Estimation of Indicated– and Load–Torque from Engine Speed Variations

Master's thesis
performed in **Vehicular Systems**

by
Fredrik Bengtsson

Reg nr: LiTH-ISY-EX--06/3793--SE

June 21, 2006

Estimation of Indicated– and Load–Torque from Engine Speed Variations

Master's thesis

performed in **Vehicular Systems**,
Dept. of Electrical Engineering
at **Linköping University**

by **Fredrik Bengtsson**

Reg nr: LiTH-ISY-EX--06/3793--SE

Supervisors: **Per Andersson**
Linköping University/
GM Powertrain Sweden
Martin Gunnarsson
Linköping University
Richard Backman
GM Powertrain Sweden

Examiner: **Associate Professor Lars Eriksson**
Linköping University

Linköping, June 21, 2006

Avdelning, Institution Division, Department	Vehicular Systems, Dept. of Electrical Engineering 581 83 Linköping	**Datum** Date: June 21, 2006

Språk / Language
- ☐ Svenska/Swedish
- ☒ Engelska/English
- ☐ ______

Rapporttyp / Report category
- ☐ Licentiatavhandling
- ☒ Examensarbete
- ☐ C-uppsats
- ☐ D-uppsats
- ☐ Övrig rapport
- ☐ ______

ISBN
—

ISRN
LiTH-ISY-EX--06/3793--SE

Serietitel och serienummer / Title of series, numbering
ISSN
—

URL för elektronisk version
http://www.vehicular.isy.liu.se
http://www.ep.liu.se/exjobb/isy/06/3793/

Titel Skattning av indikerat– och last– moment från vevaxels varvtalsvariationer

Title Estimation of Indicated– and Load– Torque from Engine Speed Variations

Författare / Author: Fredrik Bengtsson

Sammanfattning
Abstract

The importance of control systems and diagnostics in vehicles are increasing and has resulted in several new methods to calculate better control signals. The performance can be increased by calculating these signals close to optimum, but that also require more and precise information regarding the system.

One of the wanted control signals are the crankshaft torque and the thesis presents two different methods to estimate this torque using engine speed variations. These methods are *Modeling of the Crankshaft* and *Frequency Analysis*. The methods are evaluated and implemented on for a four cylinder SAAB engine. Measurements are made in an engine test cell as well as a vehicle.

The results show that the *Modeling of the Crankshaft* method does not produce a satisfying estimation, with a difference of about 200% between estimated and calculated torque. On the other hand, the *Frequency Analysis* provides an accurate estimation of both mean and instantaneous indicated torque, with a maximum difference of ±20% between estimated and calculated torque.

Nyckelord / Keywords: Torque estimation, Indicated torque, Load torque, Crankshaft model, Frequency analysis

Abstract

The importance of control systems and diagnostics in vehicles are increasing and has resulted in several new methods to calculate better control signals. The performance can be increased by calculating these signals close to optimum, but that also require more and precise information regarding the system.

One of the wanted control signals are the crankshaft torque and the thesis presents two different methods to estimate this torque using engine speed variations. These methods are *Modeling of the Crankshaft* and *Frequency Analysis*. The methods are evaluated and implemented on for a four cylinder SAAB engine. Measurements are made in an engine test cell as well as a vehicle.

The results show that the *Modeling of the Crankshaft* method does not produce a satisfying estimation, with a difference of about 200% between estimated and calculated torque. On the other hand, the *Frequency Analysis* provides an accurate estimation of both mean and instantaneous indicated torque, with a maximum difference of $\pm 20\%$ between estimated and calculated torque.

Keywords: Torque estimation, Indicated torque, Load torque, Crankshaft model, Frequency analysis

Acknowledgment

I would like to thank my supervisors Per Andersson, Martin Gunnarsson and Richard Backman for their valuable support during the work with my Master Thesis. I would also like to thank my examiner, Associate Professor Lars Eriksson, for his valuable suggestions and point of views.

Also all employees at the division of Vehicular Systems, Dept. of Electrical Engineering, at Linköping University shall have great thanks for always being helpful.

I would also like to thank Stefan Gustafsson at GM Powertrain Sweden for helping with measurements in the vehicle and retrieving engine parameters.

Linköping, June 2006
Fredrik Bengtsson

Contents

Chapter 1

Introduction

This master thesis is carried out for GM Powertrain Sweden in Södertälje in collaboration with the division of Vehicular Systems at Linköping University.

1.1 Background

The demands of vehicles in the aspect of comfort, performance and fuel consumption is always increasing by consumers. At the same time demands on low emissions is regulated by governments. To keep up with these demands and regulations it is essential to create new methods to calculate better control signals to the engine e.g. when to change gear in an automatic gearbox or the amount of fuel injected. The performance can be increased by calculate these signals close to optimum, but that also require more and precise information regarding the system.

The crankshaft torque is one parameter that is of interest for that cause, e.g. by avoiding sound when changing gear and to compensate for increased load created by the air conditioning. This torque can be measured by applying torque sensors at the crankshaft [11]. In the automobile industry it is also necessary to keep the cost low to survive and it is desirable to retrieve an estimation of the torque from already available signals without needing to apply additional expensive sensors. This makes it desirable to use the engine speed signal witch is measured with a relatively accurate and inexpensive sensor. Fortunately, the engine speed vary due to the torque created by the combustion and rotating masses. Hence it is hopefully possible to estimate the torque by examine the variations of the engine speed signal without expensive sensors.

1.2 Purpose

The purpose of the thesis is to investigate and evaluate methods in estimating the torques affecting the crankshaft on the basis of variations of the engine speed. If a method is found that satisfies the requirements on performance and computational efforts, the estimated torque will be used as an input in future control systems.

1.3 Goal

Different methods of estimating the torques affecting the crankshaft will be investigated and evaluated. To examine the performance of the methods they will be implemented and tested with measurements from an engine test bed. The performance will also be tested with measurements from a vehicle.

1.4 Method and Outline

Methods of estimating the torque are described, and two of them are evaluated and tested to find an appropriate for the purpose. The methods and the selection are described in chapter 4.The methods are used to estimate both mean (over a cycle) and instantaneous (at every even crankshaft angle) indicated torque. Also the mean load torque is evaluated.

1.5 Test Setup

All measurements are performed on a four cylinder SAAB L850 engine. For the measurements in the engine test bed, the engine speed is measured with an Hewlett Packard measuring system and a Leine & Linde sensor is used. On the basis of the data, the methods of estimating the engine torque are implemented and tested in MATLAB.

Chapter 2

System Description

The used engine is of four stroke with a cycle as two crankshaft revolutions. In its four strokes chemical energy in the fuel is first transformed into an oscillating motion on the piston due to the pressure in the cylinder created from the combustion. This creates a torque and a rotational motion on the crankshaft which depends on e.g. load, pressure in the cylinder, mass of all parts in motion and the geometry of the engine. The notation of the relevant parts of the engine is described in figure 2.1.

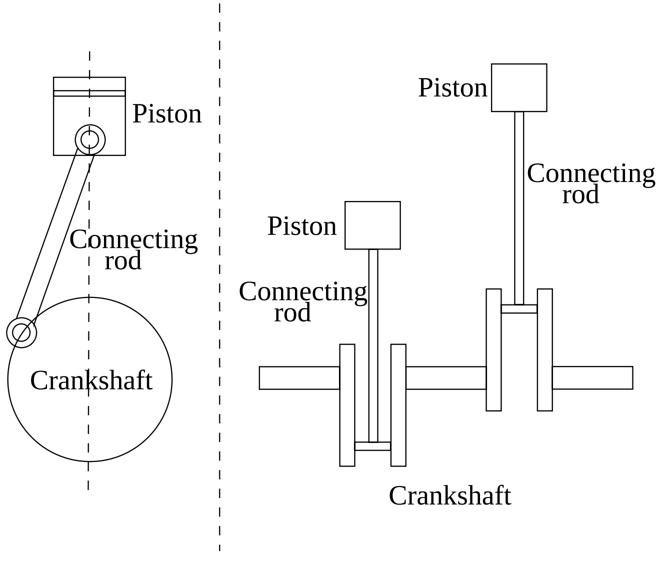


Figure 2.1: Description of the crankshaft and piston.

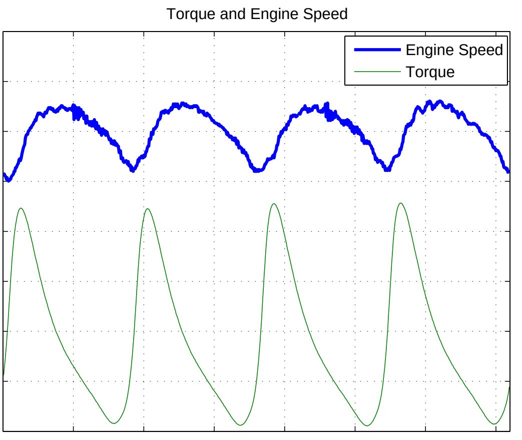


Figure 2.2: Engine Speed and Torque to illustrate the similarities between the two signals. The engine speed variations is greatly affected by the variations of the torque.

Figure 2.2 shows the engine speed and torque. As seen in the figure the engine speed is not constant. The torque greatly affects the variations of the engine speed, which variates slower than the torque due to inertia. The scales are modified to illustrate the variations of the engine speed that is caused by the combustion.

The crankshaft is also affected by other forces apart from the torque due to combustion. To be able to handle the total torque it is approximated into four basic groups.

- Indicated Torque (T_i)
- Mass Torque (T_m)
- Friction Torque (T_f)
- Load Torque (T_l)

Of these four torques only the *Indicated Torque* and the *Load Torque* is of interest to estimate.

Indicated Torque, T_i

The *Indicated Torque* or *Torque due to Combustion* is created from the combustion in the cylinder. This increases the pressure and generates a force on the piston. The force creates a oscillating motion of the piston through the connecting rod and a rotating motion of the crankshaft, *i.e.*, the indicated torque creates a rotation of the crankshaft. Hence, the indicated torque and the cylinder pressure are very close connected. For that reason the cylinder pressure is measured to calculate the indicated torque for validation. A more common notation for the work produced by the combustion is the Indicated Mean Effective Pressure (IMEP) which can be described by the integral

$$IMEP = \frac{1}{V_d} \oint_{\gamma} T_i \left(\theta\right) d\theta, \tag{2.1}$$

where γ represents a cycle. Hence, it is easy to switch between the notations.

Mass Torque, T_m

The *Mass Torque* is created by the inertia of all rotating and oscillating masses. This torque gives more influence on the total torque at higher engine speeds [6].

Friction Torque, T_f

The *Friction Torque* is a counteracting torque to the indicated torque caused by all friction affecting the crankshaft and the piston.

Load Torque, T_l

The *Load Torque* is the load acting at the crankshaft. For the experiments made in a testbed the load torque is created by a controllable brake and is of course measurable. In the experiments made in a car the load torque is naturally created but not possible to measure. Another notation for the load torque is Brake Mean Effective Pressure (BMEP) which can be calculated according to

$$BMEP = \frac{1}{V_d} \oint_{\gamma} T_l d\theta \tag{2.2}$$

where γ represents a cycle. Thus, for each cycle BMEP is

$$BMEP = \frac{4\pi \overline{T_l}}{V_d}. \tag{2.3}$$

Chapter 3

Engine Geometry

The geometry of the engine is of great importance when modeling the motion of the crankshaft. This chapter describes the necessary geometry. To simplify the description a single cylinder engine is considered.

3.1 Cylinder Geometry

In figure 3.1 the geometry of one cylinder is shown, where θ is the crank angle, A_p is the piston area, r is half the stroke and $s(\theta)$ is the piston displacement. The piston pin offset, y_{off}, is deliberate manufactured to decrease unwanted noise.

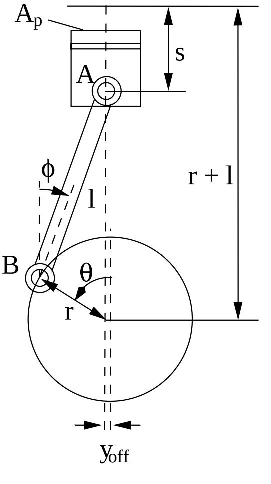


Figure 3.1: The crank-slider mechanism

The basic definitions in the figure are general but can be found in e.g. [9] and [6]. The piston pin offset, y_{off}, is given by a constant value retrieved from the manufacturer and the piston displacement is

$$s(\theta) = r + l - r\cos\theta - \sqrt{l^2 - (r\sin\theta - y_{off})^2}. \tag{3.1}$$

This gives the piston velocity

$$\frac{ds}{d\theta} = r\sin\theta + \frac{r\cos\theta(r\sin\theta - y_{off})}{\sqrt{l^2 - (r\sin\theta - y_{off})^2}} \tag{3.2}$$

and the piston acceleration

$$\begin{aligned}\frac{d^2s}{d\theta^2} &= r\cos\theta + \frac{r^2\cos^2\theta - r\sin\theta(r\sin\theta - y_{off})}{\sqrt{l^2 - (r\sin\theta - y_{off})^2}} \\ &+ \frac{r^2\cos^2\theta(r\sin\theta - y_{off})}{\left(\sqrt{l^2 - (r\sin\theta - y_{off})^2}\right)^3}.\end{aligned} \tag{3.3}$$

The derivation of $s(\theta)$,$ds/d\theta$ and $d^2s/d\theta^2$ can be found in e.g. [11] and [6].

3.2 Mass model

To simplify the calculations one can approximate the masses as two point masses — one oscillating, m_A, and one rotating, m_B as in [6]. This is done by dividing the overall rod mass into one oscillating and one rotating mass by the center of gravity as seen in figure 3.2.

The total oscillating mass for one cylinder is then given by

$$m_A = m_{piston} + m_{rod}\frac{l_{osc}}{l} \tag{3.4}$$

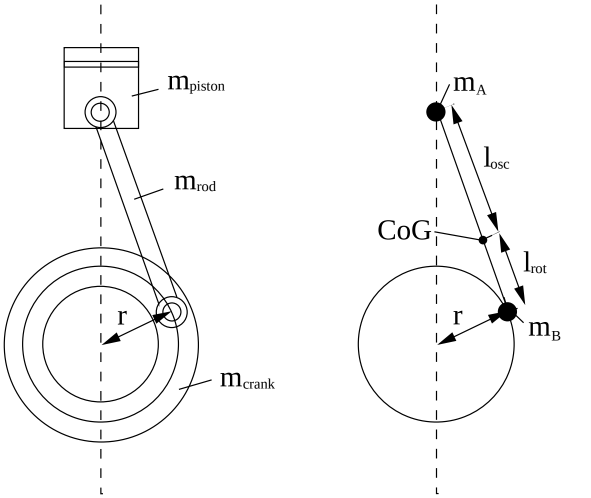


Figure 3.2: Two-mass model for oscillating and rotating masses

and the total rotating mass for one cylinder is given by

$$\frac{m_B}{N_c} = \frac{m_{crank}}{N_c} + m_{rod}\frac{l_{rot}}{l}, \tag{3.5}$$

where N_c is the total number of cylinders and the rod length is

$$l = l_{osc} + l_{rot}. \tag{3.6}$$

Chapter 4

Methods

There are a considerate amount of research made in estimating the torque affecting the crankshaft. The methods in estimating the engine torque can be grouped into four different approaches:

- Crankshaft Model [6, 11]
- Stochastic Estimation [10]
- Frequency Analysis [1, 3, 10]
- Synthetic Engine Speeds [8]

Below each method is briefly described.

4.1 Crankshaft Model

Modeling the crankshaft requires accurate knowledge of the engine and can be computationally expensive. To get a model that is possible to handle and implement, it is often needed to reduce the model. When modeling the crankshaft there are different approaches. A complex model with flexible crankshaft and dynamics of the engine gives a more accurate result but with the drawbacks of higher computational effort and a large number of parameters. Instead, by using a rigid body model of the crankshaft, the computational effort is kept low. The accuracy is of course also lower, especially at higher engine speeds where the nonlinearities increasingly affects the system [6].

4.2 Stochastic Estimation

The stochastic estimation method is based on a signal processing method. This approach has been used for estimation conditional averages from unconditional statistics, *i.e.* cross-correlation functions. The complexities in the system are self-extracted trough a number of correlation functions. When the correlation models are determined the estimation procedure is a simple evaluation of polynomial forms based on the measurements. The method has been implemented in real-time in [10].

4.3 Frequency Analysis

The engine process is of periodic nature and by examining the Discrete Fourier Transform of the engine speed signal, an accurate reconstruction of the torque can be created. This is made by calculating a frequency response function, from measurements, between the torque and engine speed in the frequency domain. The frequency response function is then used to reconstruct the torque from the engine speed signal. The main advantage of using the frequency domain is that the torque can be accurately described using only a few frequencies. Hence it is enough to utilize the corresponding frequencies in the engine speed signal to reconstruct the torque. This makes the signal filtered and only information that are strictly synchronous with the firing frequency is preserved. Also, only the first few harmonics of the engine firing frequency is of interest when performing the DFT and the other frequencies are negligible. The method has been implemented in [10].

4.4 Synthetic Engine Speeds

Synthetic engine speeds are defined as the crankshaft speeds with the effects of the reciprocating mass inertial torque removed. This is made through a pre-processing technique. Together with a frequency response function the cylinder pressure is reconstructed. The indicated torque is then easily calculated from the cylinder pressure.

4.5 Method Selection

The methods evaluated in this thesis are *Crankshaft Model* with a rigid body crankshaft and *Frequency Analysis*. The *Frequency Analysis* method is chosen for its simplicity and interesting approach. The *Crankshaft Model* method is chosen to compare performance with the *Frequency Analysis* method.

Chapter 5

Crankshaft Model

When modeling the crankshaft in the traditional way there is a balance between accuracy of the model and the computational effort [5]. In this chapter a rigid body model is described, with the goal of calculating the load torque. The rigid body model is used because of its simplicity and will give less computational effort.

5.1 Basic Dynamics

In the licentiate thesis by Schagerberg [11] and the book of Kiencke–Nielsen [6] the crankshaft torque is described using the balancing equation

$$J\ddot{\theta} = T_i\left(\theta\right) + T_m(\theta, \dot{\theta}, \ddot{\theta}) + T_f\left(\theta\right) + T_l, \tag{5.1}$$

where J is the crankshaft inertia and θ, $\dot{\theta}$, $\ddot{\theta}$ is the crank angle, angular velocity and angular acceleration respectively. The torques T_i, T_m, T_f and T_l are described earlier in chapter 2. With the geometry as described in chapter 3, the torque due to combustion or the indicated torque, T_i, is obtained trough the absolute pressure in the combustion chamber, $p(\theta)$, and the counteracting pressure on the back of the piston, p_0.

$$T_i\left(\theta\right) = \left(p\left(\theta\right) - p_0\right) A_p \frac{ds}{d\theta} = p_g\left(\theta\right) A_p \frac{ds}{d\theta}, \tag{5.2}$$

where the gas pressure is defined as $p_g = p\left(\theta\right) - p_0$. The counteracting pressure, p_0, is assumed to be the atmospheric pressure.

The mass torque is derived from the kinetic energy of the engine masses in motion, E_m.

$$E_m = \oint_\gamma T_m d\theta = \frac{1}{2} J \dot{\theta}^2 \tag{5.3}$$

The mass torque T_m is then the derivate of the kinetic energy

$$\begin{aligned} T_m = \frac{dE_m}{d\theta} &= \frac{1}{2}\left(\frac{dJ}{d\theta}\dot{\theta}^2 + J\frac{d}{d\theta}\left(\dot{\theta}^2\right)\right) \\ &= \frac{1}{2}\left(\frac{dJ}{d\theta}\dot{\theta}^2 + J\frac{d}{dt}\left(\dot{\theta}^2\right)\frac{1}{d\theta/dt}\right) \\ &= \frac{1}{2}\frac{dJ}{d\theta}\dot{\theta}^2 + J\ddot{\theta} \end{aligned} \tag{5.4}$$

The first term represents the rotating masses and the second one the oscillating masses.

With the connecting rod, the crankshaft and the piston approximated by a rigid weightless connection and two point masses, m_A and m_B, the mass torque is modeled with varying inertia and speed as

$$T_m(\theta, \dot{\theta}, \ddot{\theta}) = -\left(J_A\left(\theta\right) + J_B\right)\ddot{\theta} - \frac{1}{2}J'_A\left(\theta\right)\dot{\theta}^2, \tag{5.5}$$

with

$$J_A\left(\theta\right) = m_A\left(\frac{ds}{d\theta}\right)^2, \tag{5.6}$$

and

$$J_B = \frac{m_B}{N_c}r^2. \tag{5.7}$$

The derivative of $J_A\left(\theta\right)$ with respect to θ is

$$J'_A\left(\theta\right) = 2m_A\frac{d^2s}{d\theta^2}\frac{ds}{d\theta}. \tag{5.8}$$

5.2 Rigid Body Model

To simplify the calculations the crankshaft can be modeled as a rigid body. Then the contribution of the torque from all cylinders are simply summed together with a angular displacement, ψ_{n_c}, of $720/N_c$

degrees which gives

$$T_i(\theta) = \sum_{n_c=1}^{N_c} p_{g,n_c}(\theta) A_p \frac{ds(\theta - \psi_{n_c})}{d\theta} \tag{5.9}$$

$$T_m(\theta, \dot{\theta}, \ddot{\theta}) = -\sum_{n_c=1}^{N_c} \left\{ (J_A(\theta - \psi_{n_c}) + J_B)\ddot{\theta} - \frac{1}{2} J'_A(\theta - \psi_{n_c})\dot{\theta}^2 \right\} \tag{5.10}$$

and the friction torque modeled according to [11] as

$$T_f(\theta) = -C_f \dot{\theta}. \tag{5.11}$$

The torque balancing equation is then regrouped into an angle depended differential equation with time derivatives and formulated as

$$J(\theta)\ddot{\theta} = T_i(\theta) + \sum_{n_c=1}^{N_c} \frac{1}{2} J'_A(\theta - \psi_{n_c})\dot{\theta}^2 + T_f(\theta) + T_l. \tag{5.12}$$

where

$$J(\theta) = J + \sum_{n_c=1}^{N_c} (J_A(\theta - \psi_{n_c}) + J_B) \tag{5.13}$$

The second derivate of θ can be reformulated by substituting

$$\ddot{\theta} = \frac{d^2\theta}{dt^2} = \frac{d}{dt}\dot{\theta} = \frac{d\dot{\theta}}{d\theta} \cdot \frac{d\theta}{dt} = \frac{d\dot{\theta}}{d\theta} \cdot \dot{\theta} \tag{5.14}$$

into equation 5.12 gives

$$\dot{\theta} d\dot{\theta} = \frac{1}{J(\theta)} \left(T_i(\theta) + \sum_{n_c=1}^{N_c} \frac{1}{2} J'_A(\theta - \psi_{n_c})\dot{\theta}^2 + T_f(\theta) + T_l \right) d\theta. \tag{5.15}$$

Equation 5.15 is depending on time and crankshaft angle. If integrated it depends only on the square of the crankshaft angle speed, $\dot{\theta}$, instead of both crankshaft angle and time. This will make the differential equation event based, *i.e.* depending on the engine speed in crank angle domain.

$$\dot{\theta}^2(n+1) - \dot{\theta}^2(n) = \\ \frac{2}{J(\theta)} \int_{\theta(n)}^{\theta(n+1)} \left(T_i(\theta) + \sum_{n_c=1}^{N_c} \frac{1}{2} J'_A(\theta - \psi_{n_c})\dot{\theta}^2 + T_f(\theta) + T_l \right) d\theta \tag{5.16}$$

For a discrete angular step $\Delta\theta = \theta(n+1) - \theta(n)$ the integration may be approximated as,

$$\dot{\theta}^2(n+1) - \dot{\theta}^2(n) \approx \frac{2\Delta\theta}{J(n)}\left(T_i(n) + \sum_{n_c=1}^{N_c} \frac{1}{2} J'_A(\theta(n) - \psi_{n_c})\,\dot{\theta}^2(n) + T_f(n) + T_l(n)\right). \tag{5.17}$$

The measurements made for this thesis were using a crankshaft sensor wheel with 360 teeth, which makes the angular step $\Delta\theta = 1°$. With a 60 teeth crankshaft sensor wheel the angular step is instead $6°$. Instead of multiples of the sample time $n \cdot T_s$ there is multiples of the angular step $n \cdot \Delta\theta$. Equation 5.17 can be linearized by regarding the square of the crankshaft speed as a state variable x_1. A linear discrete state model of the crankshaft model is then obtained.

$$x_1(n+1) = \left(1 + \frac{2\Delta\theta}{J(n)} \sum_{n_c=1}^{N_c} \frac{1}{2} J'_A(\theta(n) - \psi_{n_c})\right) x_1(n) + \frac{2\Delta\theta}{J(n)} x_2(n), \tag{5.18}$$

with

$$x_1(n) = \dot{\theta}^2(n) \tag{5.19}$$

$$x_2(n) = T_i(n) + T_f(n) + T_l(n) \tag{5.20}$$

5.2.1 Indicated Torque

To retrieve the indicated torque, T_i, one can calculate x_2 at top and bottom dead center (TDC and BDC). For a four cylinder engine the indicated torque is zero at TDC and BDC as the piston stroke derivate $ds(\theta)/d\theta$ is zero in these points. This makes it possible to calculate the load and friction torque in the TDC and BDC points.

$$x_2(n_{TDC,BDC}) = T_f(n) + T_l(n) \tag{5.21}$$

The instantaneous indicated torque is derived by subtracting $x_2(n_{TDC,BDC})$ from x_2 for other angles.

$$T_i = x_2(n) - x_2(n_{TDC,BDC}) \tag{5.22}$$

The mean indicated torque can be calculated as the mean value of T_i.

5.2.2 Load Torque

The mean load torque for a cycle is also retrieved from $x_2\left(n_{TDC,BDC}\right)$ by subtracting the friction torque, T_f. It can be modeled as an absolute damper, $T_f = -C_f\dot{\theta}$, with the value of C_f depending on operating point. According to [11] an estimation of the constant C_f can be estimated for constant engine speed as

$$C_f = \frac{\overline{T_i} - \overline{T_l}}{\overline{\omega}}, \tag{5.23}$$

that also corresponds to the model made in [1] when the mean values are calculated for a cycle. The mean indicated torque is later calculated with frequency analysis in section 6.2 and the mean load torque is finally calculated as

$$T_l = \frac{x_2\left(n_{TDC,BDC}\right) + \overline{T_i\left(\theta\right)}}{2}, \tag{5.24}$$

with $\dot{\theta}$ as a mean value for a cycle, *i.e.* $\overline{\omega}$.

The friction can also be computed as a function of the engine speed trough a black box model according to [3, 4]

$$T_f = k_1 + k_2\theta + k_3\theta^2, \tag{5.25}$$

where the constants k_1, k_2 and k_3 are identified through experiments. The load torque can now be calculated as

$$T_l = x_2\left(n_{TDC,BDC}\right) - \left(k_1 + k_2\theta + k_3\theta^2\right), \tag{5.26}$$

Chapter 6

Frequency Analysis

The periodic nature of the engine speed makes it appropriate to use the Discrete Fourier Transform (DFT) as a tool for the analysis on this signal. According to [5] and Rizzoni et al. in [10], one can use the DFT to make an estimation of the instantaneous indicated torque and the accuracy of the estimation is improved by using the frequency domain rather than the time or crank angle domain. In the frequency domain, the DFT acts as a comb filter on the speed signal and preserves the desired information. Thus it is possible to use only a few frequency components of the measured engine speed signal to get an accurate result. This chapter describes and evaluates methods of estimating both instantaneous indicated torque and mean indicated torque with a low computational demand.

6.1 Instantaneous Indicated Torque

By using the frequency domain instead of time or crank angle domain it is only necessary to have a dynamic model representing the rotating motion at the frequencies that are harmonically related to the firing frequency. Then the engine dynamics can be described as a simplified Single-Input-Single-Output model as in figure 6.1.

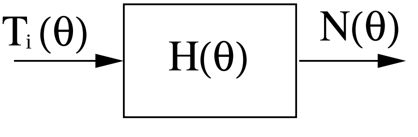


Figure 6.1: SISO model for the engine dynamics

In the figure the indicated torque, $T_i(\theta)$, is the input of the system, $H(\theta)$, and the crankshaft speed , $N(\theta)$, is the output resulting from the torque generated from the engine. Since both signals are obtained in the crank angle domain, the DFT generates a spatial spectrum. The relationship between the indicated torque and crankshaft speed in the spatial frequency domain can be described by

$$\hat{T}_i(f_k) = \hat{N}(f_k)\hat{H}^{-1}(f_k), \tag{6.1}$$

where f_k is the angular frequency in k:th order of rotation and $\hat{H}(f_k)$ is the engine frequency response function at that frequency [2, 7, 10]. $\hat{T}_i(f_k)$ is the DFT of the indicated torque and $\hat{N}(f_k)$ is the DFT of the crankshaft speed.

The frequency response function, $\hat{H}$, must first be obtained by experimental data at each of the first few harmonics of the engine firing frequency. Then together with the DFT of the engine speed one can calculate the DFT of the indicated torque for each selected harmonics. Finally the estimated indicated torque can be converted into crank angle domain by the inverse DFT. It is only required to use the first few harmonics to get an accurate estimation since the energy of the indicated torque is gathered at the first few harmonics as shown in figure 6.2. The method is based on simultaneous measurement of

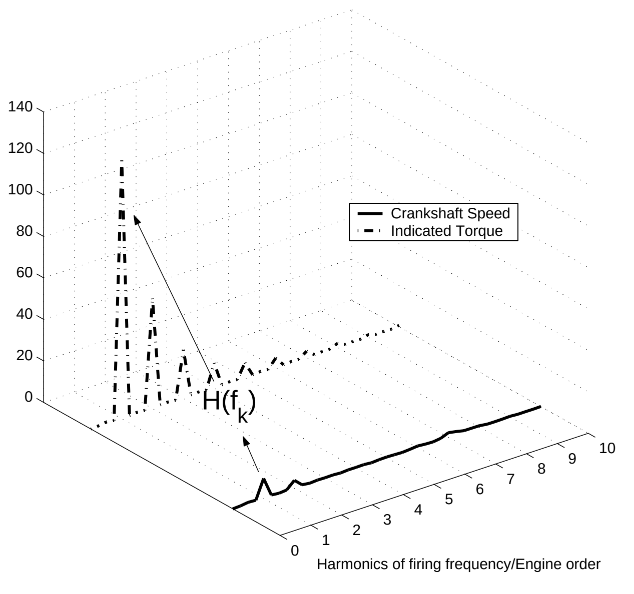


Figure 6.2: Comparison between DFT of Crankshaft Speed and Indicated Torque

the indicated pressure and crankshaft speed in the crank angle domain. The indicated torque for each cylinder can be created from equation 5.2. To get an accurate calculation of $\hat{H}(f_k)$ the measurement noise needs to be taken into account. By using the method of estimating the frequency response function made by Bendat & Piersol in [2] a more robust $\hat{H}(f_k)$ is generated. This is made by first estimating the power-spectral-density for $\hat{T}_i(f_k)$ and the cross-spectral-density for the signals with M cycles,

$$\hat{G}_{TT}(f_k) = \frac{1}{M}\sum_{j=1}^{M} |\hat{T}_i(f_k, j)|^2 \tag{6.2}$$

$$\hat{G}_{NN}(f_k) = \frac{1}{M}\sum_{j=1}^{M} |\hat{N}(f_k, j)|^2. \tag{6.3}$$

The cross-spectral-density is

$$\hat{G}_{TN}(f_k) = \frac{1}{M}\sum_{j=1}^{M} \hat{T}_i^*(f_k, j)\hat{N}(f_k, j), \tag{6.4}$$

with the $(*)$ representing the conjugate. The frequency response function is then generated by

$$\hat{H}(f_k) = \frac{\hat{G}_{TN}(f_k)}{\hat{G}_{TT}(f_k)}. \tag{6.5}$$

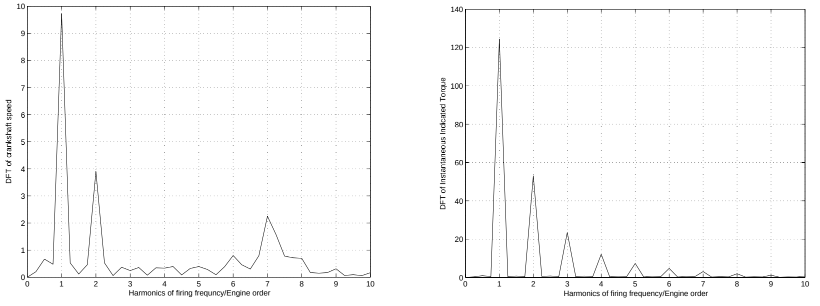
As previously mentioned, the first few harmonics of the engine firing frequency are sufficient to describe the engine dynamics and make an accurate estimation of the instantaneous indicated torque. When examining the DFT of the crankshaft speed and the calculated indicated torque in figure 6.3 it is seen that most of the energy is located at these first harmonics.

Another way to make these conclusion is to examine the coherence function.

$$\hat{\gamma}^2_{TN}(f_k) = \frac{|\hat{G}_{TN}(f_k)|^2}{\hat{G}_{TT}(f_k)\hat{G}_{NN}}(f_k) \tag{6.6}$$

$$0 \leq \hat{\gamma}^2_{TN}(f_k) \leq 1 \tag{6.7}$$

The coherence function is a measurement on how input and output of a system is related at every frequency. It is appropriate to use only



(a) DFT of Crankshaft Speed

(b) DFT of Instantaneous Indicated Torque

Figure 6.3: Illustration of the DFT of the Crankshaft Speed and the Instantaneous Indicated Torque with the mean value of the signal removed.

the frequencies which has a coherence close to one. This minimizes the influence of measurement noise in the model. In figure 6.4 it is confirmed that the first few harmonics of the firing frequency represent the process excellently since they are very close to one.

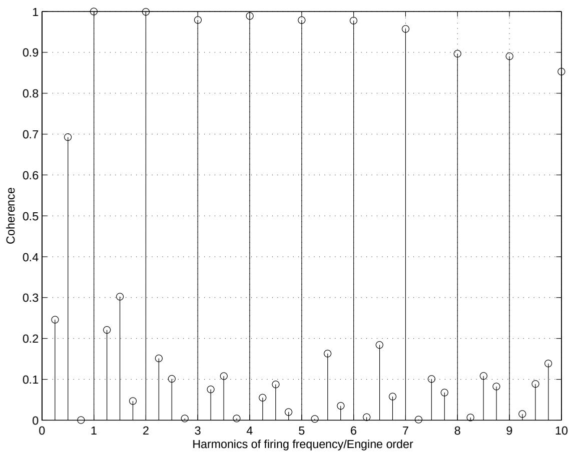


Figure 6.4: Coherence function for the DFT of the Crankshaft Speed and the Instantaneous Indicated Torque

6.2 Mean Indicated Torque

When estimating the instantaneous indicated torque using the DFT of the crankshaft speed and the frequency response function, only the fluctuations of the crankshaft speed is of interest. For this reason the mean value is removed when creating the DFT of the crankshaft speed. Otherwise the mean value will be spread over all frequencies and influence the estimation. This also removes the average component in the reconstructed indicated torque. However it is possible to retrieve information of the average torque from its fluctuating part. The Root-Mean-Square can be calculated from the other harmonics by

$$T_{RMS} = \frac{1}{\sqrt{2}} \sum_{k=1}^{P} |\hat{T}_i(f_k)|, \tag{6.8}$$

where P is the number of harmonics taken into account. The r.m.s. is strictly affine to mean indicated torque at each operating point and can be described with

$$T_{mean} = k_T \cdot T_{RMS} + m_T, \tag{6.9}$$

where k_T and m_T are constants determined from measurements. [10]

Since only a complex gain for every frequency is used when reconstructing the indicated torque from the crankshaft speed it should be possible to get a linear relationship to the mean indicated torque directly from the DFT of the crankshaft speed. Instead of using the r.m.s. from the reconstructed indicated torque it should be possible to use the r.m.s. of the crankshaft speed.

$$N_{RMS} = \frac{1}{\sqrt{2}} \sum_{k=1}^{P} |\hat{N}_i(f_k)| \tag{6.10}$$

$$T_{mean} = k_N \cdot N_{RMS} + m_N \tag{6.11}$$

Where k_θ and m_θ are constants that are determined from measurements. By using this method the used calculations to get the mean indicated torque decreases.

Chapter 7

Measurements

To validate the theory described in chapter 5 and 6 measurements from an engine are needed. Data was acquired from both an engine test bed and actual vehicle.

7.1 Engine Test Bed Measurements

All measurements were made on a four cylinder SAAB L850 engine. They were performed at the research laboratory of the Department of Vehicular Systems at Linköpings University consisting of an engine test cell and a control room. The sensors used to examine the torque due to engine speed variations was a crank angle sensor, cylinder pressure sensors and a torque sensor at the end of the crankshaft. The experimental setup is described in figure 7.1. The engine characteristics are described in appendix A, table A.1.The data was acquired at the engine operating points described in table 7.1 with approximately 100 engine cycles for each measurement. To evaluate the estimate of the mean load torque with varying operating points, some supplementary measurements were made at constant engine speed $(2000rpm)$ and varying load torque $(70Nm \to 15Nm \to 50Nm)$. Also, measurements were made with varying engine speed $(2000rpm \to 3000rpm \to 2500rpm)$ and varying load torque $(40Nm \to 70Nm \to 15Nm)$.

7.1.1 Data acquisition system

The data was acquired with two VXI-measurements instruments from Hewlett-Packard, HP E1415A and HP E1433A. The software have been customized for MatLab. The HP E1433A instrument can measure eight channels simultaneously with separate A/D converters and a

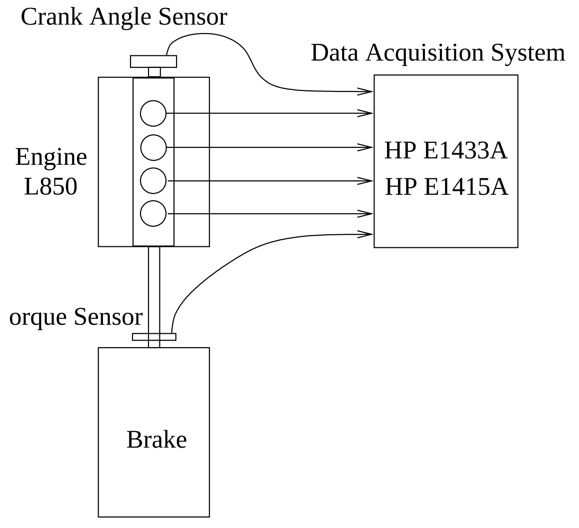


Figure 7.1: Experimental Setup

sampling frequency of 196 kHz. The HP E1415A instrument can measure 64 channels at a maximum sampling frequency of 2 kHz.

7.1.2 Crank Angle Sensor

The crankshaft angle was measured using a Leine & Linde 540 sensor with a resolution of 1°, *i.e.* 720 data points per engine cycle. The sensor was mounted at the free end of the crankshaft. The crank angle data was sampled with the HP E1433A instrument at 19.6 MHz and converted to crank angle domain.

7.1.3 Cylinder Pressure Sensor

The cylinder pressures were measured with AVL GU21D pressure sensors. They were sampled in the time domain at a frequency of 51.2 kHz with the HP E1433A instrument. The indicated pressure was then interpolated to crank angle domain off-line. In measurement with load 50, 80 and 120 Nm the indicated pressure for cylinders 1, 2 and 4 were measured, and not for cylinder 3 since the sensor for that cylinder was broken. Instead, a mean value of the pressure from cylinders 1, 2 and 4 were used as the pressure for cylinder 3. In measurement with load 15, 40, 70, 95 and 125 Nm the pressure for cylinder 1 and 2 were measured since only two sensors were available at this time. For cylinder 3 the pressure from cylinder 2 was used and for cylinder 4 the pressure from cylinder 1 was used.

Engine Speed [rpm]	**Load Torque [Nm]**							
	15	40	50	70	80	95	120	125
1000	X	X	-	-	-	-	-	-
1500	X	X	X	X	X	-	X	-
2000	X	X	X	X	X	X	X	X
2500	X	X	X	X	X	X	X	X
3000	X	X	X	X	X	X	X	X
3500	X	X	X	X	X	X	X	X
4000	X	X	X	X	X	X	X	X
4500	X	X	-	X	-	X	-	X
5000	X	X	-	X	-	X	-	-

Table 7.1: Engine Operating Points, "X" indicates a measurement and "-" indicates no measurement. The colored cells are measurements used for validation and the non colored are used for tuning.

7.1.4 Brake and Torque Sensor

With the break it is possible to operate the engine at a desired torque and engine speed. The load torque was measured at 10 Hz with the HP E1415A measuring instrument. Because of the low sampling frequency only the mean load torque was measured. Unfortunately it is not possible to receive the load torque in crank angle domain. Instead the measured load torque is fitted to the estimated load torque by hand when necessary.

7.2 Vehicle Measurements

The car used for measurements is equipped with a turbo-charged four cylinder SAAB L850 engine. The car is also equipped with a engine speed sensor and cylinder pressure sensors for all cylinders. The measurements were made using a dSpace MicroAutobox 1401. Since the vehicle have no torque sensor the load torque could not be measured. Hence, it is only possible to create an indicated torque estimation. Measurements were made in a wide spectrum of operating points.

Chapter 8

Results

The results are presented created from measurements, calculations and simulations of the methods in chapter 5 and 6. The methods are evaluated for both mean and instantaneous indicated torque. The load torque is also evaluated through the crankshaft model method. Only the frequency analysis method is tested with the measurements from a vehicle since the crankshaft model did not provide a good result.

8.1 Crankshaft Model

In this method the crankshaft is modeled as a rigid body. This makes the computational effort lower than in relation to a dynamic crankshaft model. The model is evaluated with Simulink in MATLAB.

8.1.1 Instantaneous Indicated Torque

The indicated torque is derived by subtracting the load and friction torque from the state parameter x_2 in equation 5.18. By solving the value of x_2 at TDC and BDC the load and friction torque is retrieved, since the indicated torque is zero at these points. The results of the simulations are shown in figures 8.1–8.3.

The result of the estimation is not satisfactory. For the estimation at 1000 rpm the estimation has the same shape as the calculated torque but has to low amplitude. The oscillations in the estimated torque signal for 2500 rpm and 4000 rpm can be explained by the dynamics in the crankshaft affects more with higher engine speeds.

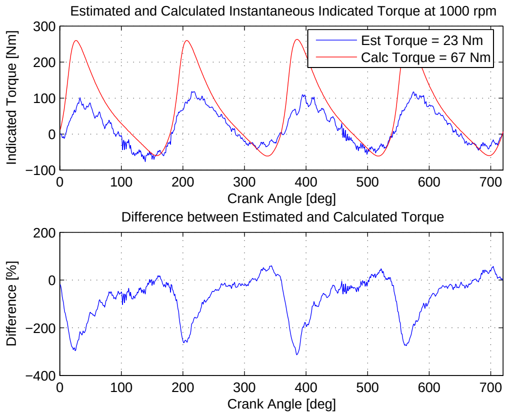


Figure 8.1: Estimated and calculated torque at 1000 rpm.

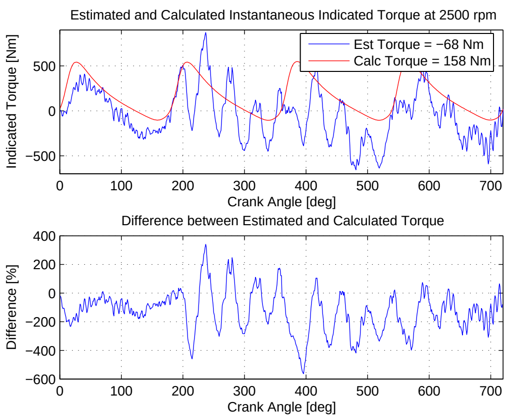


Figure 8.2: Estimated and calculated torque at 2500 rpm.

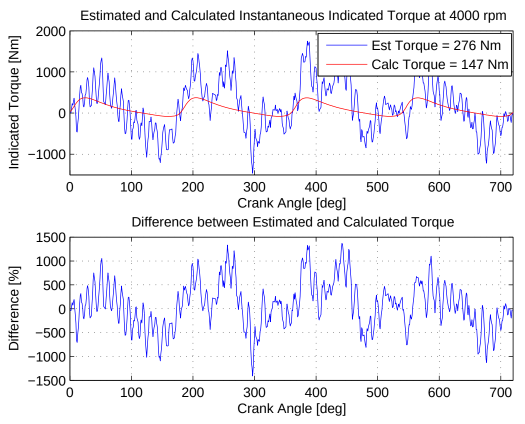


Figure 8.3: Estimated and calculated torque at 4000 rpm.

8.1.2 Mean Indicated Torque

The mean indicated torque is derived as the mean value of the instantaneous indicated torque and obviously also a poor estimation. A comparison between the estimated and calculated torque is presented in the legends of figures 8.1–8.3. The difference at high engine speeds makes the method useless.

8.1.3 Mean Load Torque

The load torque is derived from the state parameter $x_2(n_{TDC,BDC})$ in equation 5.18. The method of is illustrated in figure 8.4.

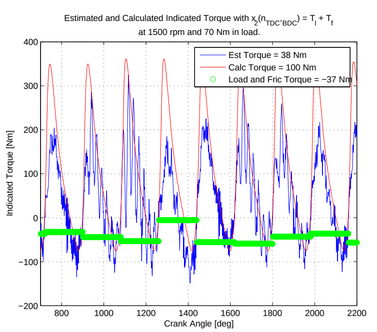


Figure 8.4: Illustration to retrieve the load torque.

The friction torque is derived according to the friction model in equation 5.25. The three constants are estimated trough identification and the friction torque is subtracted from $x_2(n_{TDC,BDC})$ to get the load torque. Since the model provided a poor estimation of the indicated torque it also provides a poor estimation of the load torque. In figure 8.5 the estimation is compared to the calculated torque for 100 cycles. The mean value of the difference between the estimated and the calculated torque is 47 % which is unacceptable.

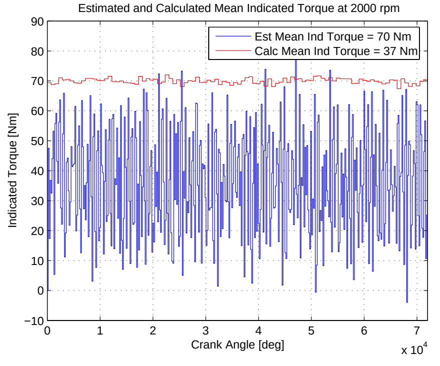


Figure 8.5: Estimated and Calculated Mean Load Torque at 1500 rpm.

Another problem with the estimation is the large variations between cycles. This could be an effect of noise in the engine speed signal. A way to handle this is to filter the engine speed. That would give a less variating estimation of the load torque but nevertheless a poor estimation.

8.2 Frequency Analysis

Only the indicated torque can be estimated when using the frequency analysis method (described in chapter 6). This is because of the low sample frequency at 10 Hz for the load torque. In chapter 6 a method with a frequency response function was used to reconstruct the instantaneous indicated torque. Another, more simple, method was also described to estimate the mean indicated torque. The estimated torque signal are compared with an torque signal calculated from the measured cylinder pressure, *i.e.* using equation 5.2 in chapter 5.

8.2.1 Instantaneous Indicated Torque

The instantaneous indicated torque is reconstructed from the DFT of the engine speed signal through a complex gain, the frequency response function, for each frequency of interest. This complex gain is pre–calculated once and for all using equations 6.2-6.5 for different engine operating points. The operating points used to calculate the frequency response function, $\hat{H}(f_k)$, are shown in table 7.1. In figures 8.6–8.8 the calculated $\hat{H}(f_k)$ is shown for all operating points at order 1-6. The surfaces is $\hat{H}(f_k)$ as a function of engine speed and mean indicated torque. In order to implement in the control system for an actual vehicle, the frequency response function is needed to be mapped or described by a function. The surfaces of the real and the imaginary part of $\hat{H}(f_k)$ at order one are harder to describe since they are not flat as the surfaces for the other orders. Unfortunately, it is also the most important order to use when reconstructing the torque since this order has the highest intensity in the torque (see figure 6.3(b)). A way to describe this order will be discussed later in this chapter.

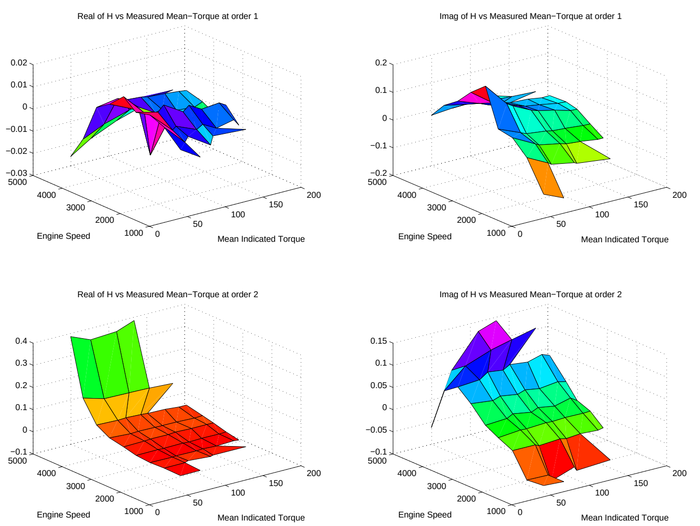


Figure 8.6: Real and imaginary part of $\hat{H}(f_k)$ for order 1–2

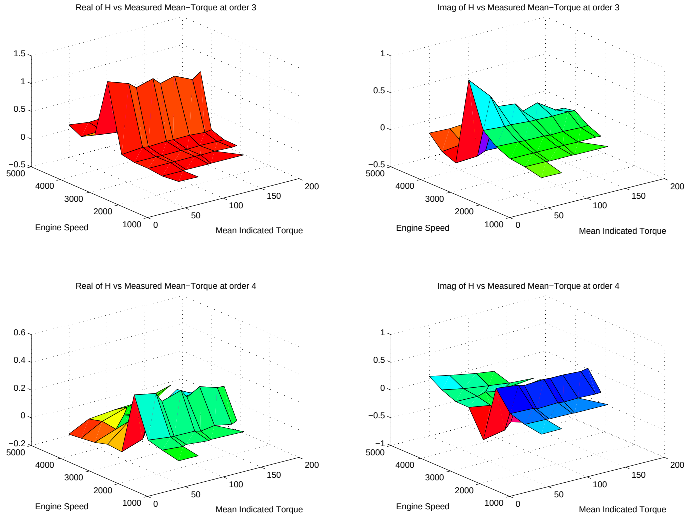


Figure 8.7: Real and imaginary part of $\hat{H}(f_k)$ for order 3–4

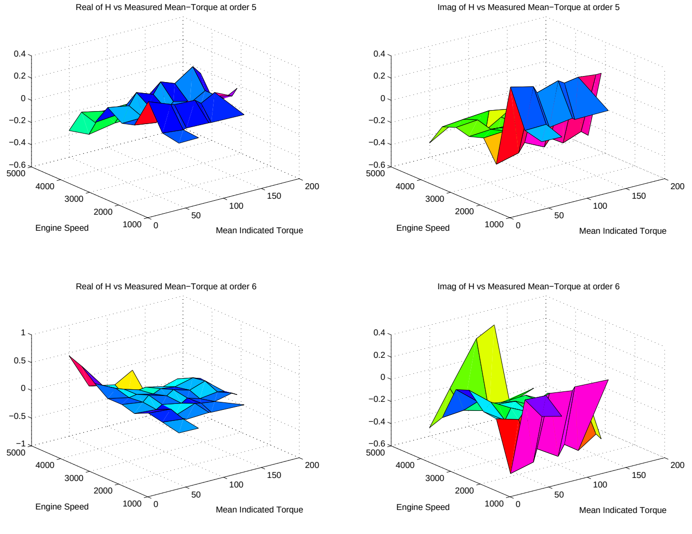


Figure 8.8: Real and imaginary part of $\hat{H}(f_k)$ for order 5–6

For the orders 2–5 the frequency response is almost constant for each engine speed independent of load. This means that the amplitude of the torque curve is decided of order one and the shape of the curve are decided from order 2 and forward. In other words, for order 2–5 only the mean value of $\hat{H}(f_k)$ for each engine speed is sufficient but for order one a function is needed to get the wanted value for $\hat{H}$.

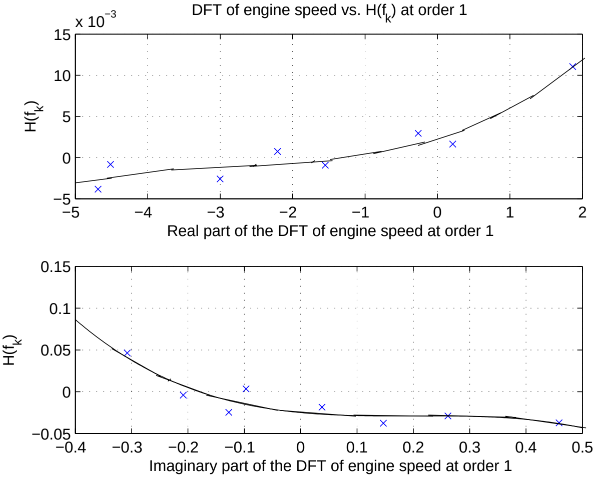


Figure 8.9: DFT of Engine Speed vs $\hat{H}$ at order 1 and 2000 rpm

Figures 8.9 and 8.10 show the DFT of the engine speed versus $\hat{H}$ at order one for 2000 and 3500 rpm. Hence, it is possible to describe $\hat{H}$ at order one as a quadratic or cubic polynomial depending on the DFT of the engine speed.

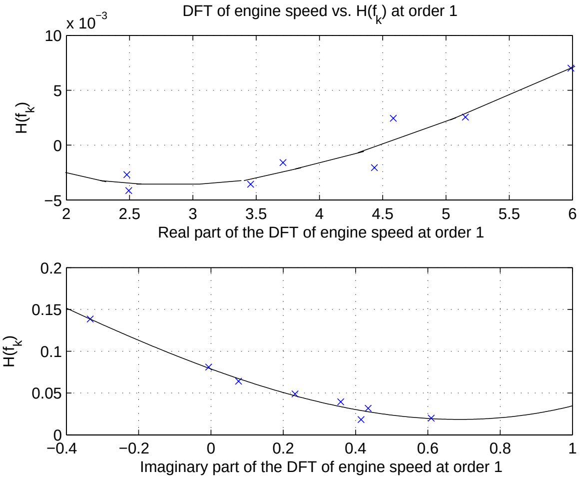


Figure 8.10: DFT of Engine Speed vs $\hat{H}$ at order 1 and 3500 rpm

For the simulations, polynomial of order three was used at each engine speed. For the other orders (2–5) a mean value was used. In chapter 6 it was shown that only a few harmonics of the engine firing frequency are sufficient to get an accurate estimation of the torque. When using only the first few harmonics, *i.e.* order 1–5, the estimations shown in figures 8.11–8.13 are received.

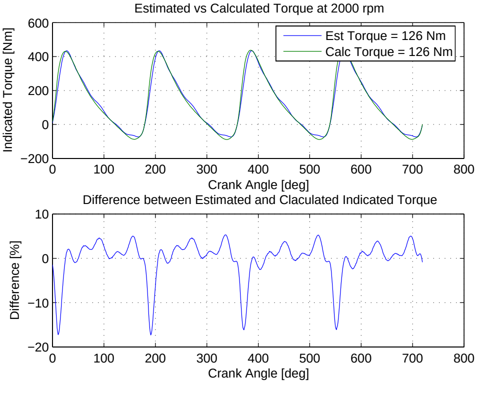


Figure 8.11: Estimated and Calculated Instantaneous Indicated Torque

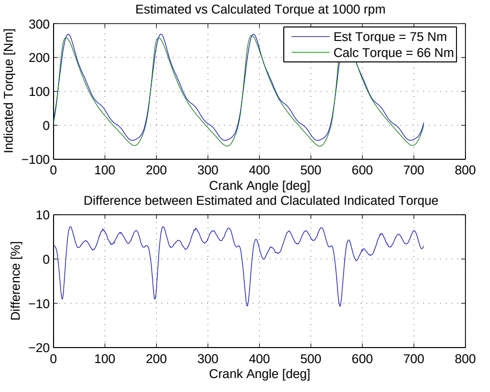


Figure 8.12: Estimated and Calculated Instantaneous Indicated Torque

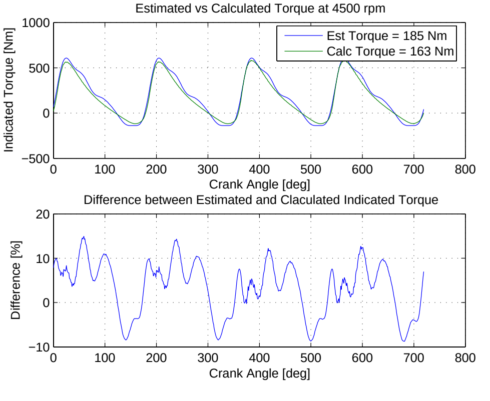


Figure 8.13: Estimated and Calculated Instantaneous Indicated Torque

As seen it is enough to use only order 1–5 to get an good estimation of the instantaneous indicated torque. The error between the estimated and calculated indicated torque is about $\pm 15\%$. Obviously it is only necessary to calculate the DFT of the engine speed at these corresponding frequencies. This prominently decreases the computational effort. The method is quite simple when $\hat{H}$ has been determined. First, the DFT for the engine speed needs to be calculated for the desired frequencies. The DFT is divided by $\hat{H}$ and the indicated torque is retrieved trough the inverse DFT. Also the mean indicated torque needs to be added according to equations 6.8 and 6.9 or equation 6.10.

8.2.2 Mean Indicated Torque

Instead of using a frequency response function when determining the mean indicated torque for a cycle, one can directly use the DFT of the engine speed since only a complex gain is used to reconstruct the torque. The mean indicated torque is computed according to equation 6.10. The sum of the DFT of the engine speed, or N_{RMS}, at the chosen frequencies is shown in figure 8.14. As seen in the fig-

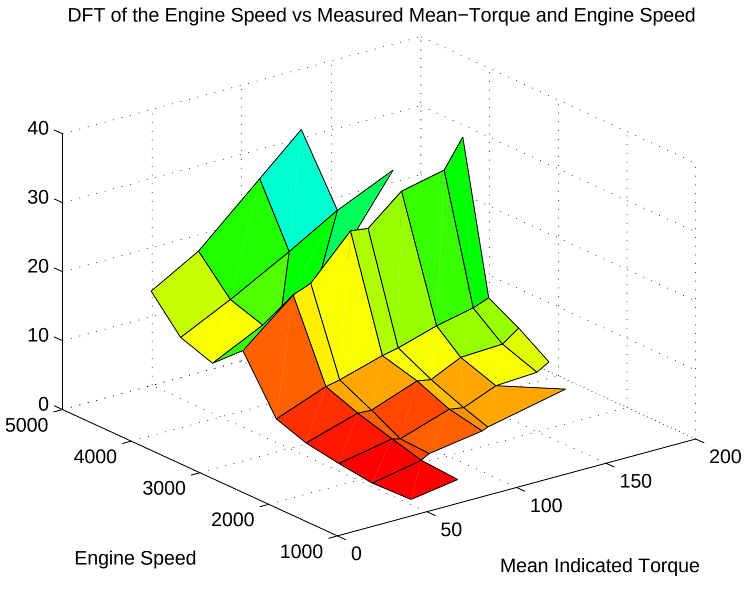


Figure 8.14: DFT of the Engine Speed vs Calculated Indicated Torque and Engine Speed.

ure and according to section 6.2 a straight line can approximately be used to describe the relation between N_{RMS} and the mean indicated torque. Figures 8.15–8.17 also show the linear relationship between them. The ”dent” in figure 8.14 could be be a undersampling effect but when increasing the sampling ratio the ”dent” remains.

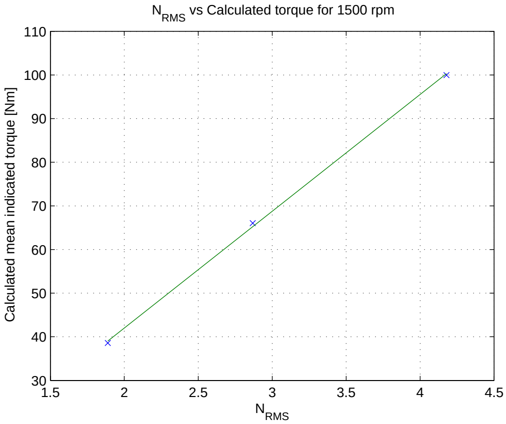


Figure 8.15: N_{RMS} vs Calculated Mean Indicated Torque

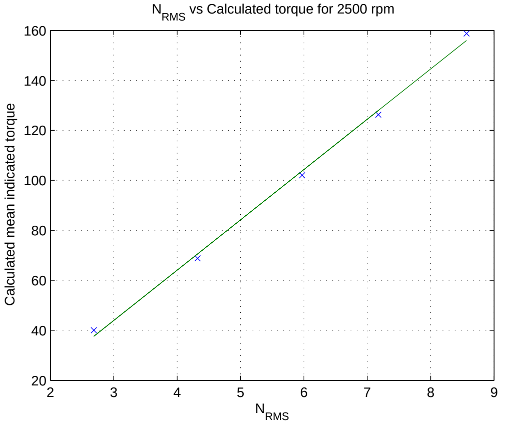


Figure 8.16: N_{RMS} vs Calculated Mean Indicated Torque

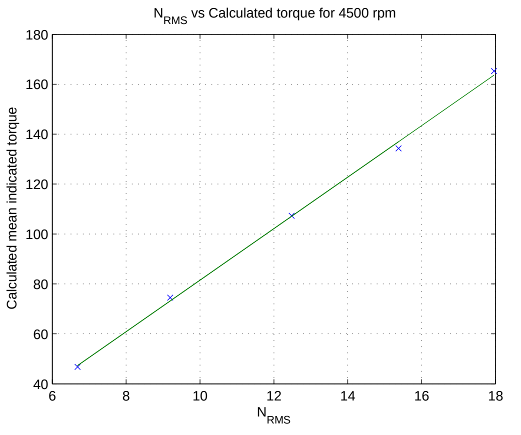


Figure 8.17: N_{RMS} vs Calculated Mean Indicated Torque

The constants for this straight–line relation , i.e. k_N and m_N, was pre–calculated and used to estimate the mean indicated torque. The results are shown in figures 8.18–8.20.

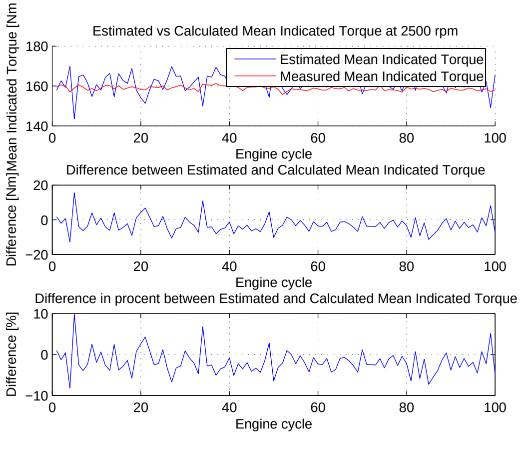


Figure 8.18: Estimated and Calculated Mean Indicated Torque for 1500 rpm

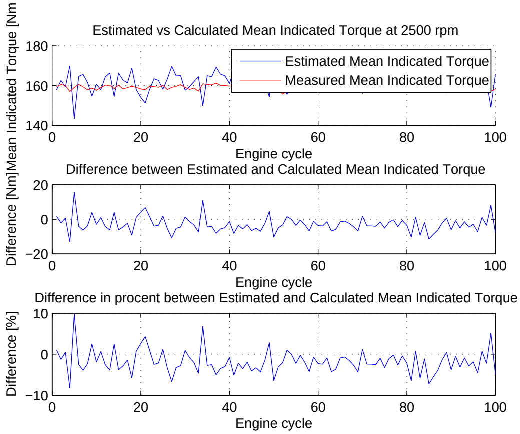


Figure 8.19: Estimated and Calculated Mean Indicated Torque for 2500 rpm

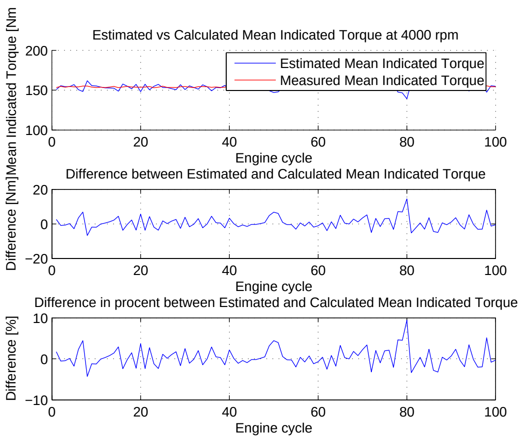


Figure 8.20: Estimated and Calculated Mean Indicated Torque for 4000 rpm

As seen in the figures the error of the estimation is between $\pm 10\%$. Thus, the estimation method of the mean indicated torque is accurate. The estimation for all engine operating points are presented in figure 8.21 which shows that the method has an error for less than $\pm 20\%$ for all 57 operating points.

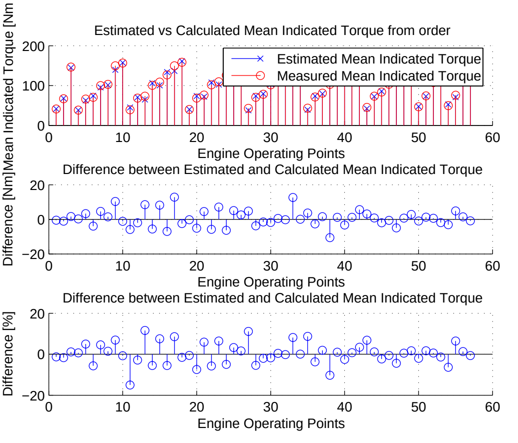


Figure 8.21: Estimated and Calculated Mean Indicated Torque for all engine operating points

Each measurement was made over 100 cycles and a histogram for N_{RMS} is shown in figure 8.22 to illustrate that the mean value for these 100 cycles can describe the relationship between N_{RMS} and the mean indicated torque.

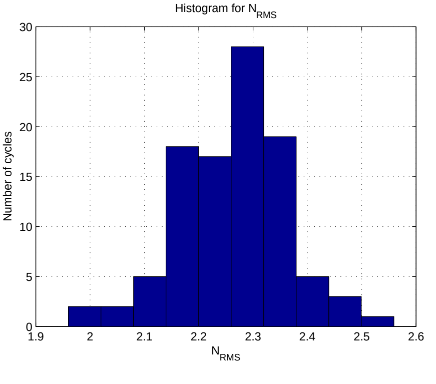


Figure 8.22: Histogram for N_{RMS} over 100 cycles.

8.2.3 Mean Indicated Torque for an Actual Vehicle

The method of frequency analysis has been shown to give an accurate estimation of the mean and instantaneous indicated torque when measurements was made in an engine test bed. Measurements were also made in an actual vehicle to see if the results differ from the earlier ones. Only the mean indicated torque for a cycle is of interest for the car and the instantaneous torque is not evaluated.

First, the same k_N and m_N as for the test bed data was used. These constants did not give an accurate estimation and new constants had to be created for the car. The reason for the bad estimation is the gear´s affection on the torque. The measurements were made at different load and gear with increments of approximately 500 rpm which gave the constants the restriction area of 500 rpm. The constants values were decided as in section 8.2.2 and are shown in figures 8.23 and 8.24.

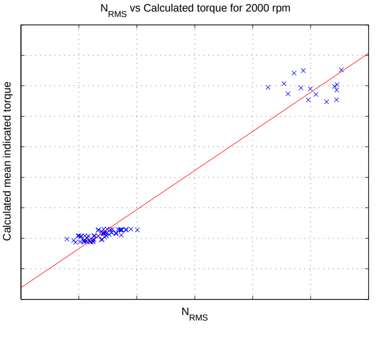


Figure 8.23: N_{RMS} vs Calculated torque for 2000 rpm and third gear

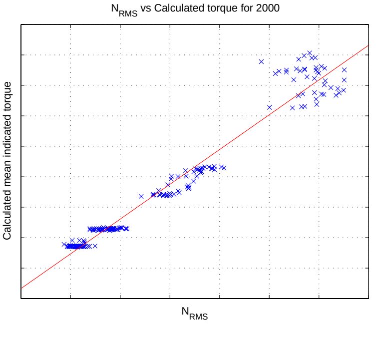


Figure 8.24: N_{RMS} vs Calculated torque for 2000 rpm and fourth gear

With the parameters which depend on engine speed and gear an estimation of the mean indicated torque can be done. Figures 8.25 and 8.26 present comparisons between the estimated and calculated mean indicated torque. The figures also shows the difference between the estimated and the calculated mean indicated torque in Nm and percent. The engine speed is also presented to show when the estimation method changes the value of the constants k_N and m_N. This is because the constants are estimated with increments of 500 rpm of the engine speed. In figure 8.25 this change can be seen at about cycle 15 and 30. In figure 8.26 the estimation is made for engine speed close to 2500 rpm and no change of the constants is necessary. Since the measurements were made in a car it was hard to keep the load and engine speed constant. If this is made it should be possible to interpolate between the constants k_N and m_N instead of switching between their values. This was not possible because of varying load and engine speed during the measurements.

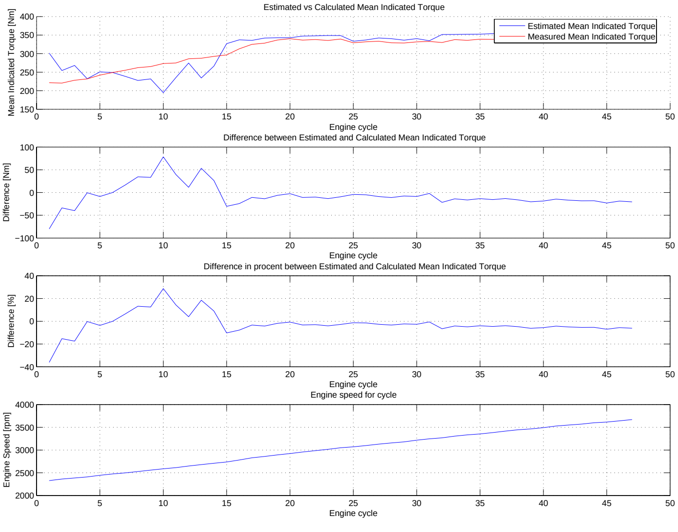


Figure 8.25: Estimated vs Calculated torque for second gear.
TOP: Estimated and calculated mean indicated torque.
TWO IN THE MIDDLE: Difference between the estimated and calculated torque in Nm and percent.
BOTTOM: Engine speed.

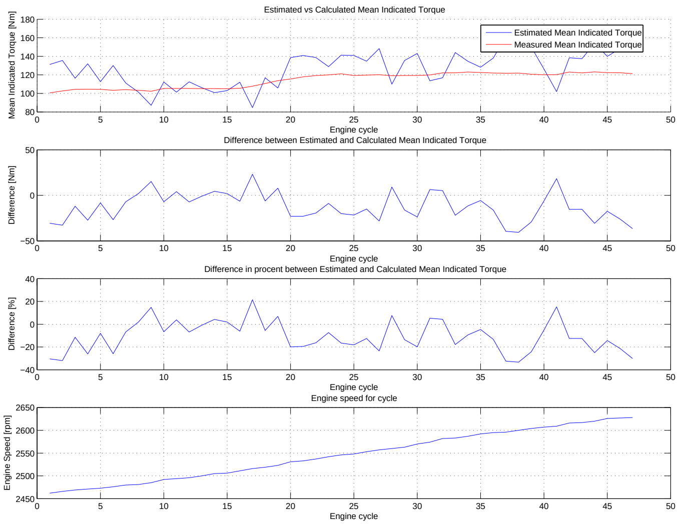


Figure 8.26: Estimated vs Calculated torque for third gear.
TOP: Estimated and calculated mean indicated torque.
TWO IN THE MIDDLE: Difference between the estimated and calculated torque in Nm and percent.
BOTTOM: Engine speed.

The figures show that the error of the estimation is between $\pm 20\%$. Thus, the method gives a fairly accurate estimation of the mean indicated torque.

Chapter 9

Conclusions

The objectives of the thesis was to evaluate the existing methods for estimating the engine torque from engine speed variations. Four known methods were described and two were chosen, implemented and tested: *Crankshaft Model* and *Frequency Analysis*

9.1 Crankshaft Model

It is not possible to use a rigid body model for the crankshaft. At higher engine speeds (over 2500 rpm) the dynamics of the crankshaft affects to much to get a good estimation. To get a more accurate estimation a more detailed model is needed for the crankshaft which also will increase the computational effort.

9.1.1 Indicated Torque

The indicated torque could not be estimated with a satisfying result with the crankshaft model method. This is explained by the lack of dynamics in the crankshaft model. The estimated instantaneous torque differed from the calculated between 0–200%, which is an unacceptable result.

9.1.2 Load Torque

The load torque could be estimated with a satisfying result with the crankshaft model method. The estimation differed from the calculated torque with approximately 50%, which is unacceptable.

9.2 Frequency Analysis

The frequency analysis method was shown to give an accurate result of both mean and instantaneous indicated torque. It is a simple method with a relatively low computational effort since only the first few harmonics of the firing frequency are needed.

9.2.1 Instantaneous Indicated Torque

The instantaneous indicated torque were evaluated for the measurements from the engine test bed. The correct frequency response function, $H(\hat{f_k})$, gives an excellent estimation of the torque. The problem is to find the right $H(\hat{f_k})$ and it was found that for order 2–5 a mean value for each engine speed could be used. For order one a polynomial of order three can describe how $\hat{H}(f_k)$ depends of $\hat{N}(f_k)$.

9.2.2 Mean Indicated Torque

The mean indicated torque could be estimated through straight–line equations of N_{RMS}, which is the sum of $\hat{N}(f_k)$ for $f_k = 2, ..., 10$. This gave an excellent estimation of the data measured in the engine test bed and a good estimation of the data measured in the vehicle. The problem is, as previously, to find the correct constants for the straight–line equations. The used constants gave satisfactory estimations but can be improved to increase the accuracy.

Chapter 10

Future Work

The frequency analysis method was shown to give an accurate estimate of both mean and instantaneous indicated torque. This can not be said about the Crankshaft Model method. A method not investigated in the thesis is the Stochastic Estimation method that could be interesting to investigate as a next step.

Because of lack of sensors the measurements made in the engine test bed were made with two or three sensors for cylinder pressure. If measurements is made for all four cylinders the results would be better.

10.1 Crankshaft Model

The crankshaft model could be expanded and improved according to [11]. It is the model of the torque due to masses, T_m, that is needed to be more complex since the crankshaft with a distributed mass does not behave as a rigid body at high engine speeds. This would give a better estimation but would also increase the computational demands of the control system.

10.2 Frequency Analysis

The most important way to improve the estimation of the indicated torque is to describe the frequency response function, H, more accurately. This could be done by using smaller increments of the engine speed when performing measurements. The reasons for the ”dents” at the surfaces from the frequency response function should be thoroughly investigated. If measurements of the load torque could be done at a higher sampling frequency it should also be possible to perform an estimation of the load torque with the frequency analysis method.

10.2.1 Ignition Timing

The shape of the torque signal is affected by the ignition timing. Especially late ignition timing which can change the torque signal peak into a two peaks signal. Hence, it should be investigated how the ignition timing affects the torque signal. It could then be used as a parameter in the frequency analysis method.

10.2.2 Actual Vehicle

The problem with the estimations for the car was the measurements. A problem was that it was not possible to perform measurements with constant load and engine speed at high loads. To improve the estimations they should be carried out with constant load and engine speed to be able to interpolate between the estimated constants k_N and m_N. Also, measurements could be made with smaller increments than 500 rpm.

References

[1] P.M. Azzoni, G. Minelli, D. Moro, R. Flora, and G. Serra. Indicated and load torque estimation using crankshaft angular velocity measurements. *SAE Technical Paper Series*, 1999-01-0543.

[2] J. Bendat and A. Piersol. *Engineering Applications of Correlation and Spectral Analysis*. John Wiley & Sons, 1980.

[3] N. Cavina, F. Ponti, and G. Rizzoni. Fast algorithm for on-board torque estimation. *SAE Technical Paper Series*, 1999-01-0541.

[4] P. Falcone, G. Fiengo, and L. Glielmo. Nicely nonlinear engine torque estimator. In *16th IFAC World Congress*, Prague, 2005.

[5] E. Gani and C. Manzie. Indicated torque reconstruction from instantaneous engine speed in a six-cylinder si engine using support vector machines. *SAE Technical Paper Series*, 2005-01-0030.

[6] U. Kiencke and L. Nielsen. *Automotive Control Systems For Engine, Driveline and Vehicle*. Springer, 2000.

[7] L. Ljung. *System Identification - Theory for the User*. Prentice-Hall, 2 edition, 1999.

[8] D. Moro, N. Cacina, and F. Ponti. In-cylinder pressure reconstruction based on instantaneous engine speed signal. In *Transactions of the ASME*, 2002.

[9] L. Nielsen and L. Eriksson. *Coures material Vehicle Systems*. Bokakademin, 2004.

[10] G. Rizzoni, B. Lee, Y. Guezennec, A. Soliman, M. Cavalletti, and J. Waters. Engine control using torque estimation. *SAE Technical Paper Series*, 2001-01-0995.

[11] S. Schagerberg. *Torque Sensors for Engine Applications*. Lic thesis 472L, Chalmers University of Technology, 2003.

Notation

Variables and parameters

θ	Crank angle	$[rad]$
$\dot{\theta}$	Crank angle velocity	$[rad/s]$
$\ddot{\theta}$	Crank angle acceleration	$[rad/s^2]$
r	Half of the stroke	$[m]$
y_{off}	Piston pin offset	$[m]$
s_θ	Piston displacement	$[m]$
l	Length of rod	$[m]$
l_{osc}	Length of the oscillating part of the rod	$[m]$
l_{rot}	Length of the rotating part of the rod	$[m]$
m_A	Oscillating mass	$[kg]$
m_B	Rotating mass	$[kg]$
m_{piston}	Mass of piston	$[kg]$
m_{rod}	Mass of rod	$[kg]$
m_{crank}	Mass of crankshaft	$[kg]$
A_p	Piston area	$[m^2]$
N_c	Number of cylinders	$[-]$
N	Engine speed in crank angle domain	$[rpm]$
T_i	Indicated torque, i.e. torque due to combustion	$[Nm]$
T_m	Torque due to motion of masses	$[Nm]$
T_f	Friction torque	$[Nm]$
T_l	Load torque	$[Nm]$
p_g	Gas pressure, i.e. pressure due to combustion	$[Pa]$
p_0	Counteracting pressure on the back of the piston	$[Pa]$
J	Crankshaft inertia	$[kgm^2]$
$J(\theta)$	Total Inertia	$[kgm^2]$
J_A	Inertia of oscillating masses	$[kgm^2]$
J_B	Inertia of rotating masses	$[kgm^2]$
E_m	Energy of the masses in motion	$[J]$
$IMEP$	Indicated Mean Effective Pressure	$[Pa]$
$BMEP$	Brake Mean Effective Pressure	$[Pa]$
H	Frequency response function	$[-]$
f_k	Order k or frequency at harmonic k	$[-]$
γ	Coherence function	$[-]$

Operators

$\hat{Q}$	Fourier transform of Q
Q^*	Conjugate of Q
$\overline{Q}$	Mean value of Q

Appendix A

Engine Characteristics

The engine used in the examination is a four cylinder L850 in a testbed with brakes and a torque sensor. Data for the engine is presented in table A.1.

Engine Type	Four cylinder, spark ignited
Stroke	86 mm
Connecting Rod Length	145.5 mm
Displacement Volume	2.3 dm^3
Number of Valves	8
Compression Ratio	9.3

Table A.1: Engine data for the SAAB L850 engine.

Copyright