engine test stand, where the bypassed second stage
sometimes stops in friction. For transient two stage control, it is important for
the controller to know the shaft speed of the turbos; if the high pressure turbo
has stopped or is rotating along in 30,000 rpm is then vital information.

Diagnosis and supervision of the actuator system of the two stage system is
also interesting, e.g. to supervise the high pressure stage bypass valve. Such a
path would include evaluation and actuator modeling, which is also beneficial for
control purposes. To use a model based feed-forward term to remove a difficult
non linearity of a system, has been successfully used in many applications. Also
diagnostic and modeling of automotive sensors is an interesting path. To be able
to measure accurately, and to trust the measurement reading is the foundation
for any feedback control.

More measurements of turbo performance for larger variations in inlet condi-
tions, and for more units, compared to the reference conditions, is also a research
scope. Such an investigation is also closely coupled to interesting investigations
of how engine system geometries and properties affect both compressor and
turbine performance, compared to the gas stand measured maps.
Chapter 1. Introduction
Turbo and experimental setup

This chapter starts with a description of the main components of a turbo, followed by a presentation of turbo maps and the correction equations, used to adjust turbo performance to inlet conditions that deviate from the conditions used when the map was measured. The last section presents the experimental setup used in Paper 2 and in Paper 4.

2.1 Main components of a turbo

An automotive turbo consists of a compressor and a turbine on a common shaft, supported by bearings in a center housing. An example of a turbo is shown in Figure 2.1, where some of the important parts are marked. The turbo extracts some of the energy in the exhaust gas, and transfers this as power to the compressor. The compressor increases the intake air density, and thereby the engine power.

2.1.1 Compressor

Almost all automotive compressors are centrifugal compressors. They are referred to as centrifugal, or radial compressors since the air enters axially, but leaves the compressor impeller in the radial direction. Axial compressors also exist, but are used mainly in e.g. aircraft or power generation applications.

The air is collected in the compressor inlet, and lead to the first part of the impeller, referred to as the inducer section. The impeller consists of many impeller vanes, that rotate at a high velocity and transfers energy to the air. The maximum shaft velocity of an automotive turbo is approximately 200,000 rpm. The transferred energy increases the air velocity and thereby the kinetic energy. The air then leaves the compressor impeller in the radial and tangential
Chapter 2. Turbo and experimental setup

Figure 2.1: Picture of a turbo. The air enters the compressor through the inlet (left) and through the impeller. The heated and pressurized air is then collected in the volute and delivered to the outlet connection (up). The exhaust gas enters the turbine through the turbine inlet (hidden behind the turbo), is lead through the volute into the impeller, where energy is transferred to the impeller vanes and then exits through the outlet (right).

The air enters the compressor through the inlet (left) and through the impeller. The heated and pressurized air is then collected in the volute and delivered to the outlet connection (up). The exhaust gas enters the turbine through the turbine inlet (hidden behind the turbo), is lead through the volute into the impeller, where energy is transferred to the impeller vanes and then exits through the outlet (right).

The kinetic energy is partly converted into potential energy in the impeller passages, and partly in the diffuser section when the air is decelerated. Both air pressure and temperature increase in the compression process. The temperature increase is inevitable but undesired, since an increase in temperature decreases the air density. The air with increased pressure and temperature leaves the diffuser section and is collected in the volute, that leads the air to the compressor outlet connection.

A valve can be mounted in close connection to the compressor. This valve is referred to as a surge or recirculation valve, and is used to decrease the pressure after the compressor. The valve opens a connection from the pipes after the compressor to the pipes before, to avoid compressor surge, that is described in Section 2.2.1. Control of the surge valve is connected to engine performance, and is studied in Paper 1. How a compressor can be modeled is presented in Chapter 3.
2.1.2 Housing and bearings

The bearings of the turbo shaft are mounted in the center housing of the turbo, and are critical components, needed to support the shaft at high speeds. Fluid bearings and ball bearings are commonly used, and are supplied with oil to reduce friction. The oil supplied, or additional water cooling, is used to cool the turbo. Turbo bearing friction is analyzed and measured e.g in [12], and modeled e.g in [13]. The effect of the heat transfer in an automotive turbo system is analyzed e.g. in [14, 15, 16, 17].

2.1.3 Turbine

The turbine recycles some of the otherwise lost energy from the hot exhaust gas of the combustion engine. The exhaust gas flows through a turbine impeller, where energy is extracted and transferred to the compressor side. A waste gate valve is commonly used to control the amount of exhaust gas that flows through the turbine, and thereby the turbine power. Some turbines use a variable nozzle, or a variable geometry, to control the turbine power, instead of a waste gate. These actuators effectively change the flow geometry within the turbine.

Turbine performance measurements and analysis is presented e.g. in [18] and models of the turbine can be found e.g. in [13, 19].

2.2 Turbo maps

Turbo performance is usually presented in maps using corrected performance variables. The corrections are important, since the performance maps are otherwise only valid for the conditions under which they were measured. The basis for the corrections is dimensional analysis [20], and the correction equations relevant for turbochargers are presented e.g. in [7, 21, 22]. The correction equations scale the turbine and compressor performance variables, based on the current inlet temperature and pressure. An experimental investigation of the correction quantities for the compressor is presented in Paper 4.

There are standards describing the procedures involved in measuring a turbo map, see e.g. [23, 24, 25, 26]. The definition of when surge occurs, which gives the smallest mass flow point for a corrected compressor speed, have been discussed in recent works [27, 28]. A summary of some different turbocharger test facilities is presented in [27]. Methodology to measure turbo performance on an engine in a test stand is presented in Paper 2.

2.2.1 Compressor map

There are four performance variables for the compressor map: corrected mass flow, pressure ratio, corrected shaft speed and adiabatic efficiency. The corrected compressor mass flow is given by

\[ \dot{m}_{c,\text{corr}} = \dot{m}_c \sqrt{\frac{p_{c,\text{std}}}{p_{e,\text{std}}}} \text{ [kg/s]} \] (2.1)
Figure 2.2: Example of a compressor map. The numbers in boxes indicate corrected shaft speeds in krpm, i.e. 180 means 180000 rpm. Circles indicate measured points, and the contours represent the adiabatic efficiency. The surge line is also marked.

where $\dot{m}_c$ [kg/s] is the compressor mass flow, $T_{01}$ [K] is the compressor inlet temperature, and $p_{01}$ [Pa] is the compressor inlet pressure. The temperature $T_{c, std}$ [K] and the pressure $p_{c, std}$ [Pa] are the reference states. The reference states must be supplied with the compressor map, since these states are used to correct the performance variables. The compressor pressure ratio is given by

$$\Pi_c = \frac{p_{02}}{p_{01}} [-] \quad (2.2)$$

where $p_{02}$ [Pa] is the compressor outlet pressure. The corrected shaft speed is defined as

$$N_{c, corr} = N_{tc} \frac{1}{\sqrt{\frac{T_{01}}{T_{c, std}}}} \quad [\text{rpm}] \quad (2.3)$$

where $N_{tc}$ is the turbo shaft speed. The adiabatic efficiency of the compressor

$$\eta_c = \frac{\left( \frac{p_{02}}{p_{01}} \right)^{\frac{\gamma_c - 1}{\gamma_c}} - 1}{\frac{p_{02}}{p_{01}} - 1} [-] \quad (2.4)$$

where $\gamma_c [-]$ is the ratio of specific heats for air. The adiabatic efficiency describes how efficient the compression of the gas is, compared to an ideal adiabatic process. Or in other words, how much the pressure increases, compared to how much the temperature increases.

Points measured with equal $N_{c, corr}$ are connected in the compressor map, and are referred to as speed lines. A speed line consists of a number of measure-
ments of $\Pi_c$ and $\dot{m}_{c,corr}$, and gives the characteristics of the compressor. Compressor efficiency $\eta_c$ is also measured for each point, and contours of constant $\eta_c$ are normally superimposed over the speed lines. The mass flows measured on each speed line range from the surge line into the choke region. An example of a compressor map is shown in Figure 2.2.

The surge line is the boundary of stable operation of the compressor. A compressor will enter surge if the mass flow is reduced below this point. Surge is an unstable condition, where the mass flow oscillates. These oscillations can destroy the turbo. Compressor choke is found for high mass flows, and indicates that the speed of sound is reached in some part of the compressor. Measurements are conducted at different $N_{c,corr}$ up to the maximum allowable, and mechanical failure of the turbo can result if the speed is increased further.

### 2.2.2 Turbine map

As for the compressor map, there are four performance variables used in the turbine performance map: corrected mass flow, expansion ratio, corrected speed and adiabatic efficiency. It is further common to define two more variables for the turbine: turbine flow parameter and turbine speed parameter. The corrected turbine mass flow is given by

$$\dot{m}_{t,corr} = \dot{m}_t \sqrt{\frac{T_{03}}{T_{t, std}} \frac{p_{03}}{p_{t, std}}} \ [kg/s] \quad (2.5)$$

where $T_{t, std} [K]$ and $p_{t, std} [Pa]$ can be other standard states, than are used in the compressor map. The turbine mass flow $\dot{m}_t [kg/s]$, is the combustion products and thus normally the sum of fuel and air. The pressures $p_{03} [Pa]$ and $p_{04} [Pa]$ are the turbine inlet and outlet pressure, respectively, and $T_{03} [K]$ and $T_{04} [K]$ are the turbine inlet and outlet temperature, respectively. It is common to neglect the standard states in (2.5), and present turbine data using the turbine flow parameter, or TFP

$$TFP = \dot{m}_t \sqrt{\frac{T_{03}}{p_{03}}} \ [kg \sqrt{K/ s kPa}] \quad (2.6)$$

where $p_{03}$ is usually in [kPa], as indicated by the unit of (2.6). The turbine expansion ratio is given by

$$\Pi_t = \frac{p_{03}}{p_{04}} \ [-] \quad (2.7)$$

Some authors prefer to have the pressure after the component divided by the pressure before, as is the case for the compressor pressure ratio (2.2). The corrected turbine shaft speed is given by

$$N_{t,corr} = N_{te} \frac{1}{\sqrt{T_{03} / T_{t, std}}} \ [rpm] \quad (2.8)$$

It is common to neglect $T_{std}$ in (2.8) and define the turbine speed parameter, or TSP as

$$TSP = N_{te} \frac{1}{\sqrt{T_{03}}} \ [rpm/K^{0.5}] \quad (2.9)$$
Chapter 2. Turbo and experimental setup

Since \( p_{\text{std}} \) and \( T_{\text{std}} \) are constants, neglecting them in equations (2.5) and (2.8) to give equations (2.6) and (2.9) respectively, gives only a scaling. The adiabatic efficiency of the turbine is given by

\[
\eta_t = \frac{1 - \frac{T_{04}}{T_{03}}}{1 - \left(\frac{p_{04}}{p_{03}}\right)^{\frac{\gamma_t}{\gamma_t - 1}}} \quad [-] \quad (2.10)
\]

where \( \gamma_t \) is the ratio of specific heats for the exhaust gas.

The high temperatures on the turbine side cause large heat fluxes. Measurement of \( T_{04} \) can have substantial systematic errors, due to the heat fluxes. An alternative efficiency definition for the turbine side is therefore commonly used, where no measurement of \( T_{04} \) is needed. The heat transfer effects are less pronounced on the compressor side, and the compressor power

\[
P_c = \dot{m}_c \cdot c_p (T_{02} - T_{01}) \quad (2.11)
\]

can be used to define an alternative efficiency. This alternative turbine efficiency definition includes the shaft friction, and the equation is

\[
\tilde{\eta}_t = \eta_t \cdot \eta_m = \frac{\dot{m}_c c_p (T_{02} - T_{01})}{\dot{m}_t c_{p,t} T_{03} \left(1 - \left(\frac{p_{04}}{p_{03}}\right)^{\frac{\gamma_t - 1}{\gamma_t}}\right)} \quad (2.12)
\]

where the shaft friction is included in the mechanical efficiency \( \eta_m \). Figure 2.3 shows an example of a turbine map.
2.3 Experimental setup

This section describes the experimental setup used for the measurements presented in Paper 2 and in Paper 4. The experimental setup varied slightly between the measurements, but most parts were kept intact. The experimental setup used in Paper 1 and in Paper 3 is described in [13].

2.3.1 Engine, dynamometer and measurement systems

The base line engine is a GM LNF, and is a four cylinder, SI DI 2.0 liter gasoline engine. It has a rated power of approximately $190 \text{ kW (260 hp)}$, and a rated torque of $350 \text{ Nm}$. The original single stage turbocharger is exchanged for a two stage system, see Figure 2.4. The maximum power of the engine, using the two stage system, is reduced. This since the largest compressor of the two stage system is smaller than the single stage system compressor, as discussed in Chapter 1. The engine control system is based on a dSPACE MicroAutoBox and RapidPro architecture. The control system is built around a Simulink model. The model is compiled using Real Time Workshop, and executed in real time on the MicroAutoBox.

The dynamometer is a Schenck Dynas3 LI 250. The rated speed of the dynamometer is $10000 \text{ rpm}$, the rated power $250 \text{ kW}$ and the rated torque $480 \text{ Nm}$. The dynamometer also acts as the start motor for the engine. The electricity the dynamometer generates, is fed back to the electric grid, and the heat expelled by the engine to the coolant system, supports the heating of the university buildings.
Chapter 2. Turbo and experimental setup

Figure 2.5: Schematic picture of the experimental setup. The electrical heating section is removed when not required. The extra throttle is used to decrease compressor inlet pressure. The air filters are used to straighten the air flow for the mass flow and pressure measurement locations. Air comes in from the left, runs through the compressor inlet variation rig, goes through the compressor stages, the engine, the turbine stages, catalyst, muffler and are expelled to the right.

The test cell measurement system consists of a HP VXi system, with a HPE1415A module and a HPE1433A module. The HPE1415A module is used to measure analog and digital signals, with a sample frequency of up to 2000 Hz, and has built in support for thermocouples. The HPE1433A module is a fast 8 channel converter with separate A/D-converters for each channel. It is used with sampling frequencies of up to 192 kHz, and can also be used to sample in the crank angle domain.

The signals are measured by the HP VXi system, where the most important signals are measured with both the HPE1415A and the HPE1433A modules. A sampling frequency of 1000 Hz is used for the HPE1415A and 128 kHz is used on the HPE1433A.

2.3.2 Sensors

Temperature

All temperature sensors are K-type thermocouples from Pentronic. A sensor width of 1.5 mm is used for $T_{02}$, and 3 mm sensor widths are used for $T_{01,1}$, $T_{01,2}$, $T_{01,3}$, $T_{03}$ and $T_{04}$. The temperature sensor $T_{LFE3}$ is used by the LFE3 mass flow sensor, and is also 3 mm in width.

The recovery factor, used to calculate the total temperature from a measured temperature [25], is assumed to be 1. This means that the measured temper-
2.3. Experimental setup

Figure 2.6: Photo from the engine test cell, showing the compressor inlet condition variation rig, the Schenk dynamometer and the engine test stand with the two stage system mounted to the LNF engine. The high pressure turbo (HP) and the low pressure turbo (LP) are marked.

Temperature is assumed to be the total temperature. Ice water and boiling water are used to calibrate the temperature sensors, before the measurement series.

The sensor width introduces dynamic to the measured temperature. The sensing element is mounted within a sensor body, and it takes time to heat the body. This gives a low pass filter effect on the measured temperature [29, 30]. However, this is not a problem for the measurements of Paper 2 and Paper 4 since they are stationary.

Pressure

The pressure sensors are either the 4260-series or the 4295-series from Kistler, except $p_{LFE3}$ which is measured internally by the LFE3 mass flow sensor system. The pressure sensors for the compressor inlet pressure measurement, $p_{01,1}$, $p_{01,2}$ and $p_{01,3}$ in Figure 2.5, are placed on a straight pipe, following an air filter to reduce flow disturbances. Four pressure taps are connected together, and the sensors $p_{01,1}$ and $p_{01,2}$ are connected to the four taps, while $p_{01,3}$ uses a single pressure tap. The positioning of the pressure taps for $p_{02}$, $p_{03}$ and $p_{04}$ are restricted due to the packaging, but placed at the most straight sections of the respective pipes. The exhaust side pressure sensors, $p_{03}$ and $p_{04}$, are mounted on pipes approximately 0.50 m in length, due to restricted temperature limits of the sensors.

The total pressure is calculated from each pressure measurement using the
measured mass flow, measured temperature and the pipe area using the equation

$$p_0 = p + \frac{1}{2} \rho v^2 = p + \frac{\dot{m}^2}{2A^2 \rho}$$

where $p_0$ is the total pressure, $p$ is the measured static pressure, $v$ is the flow velocity, $\rho$ is gas density, $\dot{m}$ is measured mass flow and $A$ is the cross sectional area at the measurement location. The difference between $p_0$ and $p$ is the dynamic pressure, describing the increase in pressure that the gas experiences when it is brought to stand still.

All pressure sensors are measured at engine off conditions, both before and after each measurement sequence, to indicate any sensor drift during the measurements. A reference sensor is also used to calibrate the pressure sensors for the measurements. Using long connection pipes between the measurement location and the sensor can cause a low pass filtering effect [30]. This is not a problem here, since the measurements are stationary.

**Mass flow**

Three different mass flows are measured. $\dot{m}_1$ in Figure 2.5 is measured using the LFE3 system. The LFE3 sensor system is purpose built for automotive research by the Technical University of Denmark (DTU), and uses the differential pressure principle. The differential pressure is measured over a laminar flow element, that gives the volumetric flow. $T_{LFE3}$ and $p_{LFE3}$ are then used to calculated the density, and the mass flow is determined according to

$$\dot{m}_1 = \dot{V} \cdot \rho_{LFE3}$$

where $\dot{V}$ is the volumetric flow and $\rho_{LFE3}$ is the density at the measurement location. The LFE3 differential pressure sensor is a 164PC0137 from Micro Switch Honeywell. The second mass flow sensor, $\dot{m}_2$, is a production sensor, based on the hot-wire principle, and produces a digital signal. An extra differential pressure sensor, is mounted in parallel with the LFE3 differential pressure, for diagnosis purposes. An extra mass flow, $\dot{m}_3$ is calculated also for extra differential pressure sensor, using $T_{LFE3}$ and $p_{LFE3}$ to calculate the air density. The extra differential pressure sensor is a Kistler 4264AB03-sensor.

Both mass flow measurement locations are located on straight pipe sections following air filters, to straighten the air flow.

**Turbo speed**

The turbo speed $N_{tc}$ of Figure 2.5 is measured using an ACAM PicoTurn BM-V6 system. The BM-V6 system is capable of both analog and digital output signal, where the latter is used to reduce noise sensitivity. The speed sensor position is adjusted using the built in sensor positioning functions of the BM-V6, to ensure the quality of the turbo speed measurement.
Compressor modeling

This chapter gives a summary of some of the research that is relevant for automotive compressor modeling, and related to the work in this thesis. A description of the operation and modeling of turbomachinery in general is found in the literature [1, 7, 21]. The chapter begins with a discussion of model based control and mean value engine modeling, while the rest of the chapter is devoted to compressor modeling. The compressor modeling presentation is divided into three parts: modeling of nominal operation, choke or a restriction like operation, and surge, see Figure 3.1 for a sketch of the different regions.

3.1 Model based control and mean value engine modeling

The use of mathematical models in an automotive control system is gaining increased interest from the industry. This increased interest comes from the complex engine concepts used, where additional actuators and degrees of freedom are added to the systems. Model based control is proposed as a way of handling the increased complexity. The models are used for a number of things. Simulation environments can be constructed around the models to aid for example in controller design, in concept evaluations, or in the parametrization process of other controller structures [8, 31, 32]. Observers can be built around the models to estimate non measured states of the system [13]. A direct use of an inverse model can be made, to handle a nonlinearity of a system [33, 34].

The model based control approach have been studied for different automotive control applications, for example in [14, 35, 36, 37].

Mean Value Engine Modeling [38, 39, 40, 41, 42] (MVEM) is a modeling framework used in the automotive society. Here it is used for the full engine simulations in Paper 1, Paper 3 and Paper 4. MVEM usually means that the
Figure 3.1: Schematic picture of the different compressor operating regions discussed in the text. The nominal region can be approximated with the efficiency contours shown. A decrease in mass flow to the left of the surge line puts the compressor in surge. The compressor is a restriction for $\Pi_c < 1$.

model is based on average values of the engine cycle, i.e. in-cylinder processes such as valve opening and closing are averaged out. This simplification means that vehicle test cycles, consisting of many minutes of driving, can be simulated on a normal PC, with short calculation times.

### 3.2 Different model families

Modeling of nominal compressor operation is divided into three subsections, depending on the model structure.

#### 3.2.1 3D and 1D models

Gas motion can be modeled in 3D, e.g. solving the Navier-Stokes equations of gas motion numerically. Such modeling needs accurate geometric information of the system, see e.g. the complex impeller geometries of Figure 3.2 and Figure 3.3. The boundary conditions of the model are further important, i.e. how the gas enters and leaves the modeled component. Due to the complexity and the computational effort, these models are most often only used to model components of the engine [43, 44, 45]. The solutions obtained, give valuable information of for example the gas motion, that can be used also for less complex model families. Also the reverse is true [46]; good models from less complex model structures can be used on a component level for a 3D simulation.

Another level of detail that is frequently used, is the 1D model family. They model the gas flow along pipes and account for properties in this dimension. 1D models of compressors are however rarely found. The computational cost is reduced, compared to 3D models, and large parts of an engine system can be simulated with reasonably short simulation times.

#### 3.2.2 Physical 0D compressor models

For the physical compressor model, an ideal compression process is frequently assumed, and different losses are then described and subtracted from the ideal
3.2. Different model families

Figure 3.2: Picture of the low pressure stage impeller (left) and high pressure stage impeller (right) used for the measurements presented in Paper 2 and Paper 4. Both impellers rotate clockwise.

component performance. This model structure often makes use of the velocity triangles, exemplified for the impeller entry in Figure 3.3.

This section follows a gas element through the compressor, and describes important losses along the way, that are compiled to the model. The air flow into the compressor is assumed to have no circumferential velocity, i.e. no pre-whirl, and the diffuser section is assumed to be vane-less. An automotive compressor is normally vane-less and without intentional pre-whirl, due to the fact that a vaned diffuser normally has a narrower flow range, and a pre-whirl system is avoided due to additional cost and packaging constraints.

The first losses occur since the gas has to comply with the vane geometry at the impeller inlet. These losses are referred to as incidence losses [47], and are due to that the inducer relative velocity vector W does not agree with the vector parallel to the vane surface, V, see Figure 3.3. The impeller vane angle varies with the radius of the impeller, since the outer points on the impeller have higher relative velocities [48]. Studying Figure 3.3, the incidence losses are minimized if \[ I = 0 \], however [47] states that the actual velocity vectors are not given simply by the geometries of the compressor, due to inertial effects of the gas.

The fluid friction losses due to the gas viscosity and motion through the compressor are modeled e.g. in [22, 35, 49], where slip is used to model the gas flow through the impeller. Slip describes how well the gas is guided by the impeller vanes, and is discussed and modeled in [7, 21, 50]. Generally, the less guidance the gas attracts from the vanes the more slip. The more guidance, the more friction.

Due to the potentially large pressure gradient through the compressor, flow can recirculate unintentionally. These flow recirculation losses occur due to the clearance between the impeller, rotating at high velocity, and the compressor housing. Flow recirculates both from the pressure side of the impeller vanes to the suction side, and along the compressor housing, from after the impeller,
Compressor impeller vane

Figure 3.3: Picture of the impeller used for the measurements presented in Paper 1 and the inducer velocity triangle. The Z-axis goes through the turbo shaft, W is the relative flow vector between the gas velocity vector C and the vane velocity vector U. The incidence loss in the inducer is connected to the velocity Ι. δ denotes the vane angle.

to the impeller entry. Models of these losses are presented e.g in [47, 51]. The air that recirculates to the impeller entry, is already heated by the compression process. In [52] it is stated, that the temperature of the recirculated air increases with increased compressor pressure ratio, and that the amount of recirculated air is a function of pressure ratio and not turbo speed. In [45], this recirculation is said to occur, where the local gas pressure is high and velocity is low. Experimental data of recirculation is also presented in the literature, see e.g [28]. The radial temperature profile at the impeller entry, discussed in [52], is experimentally shown in [28].

For compressors with vaned diffusers, an incidence loss can be associated with the air leaving the impeller, and entering the diffuser passages. These losses are however not simply given by the geometries of the impeller and diffuser vanes, but also of the flow physics inside the impeller [48]. Experimental investigations of the gas motion in the diffuser is found e.g in [53]. The main cause of the diffusion process losses, are in [7] said to be separation of boundary layers and fluid friction.

Losses in the volute [1], and losses due to disc friction [54] and choking [46] can also be modeled. The losses associated with the volute are more pronounced for a vaned diffuser, and are modeled in [47]. In [51] it is noted that the relative magnitude of the clearance, backflow and volute losses decreases with increasing mass flow, since the losses associated with incidence and friction increase more.

### 3.2.3 Curve fitting 0D based models

The curve fitting based approach is another subset of the 0D model family, and recognizes that all performance variables are conveniently given by the speed lines and the efficiency contours of the map. The modeling effort is then to fit different curves to the map, or to a transformed map.
3.3 Choke flow and restriction modeling

Semi-physical modeling usually transforms the compressor map variables into the dimensionless head parameter $\Psi$ and the dimensionless mass flow coefficient $\Phi$. A connection between $\Psi$ and $\Phi$ is then parametrized and used as a model [39, 55, 56].

Curve fitting directly to the map variables is another way to produce a model. The modeling effort is then to create functions describing the speed lines and iso-contours of efficiency of the map. A summary of curve fitting models for automotive control applications is presented in [57], and both speed line shapes [58] and efficiency contours [13, 59, 60] are modeled.

Both the models of Paper 1 and Paper 3 use a parametrized ellipse to represent the speed lines of the map, and are therefore of the curve fitting family. Curve fitting is also used to represent the efficiency contours of the map in both papers.

3.3 Choke flow and restriction modeling

The previous section described nominal compressor operation and this section presents research relevant for modeling choked flow, and when the compressor only restricts the air, i.e. compressor operation with a pressure ratio lower than unity.

If the inlet section of the compressor chokes, the choking is independent of compressor speed. This since the flow is choked, before it reaches the impeller blades. A varying choke mass flow with shaft speed, can be expected if choking conditions are established further into the compressor. This since the density of the gas arriving at the choking section, can be increased through an increase in compressor speed. In [49] the choke mass flow, assuming that choking occurs in the impeller, is described as an increasing function in shaft speed. A model that extrapolates compressor performance maps to smaller pressure ratios, including choking effects, is described in [46]. For pressure ratios lower than unity the compressor is assumed to work as an restriction for the flow, and the behavior of the compressor is compared to a nozzle discharge characteristic in [61], where further a constant efficiency of 20% is assumed in this operating region.

Paper 3 uses a choke mass flow model that is affine in corrected shaft speed. This is physically motivated by that a compressor impeller that stands still has a non-zero choke flow, and an increase in choke flow can be expected for increases in corrected shaft speed, up to the point where the compressor inlet chokes. The speed line model of Paper 1 focuses on the nominal compressor map, but the speed lines are extended to also cover pressure ratios less than unity. Constant compressor efficiency is assumed in this region in both Paper 1 and Paper 3.

3.4 Surge and zero mass flow modeling

The last region of the compressor map is the one left of the surge line in Figure 3.1. When the compressor operating point moves beyond this line, surge will occur since the compressor is unable to maintain flow. When the flow breaks down completely, the highly pressurized air travels upstream, reversing the mass
flow. This reversed flow reduces the pressure ratio, until the compressor is able to maintain positive mass flow. The pressure ratio then increases again and, if no other changes are applied to the system, the compressor enters a new surge cycle. Surge in automotive applications can be encountered for example during a gear change in an acceleration phase. When the accelerator pedal is released, a sharp reduction in throttle mass flow results. Due to the inertia of the turbo, the compressor wheel does not slow down fast, and the compressor continues to build pressure.

To model surge many authors follow the Moore-Greitzer approach [62]. An extra state is introduced in the model to handle changes in mass flow through the compressor, where, due to the gas inertia, the compressor mass flow deviates from stationary performance curves for a transient [63].

Surge can be established from low to high turbo speeds [64]. The frequency of the surge phenomenon is mainly given by the system properties, where the downstream volume is most important. Most of the time in a surge cycle, is spent in either emptying or filling the downstream volume [27, 58, 62], and measurements of a clear connection between increased surge frequency and decreased downstream volume is presented in [63]. The filling period of the cycle is longer than the emptying, due to the flow through the throttle downstream of the compressor [62]. The surge frequency also depends on the compressor characteristic, and compressor speed will therefore also affect the surge frequency [27, 62].

The surge phenomenon has a hysteresis effect, where the breakdown of the flow does not follow the same path in the compressor map as the build up of flow [28, 62]. Mass flow measurements of surge presented in [28], show that the flow reversal is conducted at nearly constant pressure, followed by an increasing mass flow at lower pressure, and finally a rapidly increasing mass flow to a steady flow compressor speed line.

The unstable branch of a compressor speed line is modeled using a third order polynomial in [52], and is said to influence the modeled surge cycles to a small degree, since the time spent there is small [64]. Compressor pressure ratio at zero mass flow has been modeled in different versions in [52, 63, 65, 66]. The negative flow branch of an extended compressor map, is modeled using a parabola with good accuracy in [64], and as a second order polynomial in [52]. Further, [52] assumes an efficiency for surged mass flow of 20 %, and an isothermal expansion is assumed in [63].

The Moore-Greitzer approach is followed in both Paper 1 and Paper 3, and a third order polynomial in corrected mass flow is used in both papers for the unstable branch. This polynomial is parametrized to give zero derivative for the zero mass flow pressure build up point. The model of Paper 3 then uses a turbine flow characteristic for the negative flow branch, while Paper 3 uses the third order polynomial also for reversed flow. Constant compressor efficiency is assumed for surging mass flows in both papers.
Bibliography


Chapter 3. Compressor modeling


3.4. Surge and zero mass flow modeling


Chapter 3. Compressor modeling


3.4. Surge and zero mass flow modeling

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