Automatic Tuning of Control Parameters for Single Speed Engines

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Abstract
In Scania’s single speed engines for industrial and marine use, the engine speed is controlled by a PI-controller. This controller is tuned independent of engine type and application. This brings certain disadvantages since the engines are used in a wide range of applications where the dynamics may differ. In this thesis, the possibility to tune the controller automatically for a specific engine installation has been investigated.

The work shows that automatic tuning is possible. By performing an identification experiment on the engine, the parameters in a first order model describing the dynamics of the engine and the load aggregate are determined. The control parameters are then determined as functions of the model parameters.

Measurements on a generator set show that the proposed method provides a controller which is able to follow changes in the reference value, and to compensate for load disturbances.

Keywords: PID-control, GenSet, automatic tuning.
Sammanfattning
I Scania’s envarvsmotorer för industri- och marin-bruk regleras varvtalet av en PI-regulator. Denna regulator är inställd oberoende av motortyp och applikation. Detta medför vissa nackdelar då motorerna används i flera olika typer av applikationer där dynamiken kan variera. I detta arbete har det undersöks huruvida det är möjligt att automatiskt ställa in parametrarna i en PID-regulator för en enskild motorinstallation.


Mätningar på ett generatoraggregat visar att man med hjälp av den föreslagna metoden erhåller en inställning av regulatorn som både klarar av att följa börvärdesförändringar och kompensera för laststörningar.

Nyckelord: PID-reglering, GenSet, auto-tuning.
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Outline of the Report

Chapter 1 This chapter gives an introduction to the master’s thesis project. The conditions and ideas behind the work are described. The problem is specified. The purpose of the thesis and its boundaries are formulated.

Chapter 2 An overview of previous work in the area is given.

Chapter 3 Scania single speed engines are described. Single speed applications are discussed, in particular generator sets. The control problem for generator sets is formulated. The dynamics of a generator set are discussed. Finally, thumb rule requirements in terms of control performance are introduced.

Chapter 4 The Engine Management System in general and the Engine Control Unit in particular, is described. The processor used in the Engine Control Unit is described. An overview of the software architecture is given and the CAN-bus system is presented. A short description of the application tool Gredi is given.

Chapter 5 A summary of the control theory behind the tools used in the thesis is given. The control problem and its relation to this thesis are formulated. The concept of feedback is described. Sensitivity and robustness is discussed and a first order model with time delay is introduced. An introduction to PID control and automatic tuning is given.

Chapter 6 The implementation of the method used to achieve the objectives of this thesis is described and explained. The implementation of the controller is also presented.

Chapter 7 The experimental setup is described and technical data for engines and other equipment are presented.

Chapter 8 The results achieved are presented in the form of measurements. The measurements are analyzed and the results are discussed.

Chapter 9 The results are discussed in detail and conclusions are drawn. The outcome of the thesis is discussed along with the strengths and weaknesses of the tuning procedure developed.

Chapter 10 Ideas for future work are given.
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Appendix A, Plots and Figures
1 Introduction

Scania is one of the leading manufacturers of heavy trucks in Europe. Over 50,000 vehicles are sold worldwide every year. This is a part of Scania that most people are familiar with, but this thesis will focus on Scania industrial and marine diesel engines. These are basically truck engines that are modified for industrial and marine applications. The annual volume of Scania Industrial and Marine engines is approximately 2000 units. Of these, approximately 40% are single speed engines. That is, engines running at one pre-determined speed only. These engines, as all Scania engines, are equipped with an electronic injection system controlled and monitored from an Engine Control Unit (ECU).

1.1 General

The separate diesel engines that Scania manufactures can be used in a wide range of different applications. Some applications are vehicle similar, for example military vehicles, container lifters and boats. In other applications a so called single speed engine is required. In these applications the engine runs at a predefined, constant speed. The vast majority in this category are engines which are mounted into generator sets. A generator set consists of an engine and a three phase generator. These two units are mechanically connected with a coupling. The engine drives the generator which produces electrical energy.

Single speed engines are also used in for example wood chippers and pumps, but generator sets, or GenSets, are the most demanding applications in terms of control. They are also the most common application for single speed engines. As many as 90% of the single speed engines sold by Scania every year are used in GenSets. Therefore the focus of this thesis will be on GenSets.

1.2 The Controller

A controller for the engine in a generator set has two major tasks. The first one is the synchronization phase where the engine must be able to follow small changes in the reference speed in order to synchronize the generator with the mains grid. This means that the engine speed and position must match the frequency and the phase of the mains grid before the generator can be connected. The second task is the load compensating phase. When the generator set is connected to the mains grid, the engine must be able to compensate for load changes. Important features in the controller are thus stability during the synchronization phase and fast response during the load compensation phase.

1.3 The Purpose of the Thesis

To optimize a controller for different engine types and different types of drive trains between the engine and the generator may be difficult. It would therefore be interesting to investigate the possibility of developing functionality in the Engine Control Unit for automatic tuning of the controller based on knowledge of the dynamics in every specific installation. This functionality could be activated on demand from service personnel by the press of a button or via a PC.

1.4 Boundaries of the Work

This thesis covers the following areas:

- A short survey of previous work is made. Similar systems on the market are looked into.
- Generator sets, which is the most common application for Scania single speed engines, are investigated. Other applications are mentioned, but not analyzed.
- A simple mathematical model is produced to describe a generator set.
- A method for controller tuning based on simple knowledge of the dynamics in a generator set is developed.
- A test version of a function for automatic tuning of the engine controller is implemented in the engine control system.
- Verifications of the results are carried out through measurements in a truck and in a generator set.
- Different controllers are evaluated.
2 Previous Work

Very little work in the field of control algorithms for single speed engines exists. The reason for this is probably that the single speed engine market is very limited. Most manufacturers of heavy duty diesel engines sell single speed engines, but these engines are almost exclusively modified carry-over type engines from truck engines. This is certainly the case for Scania’s single speed engines.

The fact that these engines are so inspired by the world of truck engines implies that the controllers often are designed for use in a truck. Scania’s present single speed controller is a modified version of the engine speed controller used in trucks.

Although not in line with the work done in this thesis, a few works with control design for single speed engines can be mentioned.

In [8], the performance of various types of control systems for an electric governor of a diesel engine is examined. A neural network is then applied to regulate the control parameters in the PID controller controlling the injected amount of fuel. The authors show good performance under various operating conditions. However, the engine used is an old fashioned type using an in-line fuel injection pump. This pump is controlled via a DC motor, which is controlled by the PID controller. Modern diesel engines use unit injectors governed from a control unit giving more possibilities in terms of injection timing and precision.

In [9], an overview of the design process used in the development of an electronic diesel engine controller for industrial applications is presented. The paper gives a suggestion of how the entire control system can be built. The focus of the paper is to specify which functionality is needed and what the interface towards the engine should look like. No actual control design in terms of control structure or choice of control parameters is discussed.
3 Scania Single Speed Engines

As said in chapter 1, Scania manufactures and sells engines for industrial and marine use beside the Scania trucks. These engines are basically truck engines modified to meet the demands of the industrial and marine market. One major difference between the market for truck engines and the market for single speed engines is the diversity of applications on the industrial market [9]. Examples such as front-end loaders, rail cars, electric drive haul trucks, generator sets and air compressors can be mentioned. In all of these applications, a single speed engine is often used.

In this thesis, the most common single speed engine application will be studied in greater detail: generator sets. This chapter gives an overview of generator sets, the demands that the controller in such an engine must meet and what dynamics can be expected.

3.1 The GenSet Control Problem

The most common application for single speed engines is generator sets, also called GenSets. A GenSet consist of a diesel engine coupled to a three phase generator and sometimes an external control system. GenSets are used where no mains grid is available for power supply, or where there is a need for back up power in case of mains failure. Industries also use them to handle power peaks during limited periods of time. Scania does not sell complete GenSets, only the engine with its own control system and sensors. The GenSet manufacturers add the generator and usually a separate control system with its own sensors.

There are two main objectives for the controller in a single speed engine. The controller must provide accurate set point tracking as well as rapid compensation of load disturbances without steady state error. The closed loop system is shown in Figure 1, where the load disturbance $l$ enters the system at the process input and measurement noise $n$ enters the loop at the output. The noise is negligible in this case and will not be taken into account in this thesis. In Figure 1, $r$ is the reference signal, $e$ is the control error, $u$ is the control signal, $x$ is the undisturbed output and $y$ is the measured output.

The tracking of set point changes is important when trying to synchronize the GenSet to an electric grid. The compensation of load disturbances is important during normal operation.

![Figure 1. The closed-loop system](image)

3.1.1 Synchronization

Before a GenSet can be connected to an electric grid, its output must be synchronized to that grid. This implies that the voltage delivered from the generator must be equal to the voltage of the grid. This is handled with a voltage regulator separated from the Engine Control Unit. The frequency of the generator, i.e. the speed with which it is turning, must also be matched to the frequency of the grid. A third condition must also be fulfilled. The phase angle of the generator must match the phase angle of the grid. The synchronization of frequency and phase angle introduce requirements on the engine speed controller. The engine speed must be controlled to a stable level matching the frequency of the grid, but the controller must also be able to respond to small changes in the reference speed in order to make it possible to match the phases of the generator and the grid. When the generator is connected to the grid, different characteristics in the controller become important.
3.1.2 Load Disturbances

When the GenSet is running in normal operation producing electric power, the engine speed and thus the generator frequency, must be held at a constant level. In this thesis, it will be assumed that the engine is operating isochronous, that is, no droop regulation is used. Droop regulation means that the nominal speed is changed depending on the engine load. This is used to share load between several GenSets coupled in parallel. This will not be pursued any further.

Scania sells engines for 50 and 60 Hz operation which corresponds to engine speeds of 1500 revolutions per minute [rpm] and 1800 rpm. This thesis will be focused on 1500 rpm nominal engine speed.

As the load on the generator changes, the engine will experience changed resistance in turning the generator. This will cause a change in engine speed. If the load is increased, the engine speed will drop. The task of the controller is now to compensate for the load disturbance in a way such that the engine speed returns to the desired value as smoothly and fast as possible.

If load disturbances where measurable in some way, it would be an advantage to use this information by applying feed forward. Unfortunately the Engine Control Unit does not have access to any information concerning disturbances entering the loop. In some cases, an external control system monitoring the output from the generator is present. In these cases feed forward of rapid changes in the output current might provide faster compensation of load disturbances, but there is no communication between the Engine Control Unit and these external systems. Therefore no feed forward will be considered in this work.
3.2 GenSet Dynamics

The dynamics of a GenSet are mainly of the first order. Later, a first order model will be introduced to describe such a system. The reason for this is the short and comparably stiff driveline used between the engine and the generator. The engine and the generator alone can both be approximated as first order systems if viewed as rotating masses. Since the coupling often is very stiff, the GenSet can be viewed as a large rotating mass and thus a first order system. The drive train is very different from that of a truck, which is much more flexible, giving rise to oscillations.

There are numerous plates and axles used as couplings with Scania’s engines. Scania has no way of specifying which couplings that should be used. This is entirely up to the customer. Scania does however get some information concerning the couplings on the market.

One commonly used coupling is the so called flex plate. A flex plate has been used in this work. The plate is mounted between the generator and the engine. It is flexible along the direction of the crankshaft in order to absorb vibrations. Flex plates are very torsionally stiff giving good robustness to low frequency oscillations. It is likely that a GenSet with such a coupling can be approximated to a first order model.

Another coupling is the rubber coupling. It is not as common as the flex plate. As the name suggests, it is partially made of rubber. A disc with beams is mounted on the engine flywheel. A disc with holes lined with rubber dampers is mounted on the generator. When the engine is mounted to the generator, the beams are inserted into the holes and the engine and generator are fixed to a steel frame. This type of coupling permits fast and simple replacement of an engine in case of failure or service. Half of a rubber coupling is shown in Picture 1, where the disc mounted on the generator can be seen with its rubber dampers. These dampers are reasonably stiff, but it is likely that the rubber coupling is more flexible than the flex plate. This has not been investigated any further in this work. It is not wise to say that a GenSet with rubber coupling can be viewed as a first order system. This would have to be investigated further.

![Picture 1. A rubber coupling.](image)

3.3 Requirements

As always, certain requirements set by the market must be met. In the world of GenSet, important aspects are of course price and reliability. Apart from these demands, customers also require adequate performance in terms of load disturbance damping, control speed and accuracy. These requirements must be taken into consideration when designing a controller. Unfortunately, no clear statement concerning which requirements to meet can easily be found. [10] and [11] specify requirements on GenSets and engines used in GenSets, but no information concerning control requirements can be found in these standards. Instead, much of the input in these matters is taken directly from the customers and their general idea of what is acceptable and what is not.
Requirements stated by a Scania customer are illustrated in Figure 2. The figure shows how the engine is started and ramped up to its low idle speed, usually 1000 rpm. The time at the low idle level can be specified. Some customers want to warm up the engine before loading it, others use their GenSet as backup power and then the engine must be ready for load as fast as possible.

When the low idle phase is finished, the engine is ramped up to its nominal speed. A maximum of 1% overshoot is acceptable. The overshoot must be eliminated within 1 second. This is not really a control problem since the slope of the ramp can be adjusted to get good performance. After being ramped up to nominal speed, the engine must be synchronized to the electrical grid. This requires less than 0.2% control error corresponding to 3 rpm for a nominal speed of 1500 rpm. 5s of synchronization is tolerated, and then the GenSet must be synchronized to the grid.

When the synchronization is complete, the engine must be able to handle load disturbances in the grid. When the load increases from 0% to 60% of maximum load, an undershoot of maximum 3% with settling time of less than 2s is required. When the load is decreased from 60% to 0%, an overshoot of 2.5% and settling time of less than 1s is required.

![Figure 2. Thumb rule requirements stated by Scania customer.](image-url)

Figure 2 shall not be seen as a formal stated requirement, but just a thumb rule. If the requirements of Figure 2 are met, it is likely that all customers are satisfied.

As stated in chapter 3, single speed engines are not only used in GenSet applications, but in most other applications control performance is not as vital as in a GenSet. In a GenSet, the frequency of the generated voltage is directly depending on the engine speed. If equipment sensitive to frequency disturbances is coupled to the grid powered by the GenSet, it is important to minimize these disturbances. Such equipment is for example power converters present in many products, for example computers.

Since the toughest requirements are stated by GenSet customers, these requirements will be taken into account in this work.
4 Scania Engine Control Unit

Combustion engines of today are in general controlled and monitored from an electronic control unit. This unit gathers information concerning the state of the engine via numerous sensors. A micro computer reads the sensor values and performs the calculations specified in its software. Output signals are generated and sent to actuators, for example fuel injectors.

Scania develops software for their own engine management system. The system in use today is called S6. S6 is the system where all function implementation described in this work will be done. This chapter describes the different parts of the Scania EMS (Engine Management System). The processor used in the control unit is described along with the software architecture. The communication bus used by the EMS is briefly looked into. Finally, an interface to the EMS used for development purposes is presented.

4.1 Motorola MPC555

The processor used in Scania Engine Control Unit is a Motorola MPC555. The MPC555 processor is a member of Motorola’s MPC500 PowerPC™ RISC Micro-controller family. It has the following features according to [12]:

- PowerPC core with floating point unit
- 26 Kbytes fast RAM and 6 Kbytes TPU microcode RAM
- 448 Kbytes flash EEPROM with 5-V programming
- 5-V I/O system
- Serial system: queued serial multi-channel module (QSMCM), dual CAN 2.0B controller modules (TouCAN™)
- 50-channel timer system: dual time processor units (TPU3), modular I/O system (MIOS 1)
- 32 analog inputs: dual queued analog-to-digital converters (QADC64)
- Submicron HCMOS (CDR1) technology
- 272-pin plastic ball grid array (PBGA) packing
- 40 MHz operation -40˚ C to 125˚ C with dual supply (3.3 V, 5 V)

For more information, please refer to the MPC555 user’s manual [12].

4.2 Software Architecture

The program used in the Engine Control Unit is constructed of modules, without global variables. This makes it easy to expand the code when new functionality is developed. This has been an advantage used in this work. The program is mainly written in C-code with the system core initialization in assembler. To enable the use of the same code for several different types of engines, functions and calibration parameters are separated.

The scheduling of task can be described as time synchronous static scheduling interrupted with cam shaft synchronous static tasks. The periodic execution time for the time synchronous tasks is 10ms or 50ms. The functions implemented in this work are executed in the 10ms loop. CAN communication can interrupt both time synchronous and cam shaft synchronous tasks.
4.3 **CAN 2.0B Controller Module**

[12] gives the following information concerning the CAN controllers of the MPC555. “The MPC555 contains two CAN 2.0B controller modules (TouCAN). Each TouCAN is a communication controller that implements the controller area network (CAN) protocol, an asynchronous communications protocol used in automotive and industrial control systems. It is a high speed (1Mbit/sec), short distance, priority based protocol that can run over a variety of mediums (for example, fiber optic cable or an unshielded twisted pair of wires). The TouCAN supports both the standard and extended identifier (ID) message formats specified in the CAN protocol specification, revision 2.0, part B.”

Each TouCAN module contains 16 message buffers, which are used for transmit and receive functions. It also contains message filters, which are used to qualify the received message IDs when comparing them to the receive buffer identifiers.

The Scania Engine Control Unit use one of the CAN controllers for communication with other control units on the CAN-bus, see section 4.5. The other CAN controller is used as an interface to application tools such as Gredi, see section 4.6.

4.4 **EMS**

The Engine Control Unit is the main component of the EMS. All engine aggregates are controlled from the Engine Control Unit and all engine sensors are connected to it. The entire EMS is shown in Figure 3. It is connected to the CAN-bus system as shown in Figure 4.

![Figure 3. The Scania EMS](image-url)
4.5 **CAN-bus System**

In a Scania truck there are numerous control units. All of them are connected to a CAN-bus. The CAN-bus system is shown in Figure 4.

There are three busses and each bus is characterized by how important its units are. The red bus is crucial for safety and operation of the truck. Units on the red bus are for example the EMS and the Brake Management System. The yellow bus holds units that are important but not crucial for the operation of the vehicle, for example the Instrument Cluster. The green bus holds units such as Automatic Climate Control and Audio System. To connect the three busses, a supervising control unit (Coordinator) is used. It is possible to communicate with the units on the CAN-bus via the diagnostic bus entering the green bus. The number of units connected to the CAN-bus may vary from vehicle to vehicle depending on specification. The EMS is however always present. Industrial and Marine engines naturally always have EMS. For most applications a coordinator is also present to enable communication with external control systems.

![Figure 4. The CAN-bus system in a Scania truck.](image)

4.6 **Gredi**

One of the CAN controllers is used as an interface to a CAN-based application tool called Gredi. Gredi is developed by a company called Kleinknecht. The tool is used to log RAM variables and to change calibration parameters. This can be done on line when the engine is running. Gredi also support recording and plotting of all variables in the Engine Control Unit, which has been used in this work.

![Figure 5. A common configuration of the CAN-bus system in a Scania Industrial and Marine application.](image)
5 Control Theory

A summary of the control theory behind the tools used in the thesis is given in this chapter. The control problem and its relation to this thesis is formulated. The concept of feedback is described. Sensitivity and robustness is discussed and a first order model with time delay is introduced. An introduction to PID control and automatic tuning is given.

5.1 The Control Problem

According to [5], page 10, the control problem may be defined as “find with guidance of the output, an input such that the purpose of the system is fulfilled”. In [5], page 11, the control problem is divided into two cases. In the servo case, the goal is to have the output signal follow the reference signal. In the regulator case, the output signal shall be kept constant independently of disturbances entering the control loop.

As stated in section 3.1, the purpose of the system described in this thesis is for the output to follow small changes in the given reference and to keep the output constant independently of load disturbances. Hence both the servo case and the regulator case are included.

5.2 The Open- and the Closed-loop System

In Figure 1 the closed-loop system for the control problem studied in this thesis is shown. In this section the general control problem will be discussed.

If the output \(z(t)\) of the system in Figure 7 is not measurable the controller needs to know the exact dynamics of the system \(G(s)\) to be able to control it. It must also have information concerning load disturbances entering the system. If disturbances are neglected, the system may look like Figure 6.

![Figure 6. The Open Loop system.](image)

![Figure 7. The closed-loop system.](image)

If the output \(z(t)\) is known through feedback, as in Figure 7, the controller \(F(s)\) only has to approximate the dynamics of the system. In Figure 7 \(l(t)\) is a load disturbance, \(w(t)\) is a system disturbance and \(n(t)\) is measurement noise. \(w(t)\) and \(n(t)\) are neglected in this work since they are small. When feedback is used, the controller can continuously adjust the input signal \(u(t)\) by observing the control error \(e(t)\). \(F(s)\) only has to approximate the dynamics of \(G(s)\) since
If the output $y(t)$ is supposed to follow the reference $r(t)$, it is desirable that Equation 2 holds. This implies that the condition in Equation 3 must be fulfilled.

**Equation 2**

$$\frac{F(s)G(s)}{1 + F(s)G(s)} \approx 1$$

**Equation 3**

$$F(s)G(s) \gg 1$$

The challenge when designing the controller $F(s)$ is to fulfill the condition in Equation 3 without making the closed-loop system unstable.

### 5.3 Sensitivity and Robustness

In reality, the model $G(s)$ used when designing a controller, does not entirely describe the real system. The real system is often more complex. Measurement noise and other disturbances may enter the system. It is important that the controller can achieve its purposes even though the model does not capture all system dynamics and there are noise and disturbances present. Sensitivity and robustness are two terms used to describe the ability of the system to handle such imperfections.

#### 5.3.1 Sensitivity

It is desirable that the closed-loop system is insensitive to noise, disturbances or errors in the model. Consider the closed-loop system in Figure 7, where $l$ and $w$ are disturbances influencing the controlled quantity $z$. $n$ is noise influencing the measured output $y$.

The transfer function from $w$ to $y$ is given by Equation 4. $S(s)$ is the sensitivity function and describes how disturbances are suppressed by the system.

**Equation 4**

$$S(s) = \frac{1}{1 + G(s)F(s)}$$

Suppose that the true system is given by Equation 5 where $G(s)$ is the model of the true system $G_0(s)$ and $\Delta G(s)$ is the relative error of the model. If noise is neglected, the true output of the closed-loop system is given by Equation 6 instead of Equation 7. The difference between $z$ and $z_0$ is given by Equation 8.

**Equation 5**

$$G_0(s) = (1 + \Delta_G(s))G(s)$$

**Equation 6**

$$z_0 = \frac{G_0(s)F(s)}{1 + G_0(s)F(s)}r$$

**Equation 7**

$$z = \frac{G(s)F(s)}{1 + G(s)F(s)}r$$
Equation 8

\[ z_0 = (1 + \Delta_z)z \]
\[ \Delta_z = S_0(s) \Delta_G(s) \]
\[ S_0(s) = \frac{1}{1 + G_0(s)F(s)} \]

The sensitivity function \( S_0(s) \) describes how the relative error of the model transforms into a relative output error. The closed-loop system is insensitive to model error at frequencies where the sensitivity function is small. It is desirable to keep \( S(s) \) small at low frequencies where model errors often exist.

### 5.3.2 Robustness

If the model \( G(s) \) is not a perfect model of the real system there exists a relative error of the model, \( \Delta_G(s) \). It is important to know how large the relative error can be without affecting the stability of the closed-loop system.

Suppose that \( \Delta_G(s) \) is a stable transfer function. This means that \( G \) and \( G_0 \) have the same stability properties. The stable transfer function given in Equation 9 is called the complementary sensitivity function. The name refers to the fact that Equation 10 is always fulfilled. Note that the complementary sensitivity function is equal to the transfer function of the closed loop system. This is a result of the fact that no pre-compensator is used in this thesis.

**Equation 9**

\[ T(s) = \frac{G(s)F(s)}{1 + G(s)F(s)} \]

**Equation 10**

\[ T(s) + S(s) = 1 \]

Since both \( G(s) \) and \( \Delta_G(s) \) are stable, the small gain theorem can be used to conclude whether the closed-loop system is stable or not. The stability condition is given by Equation 11 where the small gain theorem has been used. The small gain theorem is not discussed in this thesis. Please refer to control theory literature, for example [6], page 179, for further information.

**Equation 11**

\[ |T(i\omega)| \leq \frac{1}{|\Delta_G(i\omega)|}, \forall \omega \]

It is desirable to keep \( T(s) \) small at high frequencies where noise that disturbs measurements may be present.
5.4 Modeling

Many methods for tuning controllers are dependent on some kind of plant model which describes the plant dynamics that are of interest. The model can be of varying complexity, but a simple model can often capture enough dynamics for the control design procedure to be successful. There are different ways of obtaining a model. One approach is to model the plant using the laws of physics in combination with known relationships between plant variables. Another approach is to excite the plant in some way, most often by applying a pre-specified input. The output of the plant is then studied and conclusions regarding a suitable model structure can be drawn. Such methods can be based on the plant’s frequency response or its transient response. Transient response methods are particularly suitable for identifying simple models. In this chapter the structure and derivation of the model used in this work will be studied.

5.4.1 Model Structure and Parameter Identification

In this work, the system consists of a Scania diesel engine and an unknown generator coupled to it. In [7] it has been shown that the engine alone can be modeled as a first order system with time delay. In [7], Pettersson and Forsell show good correlation of model output with measurements. With this in mind, this model structure has been used in this work. The model is shown in Equation 12, where \( K_p \) is the static gain, \( T \) is the time constant and \( L \) is the dead time of the system. \( K_p \) is given as engine speed per injected fuel amount [rpm/mg/st]. \( T \) and \( L \) are given in seconds [s].

\[
G(s) = \frac{K_p}{1 + sT} e^{-stL}
\]

When using the model of Equation 12, the system is assumed to be linear apart from the time delay. This is not entirely true. There are non-linear aspects such as engine friction and turbo lag, which will be discussed further in chapter 8.

The input to the model is the injected fuel amount given as mg/stroke (mg/st). The output is the engine speed given as rpm. The model parameters are determined from a step response. When the engine is running at idle speed a step in the injected fuel amount is commanded, giving a rise in engine speed. The three model parameters are determined as shown in Figure 8. The step is applied at \( t=0 \). The time delay \( L \) is equal to the time it takes for the output to start its increase. The time constant \( T \) can be derived by drawing the tangent to \( y(t) \) at
the steepest point of the curve. \( T \) is the time between the intersection with the tangent and the idle level, \( y_0 \), and the intersection with the tangent and the final value \( y_f \). An alternative way to compute \( T \) is by determining the point from when the step is applied to the point where the output has reached 63\% of its final value. If the dynamics are perfectly described by Equation 12, these two methods give the same \( T \). Otherwise the first method tends to give too large values of \( T \). The steady state gain \( K_p \) is the quota of \( y_f \) and the injected fuel amount when the step is applied.

### 5.5 PID Control

The PID controller is by far the most common control algorithm. There are other algorithms available, but PID will be used in this work because of its simplicity and intuitive transparency. It is also the algorithm used by Scania today.

In this section, the basic PID algorithm is presented as well as a description of the properties of the controller in a closed-loop based on intuitive arguments regarding the performance. Integral windup, which occurs when the controller has integral action and the process has a saturating control signal, is discussed along with ways to avoid it. Some aspects of digital implementation are also given.

#### 5.5.1 General

The general form of the PID algorithm is shown in Equation 13 where \( u \) is the control variable and \( e \) is the control error which is the difference between the set point \( r \) and the measured output value \( y \). The control variable is the sum of three terms: The P-term, which is proportional to the error, the I-term, which is proportional to the integral of the error and the D-term which is proportional to the derivative of the error. The three corresponding parameters are the proportional gain \( K \), the integral time \( T_i \) and the derivative time \( T_d \).

\[
\text{Equation 13}
\]

\[
u(t) = K \left[ e(t) + \frac{1}{T_i} \int e(s) ds + T_d \frac{de(t)}{dt} \right]
\]

When transforming the control law given in Equation 13 using the Laplace transform, Equation 14 is obtained.

\[
\text{Equation 14}
\]

\[
U(s) = K \left[ 1 + \frac{1}{T_i s} + s T_d \right] E(s)
\]
Proportional Action
Proportional control is the simplest form of feedback. If pure proportional control is used, Equation 13 reduces to Equation 15 where the control signal is simply proportional to the control error with the controller gain $K$.

\[ u(t) = Ke(t) \]

A large value of $K$ will give a large loop gain and the closed loop will be insensitive to load disturbances but sensitive to measurement noise. If the loop gain is increased too much, the closed loop may become unstable. There will always be a steady state error with pure proportional control. This can be understood from Equation 15 which shows that a control error is necessary in order to have a non-zero control signal. Proportional controllers can therefore be extended with a reset term, $I$, to obtain a correct steady state. The corresponding control law is given in Equation 16. Another, and more commonly used, way of avoiding a steady state error is to use a controller which includes integral action.

\[ u(t) = Ke(t) + I \]

Integral Action
The main function of the integral action is to obtain a zero steady state error. It was seen earlier that proportional control requires a control error in order to have a non-zero control signal. When using integral action in the controller, a small positive error will always lead to an increasing control signal. The fact that the steady state error will be zero can be motivated with a simple argument. Assume that the system is in steady state with a constant control signal and a constant control error. It follows from Equation 13 that the control signal then will be given by Equation 17.

\[ u_0 = K \left[ e_0 + \frac{e_0}{T_i} t \right] \]

As long as the error is non-zero this contradicts that the control signal is constant. Thus a controller with integral action will always give zero steady state error. This makes integral action crucial for the regulator case described in section 5.1. The integral time $T_i$ determines how fast the error is eliminated. For large values of $T_i$ the output of the controlled process approaches the value of the set point slowly. For smaller values of $T_i$ the approaches is faster but on the other hand also more oscillatory.

Derivative Action
Derivative action is used to improve the stability of the closed loop. Intuitively, the instability mechanism can be described as follows. As a cause of process dynamics, a change in the control signal will not be noticed on the process output immediately. The controller is always one step behind when trying to correct an error. The derivative action can be interpreted as if the controller is predicting what will happen in the near future. In fact, the prediction reaches $T_d$ seconds ahead. Initially damping increases with $T_d$, but when the derivative time becomes too large the closed loop becomes oscillatory.
5.5.2 Integrator Windup and Anti-windup

So far the PID controller and its properties have been accounted for using linear theory. However, when controlling many processes one must consider non-linear effects as well. All actuators have limits. A valve can only be fully opened or fully closed. If (or more likely, when) the control signal reaches the actuator limits the feedback loop is broken and the system runs as an open loop since the actuator will remain at its limit independently of the process output. If a controller with integrating action is used, the error is integrated and the integral term will become very large. It “winds up”. It is then required that the error has opposite sign for a long period before the integral term has reduced and the control signal reached an interval where the actuator is not saturated. This phenomenon may result in large transients when the actuator saturates. This can be avoided by using an anti windup algorithm which prevents the integral term from growing to large.

5.5.3 Influence of Measurement Noise

If measurement noise enters the loop, the derivative action of the controller may cause problems. Measurement noise is often present in the high frequency region which implies that it will be amplified by the derivative term. Let the noise \( n \) be a sinusoid with frequency \( \omega \) as in Equation 18.

Equation 18

\[ n = a \sin(\omega t) \]

The noise will then have the contribution to the control signal shown in Equation 19 where it can be seen that the noise is amplified by a factor \( \omega \).

Equation 19

\[ u_n = KT_d \frac{dn}{dt} = aKT_d \omega \cos(\omega t) \]

To avoid this, the high frequency gain of the derivative term must be limited. This can be done by filtering the derivative term through a low pass filter. For a first order filter the derivative term becomes Equation 20 where \( N \) is adjusted to give the desired cut off frequency. This limits the high frequency gain to \( K/T_f \).

Equation 20

\[ D = \frac{sKT_d}{1 + sT_f} \]

5.5.4 Digital Implementation

PID controllers are today almost exclusively implemented digitally using micro processors. This provides a simpler synthesis than old approaches where controllers were implemented with mechanical systems or analog electric circuits, but there are a few pitfalls when using micro processors to run control algorithms. Special care has to be taken when implementing integrations and derivations, as well as when time continuous signals are made discrete.

Sampling and Anti-aliasing

A micro processor executes the control algorithm at discrete instances of time, i.e. the system is sampled. If the sampling frequency is not chosen with care, high frequency signals may appear as low frequency signals in the system. According to the sampling theorem, this can be avoided by sampling with a sampling frequency twice as high as the highest frequency that is of interest. Before sampling all signals should be filtered through an anti aliasing filter with cut off frequency such that no frequencies higher than half the sampling frequency will be present.
Discretization

When a time continuous controller as in Equation 13 is implemented in a digital computer, the proportional, integral and derivative parts have to be approximated in some way using the discrete values of the variables that are available. The proportional part is simply implemented by multiplying the discrete error with the proportional gain. The time continuous integral part is given by Equation 21.

Equation 21

$$I(t) = \frac{K}{T_i} \int_{0}^{t} e(s) ds$$

This is often implemented as shown in Equation 22 where $h$ is the sampling interval.

Equation 22

$$I(k) = I(k-1) + \frac{Kh}{T_i} e(k)$$

The derivative term is given by Equation 23 which can be implemented with a backward difference as in Equation 24.

Equation 23

$$D(t) = KT_d \frac{de}{dt}$$

Equation 24

$$D(k) = KT_d \frac{e(k) - e(k-1)}{h}$$

There are several other ways to approximate integral and derivative parts of the controller, but they will not be looked into in this work. The implementation of the controller in this work has been done as described above.

5.6 Automatic Tuning of PID Controllers

The term automatic tuning, or auto-tuning for short, refers to a method where the controller is tuned automatically on demand from a user. The user may push a button or give some other kind of command to the controller. The automatic tuning procedure consists of three steps:

- Generation of a process disturbance
- Evaluation of the disturbance response
- Calculation of controller parameters.

This can be seen as an automated form of the method often used by people with large experience in tuning controllers.

The generation of a disturbance and the evaluation of the response have already been discussed in section 5.4.1. In this chapter, the last step will be discussed. Different approaches to parameter calculation will be described. First, different tuning rules for PID controllers will be described. Then a pole placement design for PI controllers will be introduced.

5.6.1 Tuning Rules

If the process dynamics are known, tuning rules can be applied. Such rules give the control parameters as functions of parameters describing the process. The rules are often built on extensive empirical tuning experience. Controllers have been manually tuned for a wide range of processes, and the result have been formalized into equations. Different tuning rule methods give different closed-loop characteristics. Some may produce controllers that offer good set point following and others may be focused on compensation of load disturbances.
Zeigler-Nichols Method
The most widely known rule based tuning method is the Ziegler-Nichols method. According to [2], it was presented in 1942 and has been widely used, both in its original form and with some modification. The method is based on some features of the process dynamics. The control parameters are then expressed in terms of these features. There is one set of Ziegler-Nichols rules based on a frequency response. They will not be discussed here. The method evaluated in this thesis is the Ziegler-Nichols step response method.

Although widely used, the method is not state of the art. It might even be considered old fashioned. According to [2], page 142, a large drawback is that the closed-loop system becomes poorly damped and has poor stability margins. Even though the Ziegler-Nichols method is not the best method available for control design, it has been used in this work since it is simple, intuitive and easy to implement.

The Ziegler-Nichols step response method is based on two parameters, $a$ and $L$, characterizing the process. They can be determined from a step response, as suggested by the name of the method. In section 5.4.1, the identification of the process parameters $K_p$, $L$ and $T$ was described. $L$ is used as it is in this method. $a$ is computed according to Equation 25, according to [4] page 378.

Equation 25

$$a = \frac{K_p L}{T}$$

The control parameters are then given by Table 1.

<table>
<thead>
<tr>
<th>Controller</th>
<th>$K$</th>
<th>$T_1$</th>
<th>$T_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>P</td>
<td>$1/a$</td>
<td></td>
<td></td>
</tr>
<tr>
<td>PI</td>
<td>$0.9/a$</td>
<td>$3L$</td>
<td></td>
</tr>
<tr>
<td>PID</td>
<td>$1.2/a$</td>
<td>$2L$</td>
<td>$L/2$</td>
</tr>
</tbody>
</table>

Table 1. PID controller parameters given by the Ziegler-Nichols step response method.
Cohen-Coons Method

The Cohen-Coon method is based on the process model described in Equation 12, section 5.4. The equation is repeated below for convenience.

\[ G(s) = \frac{K_p}{1 + sT} e^{-sL} \]

According to [2], page 180, the main design criterion is rejection of load disturbances. The method attempts to position dominant poles that give a quarter amplitude decay ratio. For PID controllers the integral gain \( I = K/Ti \) is maximized. This corresponds to minimization of the integral error due to a unit load step disturbance. For PID controllers three closed-loop poles are assigned; two poles are complex, and the third real pole is positioned at the same distance from the origin as the other poles. The pole pattern is adjusted to give quarter amplitude decay ratio, and the distance of the poles to the origin are adjusted to minimize the integral error.

Since the process is characterized by three parameters (\( K_p, L, \) and \( T \)), it is possible to give a tuning rule where controller parameters are expressed in terms of these parameters. Such formulas were derived by Cohen and Coon based on analytical and numerical computations. The formulas are given in Table 2. The parameters \( a \) and \( I \) are given by Equation 26. They are then used in the equations in Table 2.

Even though Cohen-Coons method is slightly more refined than Ziegler-Nichols method, it is not a top-of-the-line control design method. A disadvantage of the method is that the decay ratio is too small, which means that the closed-loop system has low damping and high sensitivity according to [2], page 181. Although more complex than Ziegler-Nichols method, Cohen-Coons method is still simple and easy to use. It requires limited process knowledge and is therefore suitable for the purposes of this work.

**Equation 26**

\[
\begin{align*}
    a &= \frac{K_p L}{T} \\
    \tau &= \frac{L}{L+T}
\end{align*}
\]

<table>
<thead>
<tr>
<th>Controller</th>
<th>( K )</th>
<th>( Ti )</th>
<th>( T_a )</th>
</tr>
</thead>
<tbody>
<tr>
<td>P</td>
<td>( \frac{1}{a} \left( 1 + \frac{0.35\tau}{1-\tau} \right) )</td>
<td></td>
<td></td>
</tr>
<tr>
<td>PI</td>
<td>( \frac{0.9}{a} \left( 1 + \frac{0.92\tau}{1-\tau} \right) )</td>
<td>( \frac{3.3 - 3.0\tau}{1+1.2\tau} )</td>
<td></td>
</tr>
<tr>
<td>PID</td>
<td>( \frac{1.35}{a} \left( 1 + \frac{0.18\tau}{1-\tau} \right) )</td>
<td>( \frac{2.5 - 2.0\tau}{1 - 0.39\tau} )</td>
<td>( \frac{0.37 - 0.37\tau}{1 - 0.81\tau} )</td>
</tr>
</tbody>
</table>

Table 2. Controller parameters obtained from the Cohen-Coon method.
5.6.2 Pole Placement

Properties of the closed-loop system in terms of step response and other specifications are dependent on where the poles are placed. If a model of the process to be controlled is known, the poles of the closed-loop system can be specified and then the control parameters can be calculated to fulfill these specifications.

The speed of the system depends on the poles distance from the origin or the frequency of the poles. The damping of the system is determined by the relative damping factor which is cosines of the poles argument in the complex plane. Please refer to Figure 9 for an illustration.

For a given distance, nice damping properties are achieved for damping factors around 0.7 according to [5], page 173. Of course all poles must be located in the left half plane for the closed-loop system to be stable. Appropriate placement of poles is illustrated in Figure 10.

![Figure 9. Pole placement for second order system.](image)

For systems of low order, the closed-loop poles can be placed using PID control. In this thesis, a first order model is used. This means that if a PID controller is used, the closed-loop system will have two poles. There will be two poles for both PI and PID control since the D-part only adds a zero. Two poles to place imply that only two degrees of freedom in the controller are needed to place these poles. Therefore a PI controller will be used here. This will not give any over determined systems of equations as would be the case with a PID controller.

The system to be controlled is described in Equation 12, section 5.4. The equation is repeated below for convenience. Here, the time delay \( L \) will be disregarded since it does not affect the location of the continuous time poles. The time delays present in the systems discussed in this thesis are small implying that they have little impact when controlling the system. Please refer to chapter 8 for measured values of \( L \).

\[
G(s) = \frac{K_p}{1 + sT} e^{-sL}
\]

By controlling this process with the controller in Equation 27, a second order closed-loop system is obtained according to [2], page 173. The two closed-loop poles can be chosen by a suitable choice of the proportional gain \( K \) and the integral time \( T_i \).

Equation 27

\[
F(s) = K \left[ 1 + \frac{1}{T_i s} \right]
\]
The poles of the closed-loop system are given by the solutions to Equation 28 which is the characteristic equation.

**Equation 28**

\[ 1 + FG = 0 \]

For the process model and controlled discussed here, the characteristic equation becomes

**Equation 29**

\[ s^2 + s \left( \frac{1}{T} + \frac{K_p K}{T_T} \right) = 0 \]

where the time delay \( L \) has been disregarded. If the poles are characterized by their relative damping \( \zeta \) and their frequency \( \omega \) the characteristic equation becomes

**Equation 30**

\[ s^2 + 2\zeta \omega s + \omega^2 = 0 \]

Identifying the coefficients of Equation 29 in Equation 30 and then solving for the control parameters give the parameters stated in Equation 31.

**Equation 31**

\[
K = \frac{2\zeta \omega T - 1}{K_p} \\
T_i = \frac{2\zeta \omega T - 1}{\omega^2 T}
\]

In [7], the parameters \( K_p \) and \( T \) of Equation 12 are given the values \( K_p=76 \) and \( T=2.75 \). Based on this and on knowledge of the engine to be used in this work, the parameters are estimated to \( K_p=75 \) and \( T=3.3 \). Combining this with the control parameters \( P=1.0 \) and \( T_i=0.125 \) used in Scania’s single speed engines today gives the closed-loop parameters \( \omega=13.4 \) rad/s and \( \zeta=0.85 \).
6 Implementation

In this chapter, the implementation of the automatically tuned controller in the Engine Control Unit is described. First, the implementation of the automatic step response experiment is described. Then the implementation of the controller is described.

6.1 Automatic Step Response Experiment

The step response experiment for identification has been implemented as a state machine with four different states. This state machine performs a specified number of steps and calculates model parameters to be used for control design. The step response experiment is fully automatic and can be activated by an operator via the application tool Gredi described in section 4.6. When the step response experiment is active, the engine is completely controlled by the Engine Control Unit. No input from the operator (other than test deactivation) is considered. The states of the step response state machine are described below.

6.1.1 START

The default state of the state machine is START. The step response experiment is activated when a step enabling variable is set to true. This is done using the application tool Gredi described in section 4.6. Then the reference speed is set to a pre-specified value from which the step will start. The fuel is now commanded by a controller and the fuel amount is logged. A clock is started and when the engine speed has been stable at the start level for a specific amount of time, the state of the state machine is changed to RUNNING. If the engine speed does not stabilize in a given period of time, the state will change to STOP.

6.1.2 RUNNING

In state RUNNING, an offset from the mean value of the injected fuel amount logged in state START will be made. The engine speed is monitored by keeping the last 20 engine speed samples in an array. At every loop, a mean derivative over these 20 samples is calculated. The maximum derivative is stored and updated when the state machine is in state RUNNING. The maximum engine speed is also updated and stored at each loop. The time it takes for the last sample of the array to become larger than the engine speed start value is also stored.

When the magnitude of the calculated derivative is smaller than a given value, a clock is started. When the speed has been stable for a given time, the state is changed. If the maximum speed is high enough, the next state will be CALCULATE. If the maximum speed is too high or too low, the next state will be START and the fuel offset will be adjusted. If no stable level is found, the state will go to START. If this happens, the fuel offset might have been too large and it will be adjusted.

6.1.3 CALCULATE

The maximum values of derivative and engine speed stored in state RUNNING are now used to calculate the model parameters $K_p$ and $T$ as described in section 5.4.1. The time delay is calculated as the stored delay time from CALCULATE minus twenty times the sampling period. The parameters are then stored in arrays with room for parameters from a specified number of steps responses. When the specified number of successful steps has been completed, the state will change to STOP. Otherwise the state will be changed to START.

6.1.4 STOP

The step enabling variable is set to false and the state is changed to START.

Figure 11 gives an overview of the state machine and the conditions for switching between the states.
6.2 Controller

Control parameters are calculated using values for the model parameters obtained from the step response experiment described in section 6.1 and either of the methods described in section 5.6. The controller implemented is on standard PID form as stated in Equation 13. It is repeated below for convenience.

\[ u(t) = K \left[ e(t) + \frac{1}{T_i} \int e(s) ds + T_d \frac{de(t)}{dt} \right] \]

The integral term and the derivative term are discretized as described in section 5.5.4. When PI-control is used, \( T_d \) is simply set to zero. The time discrete PID control algorithm is shown in Equation 32.

Equation 32

\[ u(k) = \left[ Ke(k) + I(k-1) + \frac{K}{T_i} e(k) + KT_d \frac{e(k) - e(k-1)}{h} \right] \]

The derivative part is low pass filtered using a filter implemented as Equation 33 to avoid influence of high frequency measurement noise. This is a time discrete version of a standard first order low pass filter as given in

Equation 33

\[ D_{filt}(k) = D(k-1) \left( 1 - e^{-h/T_f} \right) + D_{filt}(k-1) e^{-h/T_f} \]

Equation 34

\[ D_{filt}(t) = D(t) \frac{1}{1 + sT_f} \]

A simple form of anti wind up is implemented by simply limiting the integral term from growing larger than the maximum output value, which is known. The final output of the controller is also limited between the minimum and maximum output possible.
7 Experimental Equipment

Measurements and tests have been carried out in a truck with a six cylinder inline turbo charged diesel engine, see Picture 2. The truck has primarily been used for continuous tests during the development of the automated controller tuning procedure, but also for verification and evaluation of the method and different controllers. Technical data for the truck engine are shown Table 3.

Measurements and tests have also been carried out on a generator set consisting of a V8 turbo charged diesel engine of type DC1643A and a generator, see Picture 3. The GenSet has been used for verification of the method and evaluation of different controllers. Technical data for the GenSet engine are shown in Table 3.

<table>
<thead>
<tr>
<th>Engine type</th>
<th>Truck engine</th>
<th>GenSet Engine</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of cylinders</td>
<td>6</td>
<td>8</td>
</tr>
<tr>
<td>Cylinder configuration</td>
<td>Inline</td>
<td>90° V</td>
</tr>
<tr>
<td>Maximum power at nominal speed</td>
<td>294 kW/400 hp at 1500rpm</td>
<td>400 kW/544 hp at 1500 rpm</td>
</tr>
<tr>
<td>Maximum torque at nominal speed</td>
<td>1872 Nm at 1500 rpm</td>
<td>2540 Nm at 1500 rpm</td>
</tr>
</tbody>
</table>

Table 3. Technical data for the engines used for measurements.

The generator can be loaded with resistive load to a maximum power of 500 kW. The load can be applied in steps of minimum 25kW an maximum 500kW using a switch board. The load resistors are cooled using a fan. The load changes occur as step increase or decrease of the load. No ramping is possible. Step changes correspond well to the load pattern observed in electrical grids where different loads are connected and disconnected. Technical data for the engine are shown in Table 3. Technical data for the generator are shown in Table 4. Different controllers have been evaluated using the GenSet.
Picture 3. The generator set with the 16-litre V-8 to the right and the generator to the left. Two connection boxes are seen in the foreground of the picture.

<table>
<thead>
<tr>
<th>Generator</th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>Brand</td>
<td>Newage Stamford</td>
</tr>
<tr>
<td>Output voltage</td>
<td>380 V 3-phase</td>
</tr>
<tr>
<td>Maximum output current</td>
<td>900 A</td>
</tr>
<tr>
<td>Maximum output power</td>
<td>504 kW</td>
</tr>
</tbody>
</table>

Table 4. Technical data for the generator.
8 Measurements and Results

Measurements have been carried out using the truck and GenSet described in chapter 7. There have been two main purposes of this. First data have been collected for modeling purposes. Then different controllers based on this model have been evaluated. In this chapter these measurements and tests will be accounted for. Measurement results will be presented and analyzed. The results will be commented.

8.1 Truck Measurements

In chapter 7 the test truck is described. It has been used to test the automatic tuning procedure described in chapter 6. Six different controllers have also been tested and evaluated for the servo case. Relevant measurements for the servo case, see section 5.5.4, for a GenSet can be carried out using a truck. The regulator case, see section 5.5.4, can however not be simulated in a truck in a satisfactory way. When it was seen that the automatic tuning procedure was working, efforts were redirected towards a real generator set.

8.1.1 Modeling

The model structure described in 5.4.1 can not entirely describe the real engine. The primary cause for this is that the engine is non-linear from injected fuel quantity to engine speed. The nonlinearity is illustrated by the measurement displayed in Figure 12. This suggests that gain scheduling should be used. This means that a static non-linearity is compensate in the control design, for example by varying the control parameters depending on the interval in which the engine is operating. Since a single speed engine is supposed to operate at a constant engine speed with small variations only, the engine is here assumed to be linear in an interval chosen with guidance of Figure 12. In Figure 12 it can be seen that for higher engine speeds, the nonlinearity is not as strong. Therefore step response experiments have been carried out in this region for the linear assumption to hold. Since the engine is supposed to operate at speeds around 1500rpm, a step response covering 1300 to 1600 rpm is sufficient. Lower engine speeds are not interesting in this case.

![Figure 12. The relationship between injected fuel quantity per stroke and engine speed for the test truck engine.](image)

The non-linearity in Figure 12 is not completely static. The engine friction varies with for example engine temperature. The shape of the non-linearity is likely to be the same for different temperatures, but for higher temperatures the engine friction will be lower and thus also the required fuel quantity. These variations are small and therefore they have been neglected here.
Another non-linear phenomenon occurs in transients. When a sudden increase in demanded load is made it takes a few seconds for the boost pressure to build up. This is known as turbo lag. This causes saturation in the injected fuel amount. For the truck engine no test with step load has been made and therefore turbo lag will not occur in this case. For the GenSet measurements, turbo lag will be present. This will be described in chapter 8.2.

During the experiments performed in the truck, the engine temperature was between 60°C and 70°C. As a result of this, the engine fan was running causing a higher resistance for the engine. Thus the needed fuel quantity is comparably high. The fan requires approximately 10-15 mg/st of fuel. However, the fan is part of the engine and should be included in the model. The fan was still running when controllers based on the step response experiment were tested, see section 8.1.2.

In Figure 13, the engine speed from a step response experiment is shown in red and the model output is shown in blue. The model parameters have been determined to $K_p=45.3$, $T=2.8$ and $L=0.05$ according to the method described in section 5.4.1. This has been done online using the implementation described in section 6.1.

Figure 13 does not show perfect matching of the two curves, which would of course be desirable, but it shows that the model roughly describes the engine dynamics in the interesting interval of 1300 to 1600 rpm. At approximately $t=67s$ the red curve in Figure 13 suddenly starts to increase slower causing it to differ from the blue curve. This phenomenon is hard to explain, but it might be caused by the cooling fan speeding up or a sudden increase of the engine friction. This might be caused by a drop in oil pressure or that the air compressor for the brakes is started.

The engine speed signal is very noisy which is natural for an unloaded diesel engine. The torque applied to the crankshaft from the combustion in the cylinders is not continuous. During one engine revolution, three cylinders will fire giving rise to three torque pulses. At the end of the crankshaft, a inertia in the form of a flywheel is placed to even out these variations. The flywheel is however quite light, so a smooth signal is still not obtained. In section 8.2.1, the same measurement carried out in a GenSet will be accounted for. There it will be shown that different characteristics are obtained when adding a larger inertia in the form of a generator.
8.1.2 Control

Six different controllers have been tested and evaluated. These controllers are based on the model parameters $K_p=45$, $T=2.78$ and $L=0.05$. To be able to evaluate and compare the different controllers, a test procedure has been developed. This is described below.

The aim of the test procedure is to describe control performance using numbers and compare it to the requirements stated in section 3.3. The procedure holds 3 points:

1. When the engine is running at its nominal speed, a set point increase of 2% is made. The engine speed overshoot $M_o$ and settling time $T_s$ is registered.
2. When the engine is running at its nominal speed, a set point decrease of 2% is made. The engine speed undershoot $M_u$ and settling time $T_s$ is registered.
3. When the engine is running at its nominal speed, the control error is registered for 10s or 500 samples. Then the mean relative error $E_{RM}$ is calculated according to Equation 35.

\begin{equation}
E_{RM} = \frac{\sum_{i=1}^{500} |e(k)|}{500 \cdot n_{ms,min}}
\end{equation}

The results of these tests will describe the control performance, but the engine’s behavior must also be analyzed when the engine is running. When testing a controller, important aspects are the engine’s acoustic noise and how smooth the engine is running. This will also be taken into account. Six different controllers have been evaluated. The results are shown in the coming sections. The control performance in terms of stability and response to set point changes is evaluated.

From the measurements accounted for in Table 5 to Table 7 in the coming sections, it is easy to draw the conclusion that none of the controllers provide acceptable performance. Studying plots of the measurements result, it is easy to see that the requirements of section 3.3 are not met. This is mainly a cause of too aggressive control parameters. The controllers tested are intended to work for single speed engines coupled to an aggregate, for example a generator. In this case, the engine has been completely unloaded. However, it can also be seen that the automatic tuning procedure is working and if it is desirable to tune controllers for an unloaded engine, the implemented tuning rules can be modified to produce slower controllers. This would give slower response but better stability.

Ziegler-Nichols Design

Controllers tuned using Ziegler-Nichols method, described in section 5.6.1, have been evaluated. One PI-controller and one PID-controller based on the model parameters $K_p=45$, $T=2.78$ and $L=0.05$ have been tested. Table 5 shows the control parameters and the performance for the two Ziegler-Nichols controllers. The tests and the notation in Table 5 are described in section 8.1.2.

<table>
<thead>
<tr>
<th>Controller</th>
<th>Ziegler-Nichols PI</th>
<th>Ziegler-Nichols PID</th>
</tr>
</thead>
<tbody>
<tr>
<td>Control parameters</td>
<td></td>
<td></td>
</tr>
<tr>
<td>P</td>
<td>1.12</td>
<td>1.49</td>
</tr>
<tr>
<td>Ti</td>
<td>0.15</td>
<td>0.12</td>
</tr>
<tr>
<td>Td</td>
<td>-</td>
<td>0.025</td>
</tr>
<tr>
<td>Test Results</td>
<td></td>
<td></td>
</tr>
<tr>
<td>$M_o$ [%]</td>
<td>47</td>
<td>37</td>
</tr>
<tr>
<td>$T_s$ [s]</td>
<td>0.32</td>
<td>0.36</td>
</tr>
<tr>
<td>$M_u$ [%]</td>
<td>43</td>
<td>40</td>
</tr>
<tr>
<td>$T_s$ [s]</td>
<td>0.3</td>
<td>0.3</td>
</tr>
<tr>
<td>$E_{RM}$ [%]</td>
<td>0.11</td>
<td>0.10</td>
</tr>
</tbody>
</table>

Table 5. Control parameters and test results for the Ziegler-Nichols controllers.

The two Ziegler-Nichols controllers show similar performance as shown in Table 5. Neither of them provides acceptable stability properties. A relative mean error of 0.10% is acceptable if related to the requirements given
in section 3.3, but the maximum error is over 4 rpm. The engine does not run smoothly and the engine acoustic noise is disturbing. The control activity is higher when using the PID-controller. This is probably a cause of the derivative action. The PI controller works better since the engine speed signal is so noisy in this case. The measurement from which the values of Table 5 are taken is shown in Figure 14. Here it can be seen that the engine speed is not stable enough at 1500 rpm for the PI-controller.

![Figure 14. Engine speed (top) and injected fuel quantity (bottom) for the Ziegler-Nichols PI-controller.](image)

Figure 14 shows a bode plot of the closed loop system when using the Ziegler-Nichols PI-controller. The closed loop system is stable and has a bandwidth of 7.86 rad/s. Figure 16 shows the sensitivity function when using the same controller. The magnitude of the sensitivity function is small for low frequencies and thus the system is insensitive to model error in accordance with what was said in section 5.3.1. In this thesis, no pre-compensator is used and therefore the complementary sensitivity function is equal to the transfer function of the closed-loop system as stated in section 5.3.2. In Figure 15, it is seen that the magnitude of the complementary sensitivity function is small for high frequencies. Thus the system is robust according to section 5.3.2.

Plot of measurement results, bode plots for the closed loop system and sensitivity function when using the Ziegler-Nichols PID controller are very similar to Figure 14 to Figure 16 and therefore they will not be displayed here. Instead they can be found in Appendix A, Plots and Figures.
Automatic Tuning of Control Parameters for Single Speed Engines

Figure 15. Bode plot for the closed loop system when using the Ziegler-Nichols PI-controller.

Figure 16. Bode plot of the sensitivity function when using the Ziegler-Nichols PI-controller.
Cohen-Coon Design

Controllers tuned using Cohen-Coons method described in section 5.6.1 have been evaluated. One PI-controller and one PID-controller based on the model parameters $K_p=45$, $T=2.78$ and $L=0.05$ have been tested. The control parameters and test results are given in Table 6.

The Cohen-Coon method originally gave too large control parameters. The engine speed could not be stabilized. Therefore the proportional gain was reduced by 40% giving a 40% decrease of the proportional, integral and derivative part of the controller. All three parts of the controllers were affected by this change since the controller was implemented as shown in Equation 32, section 6.2.

These controllers behave in a very similar way to the Ziegler-Nichols controllers, but the performance is slightly better. The addition of the derivative term does not improve the performance. The PI-controller works better than the PID. Although better than with the Ziegler-Nichols controller, the engine is not running smoothly for neither PI nor PID-control and the acoustic noise is disturbing.

Please refer to Appendix A, Plots and Figures for figures of measurement results and bode plots of the closed-loop systems, sensitivity functions and complementary sensitivity functions.

<table>
<thead>
<tr>
<th>Controller Parameters</th>
<th>Control Parameters</th>
<th>Test Results</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Cohen-Coon PI</td>
<td>Cohen-Coon PID</td>
</tr>
<tr>
<td>P</td>
<td>1.14</td>
<td>1.01</td>
</tr>
<tr>
<td>Ti</td>
<td>0.16</td>
<td>0.12</td>
</tr>
<tr>
<td>Td</td>
<td>-</td>
<td>0.018</td>
</tr>
<tr>
<td>1</td>
<td>$M_o$ [%] 40</td>
<td>$M_o$ [%] 40</td>
</tr>
<tr>
<td></td>
<td>$T_s$ [s] 0.34</td>
<td>$T_s$ [s] 0.5</td>
</tr>
<tr>
<td>2</td>
<td>$M_o$ [%] 40</td>
<td>$M_o$ [%] 27</td>
</tr>
<tr>
<td></td>
<td>$T_s$ [s] 0.36</td>
<td>$T_s$ [s] 0.32</td>
</tr>
<tr>
<td>3</td>
<td>$E_{RM}$ [%] 0.10</td>
<td>$E_{RM}$ [%] 0.08</td>
</tr>
</tbody>
</table>

Table 6. Control parameters and test results for the Cohen-Coon controllers.

Pole-placement Design

Two PI-controller based on the model parameters $K_p=45$ and $T=2.78$ and tuned using the method described in section 5.6.2 have been evaluated. One controller based on the closed-loop parameters $\omega=13.4$ rad/s and $\zeta=0.85$, and another based on $\omega=22$ rad/s and $\zeta=0.95$. The control parameters and test results for the two controllers are shown in Table 7.

The first set of parameters are based on the discussion in section 5.6.2 where the model parameters $K_p=75$ and $T=3.3$ were used for a six cylinder engine similar to the one used here. For this test truck engine these parameters are determined to $K_p=45$ and $T=2.78$. The large difference in $K_p$ results in a larger proportional gain for the engine used here. This results in a controller which is slightly faster. Using the control parameters used by Scania today ($P=1.0$ and $T=0.125$), which works well for this engine, with the model parameters $K_p=45$ and $T=2.78$ gives $\omega=11.4$ rad/s and $\zeta=0.73$. Here Equation 31 in section 5.6.2 has been used. Thus the closed-loop parameters used for the pole placement design are slightly larger than what they should be. This gives a too fast controller in this case.
Automatic Tuning of Control Parameters for Single Speed Engines

Table 7. Control parameters and test results for the two pole-placed controllers.

<table>
<thead>
<tr>
<th>Controller</th>
<th>Pole-placed PI 1</th>
<th>Pole-placed PI 2</th>
</tr>
</thead>
<tbody>
<tr>
<td>Control parameters</td>
<td>P 1.39 Ti 0.13</td>
<td>P 2.56 Ti 0.086</td>
</tr>
<tr>
<td>Mo [%]</td>
<td>60</td>
<td>107</td>
</tr>
<tr>
<td>Ms [%]</td>
<td>50</td>
<td>70</td>
</tr>
<tr>
<td>ERM [%]</td>
<td>0.17</td>
<td>0.34</td>
</tr>
<tr>
<td>Test Results</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 M [s]</td>
<td>0.22</td>
<td>0.28</td>
</tr>
<tr>
<td>2 M [s]</td>
<td>0.26</td>
<td>0.42</td>
</tr>
<tr>
<td>3 ERM [%]</td>
<td>0.17</td>
<td>0.34</td>
</tr>
</tbody>
</table>

The second controller based on the closed-loop parameters $\omega=22$ rad/s and $\zeta=0.95$ is designed to be faster and to provide stronger damping of load disturbances than the first controller (Observe that no test for the regulator case has been performed for the truck engine. This will be tested for the GenSet). It is evident that the second set of control parameters are too aggressive causing severe stability problems as shown in Figure 17.

Figure 17. Engine speed (top) and injected fuel quantity (bottom) for the pole-placed PI-controller with $\omega=22$ rad/s $\zeta=0.95$.

Please refer to Appendix A, Plots and Figures for figures of measurement results and bode plots of the closed-loop systems, sensitivity functions and complementary sensitivity functions.
8.2 GenSet Measurements

In section 7 the GenSet used for measurements is described. It has been used to test the automatic tuning procedure described in chapter 6. Six different controllers have also been tested and evaluated for the servo case as well as the regulator case, see section 5.1.

8.2.1 Modeling

As said in section 8.1.1, the truck engine is non-linear from injected fuel quantity to engine speed. So is the GenSet engine. The assumptions applied to the truck engine were described in section 8.1.1. The engine was assumed to be linear in the interval over which the step was applied. The non-linearity was also assumed to be static. These assumptions are applied to the GenSet engine as well.

The required fuel quantity is measured at different engine speeds. The result is shown in Figure 18 which is used to determine a suitable interval in which to perform the step response experiment.

![Figure 18. The relationship between injected fuel quantity per stroke and engine speed for the GenSet engine.](image)

The GenSet engine requires lower or equal fuel quantities compared to the truck engine which can be seen if comparing Figure 18 to Figure 12. This is not what would be expected since the GenSet engine has eight cylinders compared to the truck engine’s six cylinders. The engine friction should be higher for eight cylinders. In section 8.1.1, it was said that the measurements for the truck engine were carried out when the engine cooling fan was running. The GenSet engine cooling fan was not running during any of the GenSet measurements. This explains the difference (or rather the absence of difference) between the two engines regarding required fuel quantity. The cooling fan did not run during the controller evaluation for the GenSet. Therefore the same system that was modeled was also controlled.

Based on Figure 18, a step between 1400 and 1770 rpm is chosen to obtain a model describing the engine well in the interval. Model output and measured engine speed are shown in Figure 19. The model parameters have been calculated using the method described in section 5.4.1. The result is \( K_p = 60 \), \( T = 8.3 \) and \( L = 0.06 \).

Limitations in the turbo charger will cause saturation of the control signal when step load is applied. This will be shown in section 8.2.2.
Figure 19. Model with $K_p=60.1$, $T=8$ and $L=0.06$. Model output shown in blue and real engine speed in red.

The engine speed signal is not as noisy as was seen in section 8.1.1. The reason for this is that a large inertia in the form of the generator has been added to the system. This inertia acts as a low pass filter keeping the engine speed signal smooth. Another factor influencing engine speed signal noise is the number of cylinders. The GenSet engine has eight cylinders giving four firings per engine revolution compared to the truck engines three. More firing per revolution will give smaller engine speed variations. This is a result of the fact that the torque generated by the combustion in the cylinders is applied to the crankshaft more continuously. Since the GenSet engine runs smoother, it will also be easier to obtain good stability when controlling the engine speed. This is shown by the measurement results.

8.2.2 Control

Six different controllers have been tested and evaluated. These controllers are based on the model parameters $K_p=60$, $T=8.3$ and $L=0.06$. All six controllers are based on the same theory as the ones used for the truck measurements accounted for in section 8.1.2. This theory can be found in section 5.6. Only the model parameters used to derive the controllers are different.

To be able to evaluate and compare the different controllers, the test procedure described in section 8.1.2 has been used. The test procedure has been extended with two additional points to evaluate how different controllers handle load disturbances. The two points are:

4. When the engine is running at its nominal speed, the load is increased from 0 kW to 240 kW corresponding to a load increase of 60% for the engine in question. The engine speed undershoot $M_u$ and settling time $T_s$ is registered.

5. When the engine is running at its nominal speed, the load is decreased from 240 kW to 0 kW corresponding to a load decrease from 60% to 0% for the engine in question. The engine speed overshoot $M_o$ and settling time $T_s$ is registered.

Numerous figures of measurements and bode plots have been produced. A selection of these will be shown and discussed here. The rest can be found in Appendix A, Plots and Figures.
Ziegler-Nichols Design

One PI- and one PID-controller derived from the model parameters $K_p=60$, $T=8.3$ and $L=0.06$ using Ziegler-Nichols method described in section 5.6.1 have been evaluated. The control parameters and test results are shown in Table 8.

The two controllers show similar properties for the servo case. Both controllers fulfill the requirement of less than 0.2% control error when the reference speed is held constant at 1500 rpm, see section 3.3. Note that the stability test (test 3) gives significantly lower relative mean error than what was achieved in for the truck engine. This is a result of the generator and its low pass characteristics.

<table>
<thead>
<tr>
<th>Controller</th>
<th>Ziegler-Nichols PI</th>
<th>Ziegler-Nichols PID</th>
</tr>
</thead>
<tbody>
<tr>
<td>Control parameters</td>
<td></td>
<td></td>
</tr>
<tr>
<td>P</td>
<td>2.08</td>
<td>2.77</td>
</tr>
<tr>
<td>Ti</td>
<td>0.18</td>
<td>0.14</td>
</tr>
<tr>
<td>Td</td>
<td>-</td>
<td>0.03</td>
</tr>
<tr>
<td>Test Results</td>
<td></td>
<td></td>
</tr>
<tr>
<td>1 $M_o [%]$</td>
<td>57</td>
<td>57</td>
</tr>
<tr>
<td>$T_o [s]$</td>
<td>0.42</td>
<td>0.46</td>
</tr>
<tr>
<td>2 $M_o [%]$</td>
<td>63</td>
<td>50</td>
</tr>
<tr>
<td>$T_o [s]$</td>
<td>0.50</td>
<td>0.46</td>
</tr>
<tr>
<td>3 $E_{RM} [%]$</td>
<td>0.033</td>
<td>0.052</td>
</tr>
<tr>
<td>4 $M_o [%]$</td>
<td>7.3</td>
<td>6.5</td>
</tr>
<tr>
<td>$T_o [s]$</td>
<td>2.5</td>
<td>2.0</td>
</tr>
<tr>
<td>5 $M_o [%]$</td>
<td>3.5</td>
<td>2.3</td>
</tr>
<tr>
<td>$T_o [s]$</td>
<td>0.52</td>
<td>0.44</td>
</tr>
</tbody>
</table>

Table 8. Control parameters and test results for the Ziegler-Nichols controllers.

For the regulator case, the PID-controller works better. This is a result of the slightly faster control parameters used for the PID-controller. None of the controllers fulfill the requirement of less than 3% undershoot and 2s settling time when the 60% load step is applied see section 3.3. The PID-controller does however meet the requirement of less than 2.5% overshoot and 1s settling time when the load is disconnected.

Please refer to Appendix A, Plots and Figures for plots of measurements and bode plots of the closed-loop system, sensitivity function and transfer function from load disturbance to output.
Cohen-Coon Design

One PI- and one PID-controller derived from the model parameters $K_p=60$, $T=8.3$ and $L=0.06$ using Cohen-Coons method described in section 5.6.1 have been evaluated. The control parameters and test results are shown in Table 9. As mentioned in section 8.1.2 the proportional gain is reduced by 40% giving a 40% decrease in the proportional, integral and derivative part of the controllers.

Both controllers meet the stability requirements, see section 3.3. The PID-controller provides better set point response while the PI-controller handles load disturbances better. None of the controllers meet the load disturbance requirements.

<table>
<thead>
<tr>
<th>Controller</th>
<th>Cohen-Coon PI</th>
<th>Cohen-Coon PID</th>
</tr>
</thead>
<tbody>
<tr>
<td>Control parameters</td>
<td></td>
<td></td>
</tr>
<tr>
<td>P</td>
<td>2.09</td>
<td>1.87</td>
</tr>
<tr>
<td>Ti</td>
<td>0.19</td>
<td>0.15</td>
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<tr>
<td>Td</td>
<td>-</td>
<td>0.02</td>
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<tr>
<td>Test Results</td>
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</tr>
<tr>
<td>1</td>
<td>$M_o %$ 57</td>
<td>$M_o %$ 25</td>
</tr>
<tr>
<td></td>
<td>$T_s [s]$ 0.56</td>
<td>$T_s [s]$ 0.64</td>
</tr>
<tr>
<td>2</td>
<td>$M_o %$ 40</td>
<td>$M_o %$ 24</td>
</tr>
<tr>
<td></td>
<td>$T_s [s]$ 0.52</td>
<td>$T_s [s]$ 0.54</td>
</tr>
<tr>
<td>3</td>
<td>$E_{RM %}$ 0.022</td>
<td>$E_{RM %}$ 0.041</td>
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<tr>
<td>4</td>
<td>$M_o %$ 6.4</td>
<td>$M_o %$ 7.0</td>
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<tr>
<td></td>
<td>$T_s [s]$ 2.1</td>
<td>$T_s [s]$ 2.2</td>
</tr>
<tr>
<td>5</td>
<td>$M_o %$ 3.5</td>
<td>$M_o %$ 3.1</td>
</tr>
<tr>
<td></td>
<td>$T_s [s]$ 0.60</td>
<td>$T_s [s]$ 0.44</td>
</tr>
</tbody>
</table>

Table 9. Control parameters and test results for the Cohen-Coon controllers.

Please refer to Appendix A, Plots and Figures for plots of measurements and bode plots of the closed-loop system, sensitivity function and transfer function from load disturbance to output.
Pole-placement Design

Two PI-controllers derived using the pole-placement method described in section 5.6.2 have been evaluated. The model parameters $K_p=60$, $T=8.3$ and $L=0.06$ were used to derive the controllers.

One of the controllers is based on the closed-loop parameters $\omega=13.4$ rad/s and $\xi=0.85$, and another based on $\omega=22$ rad/s and $\xi=0.95$ giving the poles shown in Figure 20. Control parameters and test results are shown in Table 10. The second controller was designed to be faster and to provide stronger damping of load disturbances which can be seen in Figure 22 where a bode plot of the transfer function from load disturbance to output is shown. Here it can be seen that load disturbances are strongly damped for all frequencies.

<table>
<thead>
<tr>
<th>Controller</th>
<th>Pole-placed PI 1</th>
<th>Pole-placed PI 2</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Control parameters</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>P</td>
<td>3.14</td>
<td>P</td>
</tr>
<tr>
<td>Ti</td>
<td>0.13</td>
<td>Ti</td>
</tr>
<tr>
<td>Td</td>
<td>-</td>
<td>Td</td>
</tr>
<tr>
<td><strong>Test Results</strong></td>
<td></td>
<td></td>
</tr>
<tr>
<td>1</td>
<td>$M_o [%]$</td>
<td>51</td>
</tr>
<tr>
<td></td>
<td>$T_s [s]$</td>
<td>0.34</td>
</tr>
<tr>
<td>2</td>
<td>$M_o [%]$</td>
<td>33</td>
</tr>
<tr>
<td></td>
<td>$T_s [s]$</td>
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</tr>
<tr>
<td>3</td>
<td>$E_{RM} [%]$</td>
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<tr>
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<td>6.5</td>
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<td>$T_s [s]$</td>
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<td>5</td>
<td>$M_o [%]$</td>
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</tr>
<tr>
<td></td>
<td>$T_s [s]$</td>
<td>0.42</td>
</tr>
</tbody>
</table>

Table 10. Control parameters and test results for the PI-controllers obtained by pole-placement.

Both controllers meet the stability requirements, see section 3.3, but the PI-controller provides the best stability of all six controllers, see Figure 21. The set point response does however have larger overshoot and undershoot than the Cohen-Coon PID-controller. The regulator case is handled very well by the faster PI-controller. The requirements for the undershoot when the load is disconnected are met but they are not met when the load is applied. This is not primarily a control problem.
Figure 21. Engine speed (top) and injected fuel quantity (bottom) for the PI-controller obtained by pole-placement with $\omega=13.4$ rad/s and $\zeta=0.85$.

In Figure 22, it can be seen that the control signal is saturating when the load step is applied. The controller commands a high fuel quantity from the injection system, but after a short period of very rapid control signal increase, the control signal starts to increase slower. This phenomenon has physical causes. A diesel engine runs with an excess of air at steady state. When the controller demands a high fuel quantity, there is enough air for a rather large change in fuel quantity. If the change is too rapid, and too large, the air entering the engine will not be sufficient for a complete combustion. This will result in black smoke leaving the exhaust pipe. This is unacceptable, both from a customer’s and an environmental perspective. Therefore the Engine Control Unit limits the injected fuel quantity to match the available amount of air and prevent black smoke. During this
procedure, the turbocharger speeds up and the amount of air entering the engine is slowly increased. Thus the Engine Control Unit can allow higher and higher fuel quantity. In Figure 23, the engine speed, injected fuel quantity and boost pressure are displayed. The phenomenon described above can be seen as the ramp at the top of the fuel curve when the load step is applied. Note how the fuel quantity increases with the boost pressure. This saturation can cause integrator windup in the controller, but this is not a problem since the integrator sum is limited, see section 6.2. The turbo lag causes the asymmetry in the engine speed in Figure 23. The engine speed undershoot occurring when step load is applied is much larger than the overshoot when the load is removed.

![Fuel Injection and Engine Speed](image)

**Figure 23.** Engine speed (top), injected fuel quantity (middle) and boost pressure (bottom) for the PI-controller obtained by pole-placement with $\omega=22$ rad/s and $\zeta=0.95$.

Please refer to Appendix A, Plots and Figures for plots of measurements and bode plots of the closed-loop system, sensitivity function and transfer function from load disturbance to output.
9 Discussion and Conclusions

Studying the results, one can draw a number of conclusions.

It is possible to determine controller parameters for a single speed engine automatically. Using the method developed in this work gives controller parameters that can control the engine speed. This has been shown for two different types of engines and installations. One unloaded six cylinder inline diesel engine and one V8 diesel engine operating in a generator set. Good performance for the servo and regulator case has been shown for the V-8 GenSet engine.

The control problem for the unloaded truck engine and the GenSet is quite different. The speed of the truck engine is much more noisy than that of the GenSet. The reason for the stable engine speed in the GenSet is the generator. Its large moment of inertia acts as a low pass filter. This permits a faster set of control parameters compared to the truck. This is most useful when handling the regulator case. Since the engine speed signal is very noisy for the unloaded engine, a controller with derivative action is not appropriate. For the loaded GenSet engine, derivative action works better.

It has been shown that the pole placement design procedure gives the best results for the GenSet engine. Using a PI-controller based on poles with a frequency of 22 rad/s and a relative damping of 0.95 gives a maximum error of 95 rpm and a recovery time of 1.7 s when applying 60% load step. This is not enough to meet the requirements stated in 3.3 but it is certainly competitive compared to the present controller used by Scania, which gives a maximum error of 117 rpm and recovery time of 2.5 s for the same test.

Good set point tracking has been obtained for the GenSet engine using several different controllers. A PI-controller based on poles with a frequency of 13.4 rad/s and a relative damping of 0.85 gives the best result. Less than 0.1% maximum error when tracking a steady reference and fast response to set point changes implies that the controller fulfills the set point requirements in section 3.3.

The requirements stated in 3.3 are hard to meet when load step is applied. This is not primarily a problem related to the controller. Measurements show that the engine has physical characteristics limiting how fast it can respond. These limitations are first and foremost present in the turbocharger. When high output torque is demanded from the engine, a slight time delay occurs before the turbo speeds up and can provide full boost pressure. This is often described as turbo lag. The turbo characteristics can be modified, but transients will always be problematic. One way to solve the response problem could be to use a compressor instead.

The automatic tuning procedure developed in this work both has its strengths and weaknesses. A great advantage is the possibility to obtain a well tuned controller for every single installation. The method is simple and does not require much experience making it suitable for the aftermarket. A drawback is the fact that the model does not describe higher order dynamics if such should be present. Nonlinearities is another problem which can cause faulty parameter estimation if the step response experiment is carried out over the wrong interval.

The conclusions stated above are built on the results and experience gained during the completion of this work. To say that these conclusions are valid for all scenarios where a single speed engine might be used would be too a strong statement. To be able to argue that this is the case, much more tests and measurements would have to be made in different types of installation. This work has shown that automatic determination of controller parameters is possible in practice, not that it will always work.
10 Future Work

In this work controllers have been tuned and evaluated for operation at a nominal engine speed of 1500rpm. As stated in chapter 3, engines used for 60 Hz operation run at a nominal speed of 1800rpm. It is likely that the controllers tuned for 1500rpm operation will work for 1800rpm as well, but this should be investigated further. It should also be investigated how much a controller tuned for 1500rpm differs from a controller tuned for 1800rpm and how large the differences in engine dynamics are. Which engine speed is optimal from a control perspective? 1800rpm nominal speed might give better response since the turbo charger will have a higher speed due to the higher engine speed.

The method used in this work has been developed on a scientific basis. What is acceptable by customers have not been taken into a great deal of consideration. When the step response experiment is used, the engine speed varies in a large interval compared to normal operation. There might be speed restrictions for different aggregates. A generator will not be mechanically damaged by the experiment, but what about the electric equipment? Some kind of survey of how the single speed market works and what the customers are willing to do with their engines could be done.

The method developed in this thesis was originally planned to become a tool for the aftermarket provided that it turned out to work sufficiently well. The results show that the method can be used, but to be used by service personnel, some modifications and additions have to be made. First of all, some kind of PC-tool to start, supervise and control the step response test is needed. There is already Scania software used by the aftermarket for different kinds of service tools. Functionality for controller tuning could be added in one of these. Further more the calculated control parameters must be saved in the Engine Control Unit in some way. With the present implementation the control parameters are reset when the voltage to the Engine Control Unit is broken. There are memory areas that could be used for storage of parameters. This will have to be implemented if the method shall be used for other purposes than research and development.

The generator set has been modeled as a first order system with a time delay. A more advanced approach to modeling would be a physical model where the rotating masses in the engine and the generator as well as the flexibility in the coupling between them are taken into account. The engines non-linear characteristics could also be studied and taken into consideration when modeling the engine and designing controllers.

The first order model relies on the assumption that the coupling between generator and engine is stiff. It has been shown that this is a good approximation for coupling used in the GenSet where the measurements in this work have been carried out. As said in section 3.2 there exists other couplings which might have different characteristics. The dynamics of such couplings should be investigated to determine whether they are stiff enough for the assumption to hold.

Ziegler-Nichols and Cohen-Coons method are classical methods. They are simple and easy to use, but they are also old fashioned. Ziegler and Nichols present their method in 1942. Since then, many people around the world have been involved in control theory research and as a result, better methods exist today. One such method is the Åström-Hägglund method described in [2]. It possesses many of the advantages of Ziegler-Nichols and Cohen-Coons method in terms of simplicity and limited process knowledge, but provides better damping and stability margins for the closed-loop system. It would be interesting to evaluate other control design approaches, of which Åström-Hägglund tuning could be one. It might also be interesting to look into for example LQ or $H_\infty$ control.

It has been shown that the method works for two different engine types, but only two individuals have been tested. To verify that the method works for other installations more engines should be tested. When doing this, one should try to cover five, six and eight cylinder engines as well as different single speed applications. Not only GenSets but some of the other applications mentioned in chapter 3
References


Appendix A, Plots and Figures

This appendix contains plots from measurements using the different controllers discussed in chapter 8. Bode plots of the closed-loop systems, sensitivity functions and transfer functions from load disturbances to output can also be found here. The figures found in this Appendix are not unimportant, but to keep the thesis neat and easy to read, they have been excluded from the rest of the material.

**Truck Measurements**

**Ziegler-Nichols Controllers**

![Graphs showing engine speed and fuel quantity](image)

Figure 24. Engine speed (top) and injected fuel quantity (bottom) for the Ziegler-Nichols PID-controller.
Automatic Tuning of Control Parameters for Single Speed Engines

Figure 25. Bode plot for the closed loop system when using the Ziegler-Nichols PID-controller.

Figure 26. Bode plot of the sensitivity function when using the Ziegler-Nichols PID-controller.
Cohen-Coon Controllers

Figure 27. Engine speed (top) and injected fuel quantity (bottom) for the Cohen-Coon PI-controller.

Figure 28. Bode plot for the closed loop system when using the Cohen-Coon PI-controller.
Figure 29. Bode plot of the sensitivity function when using the Cohen-Coon PI-controller.

Figure 30. Engine speed (top) and injected fuel quantity (bottom) for the Cohen-Coon PID-controller.
Figure 31. Bode plot for the closed loop system when using the Cohen-Coon PID-controller.

Figure 32. Bode plot of the sensitivity function when using the Cohen-Coon PID-controller.
Pole-placement Controllers

Figure 33. Engine speed (top) and injected fuel quantity (bottom) for the pole-placed PI-controller with $\omega=13.4$ rad/s $\zeta=0.85$.

Figure 34. Bode plot for the closed loop system when using the pole-placed PI-controller with $\omega=13.4$ rad/s $\zeta=0.85$. 

Automatic Tuning of Control Parameters for Single Speed Engines
Figure 35. Bode plot of the sensitivity function when using the pole-placed PI-controller with $\omega=13.4$ rad/s $\zeta=0.85$.

Figure 36. Bode plot for the closed loop system when using the pole-placed PI-controller with $\omega=22$ rad/s $\zeta=0.95$. 
Figure 37. Bode plot of the sensitivity function when using the pole-placed PI-controller with $\omega=22$ rad/s $\zeta=0.95$. 
GenSet Measurements

Ziegler-Nichols Controllers

Figure 38. Engine speed (top) and injected fuel quantity (bottom) for the Ziegler-Nichols PI-controller.

Figure 39. Engine speed (top), injected fuel quantity (middle) and boost pressure (bottom) for the Ziegler-Nichols PI-controller.
Automatic Tuning of Control Parameters for Single Speed Engines

Figure 40. Bode plot for the closed loop system when using the Ziegler-Nichols PI-controller.

Figure 41. Bode plot of the sensitivity function when using the Ziegler-Nichols PI-controller.
Figure 42. Bode plot of the transfer function from load disturbances entering the process input to the process output for the Ziegler-Nichols PI-controller.

Figure 43. Engine speed (top) and injected fuel quantity (bottom) for the Ziegler-Nichols PID-controller.
Figure 44. Engine speed (top), injected fuel quantity (middle) and boost pressure (bottom) for the Ziegler-Nichols PID-controller.

Figure 45. Bode plot for the closed loop system when using the Ziegler-Nichols PID-controller.
Figure 46. Bode plot of the sensitivity function when using the Ziegler-Nichols PID-controller.

Figure 47. Bode plot of the transfer function from load disturbances entering the process input to the process output for the Ziegler-Nichols PID-controller.
Cohen-Coon Controllers

Figure 48. Engine speed (top) and injected fuel quantity (bottom) for the Cohen-Coon PI-controller.

Figure 49. Engine speed (top), injected fuel quantity (middle) and boost pressure (bottom) for the Cohen-Coon PI-controller.
Figure 50. Bode plot for the closed loop system when using the Cohen-Coon PI-controller.

Figure 51. Bode plot of the sensitivity function when using the Cohen-Coon PI-controller.
Figure 52. Bode plot of the transfer function from load disturbances entering the process input to the process output for the Cohen-Coon PI-controller.

Figure 53. Engine speed (top) and injected fuel quantity (bottom) for the Cohen-Coon PID-controller.
Figure 54. Engine speed (top), injected fuel quantity (middle) and boost pressure (bottom) for the Cohen-Coon PID-controller.

Figure 55. Bode plot for the closed loop system when using the Cohen-Coon PID-controller.
Automatic Tuning of Control Parameters for Single Speed Engines

Figure 56. Bode plot of the sensitivity function when using the Cohen-Coon PID-controller.

Figure 57. Bode plot of the transfer function from load disturbances entering the process input to the process output for the Cohen-Coon PID-controller.
Pole-placement Controllers

Figure 58. Engine speed (top), injected fuel quantity (middle) and boost pressure (bottom) for the PI-controller obtained by pole-placement with $\omega=13.4$ rad/s and $\zeta=0.85$.

Figure 59. Bode plot for the closed loop system when using the pole-placed PI-controller with $\omega=13.4$ rad/s $\zeta=0.85$. 
Figure 60. Bode plot of the sensitivity function when using the pole-placed PI-controller with $\omega=13.4 \text{ rad/s}$ $\zeta=0.85$.

Figure 61. Bode plot of the transfer function from load disturbances entering the process input to the process output for the PI-controller obtained by pole-placement with $\omega=13.4 \text{ rad/s}$ and $\zeta=0.85$. 
Figure 62. Engine speed (top) and injected fuel quantity (bottom) for the pole-placed PI-controller with $\omega = 22$ rad/s $\zeta = 0.95$.

Figure 63. Bode plot for the closed loop system when using the pole-placed PI-controller with $\omega = 22$ rad/s $\zeta = 0.95$. 
Figure 64. Bode plot of the sensitivity function when using the pole-placed PI-controller with $\omega=22$ rad/s $\zeta=0.95$. 