1-D Simulation of Turbocharged SI Engines
- Focusing on a New Gas Exchange System
and Knock Prediction

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Academic thesis, which with the approval of Kungliga Tekniska Högskolan, will be presented for public review in fulfilment of the requirements for a Licentiate of Engineering in Machine Design. The public review is held at Kungliga Tekniska Högskolan, Brinellvägen 64 in room M3 at 10:00 on the 15th of December 2006.
Abstract

This licentiate thesis concerns one dimensional flow simulation of turbocharged spark ignited engines. The objective has been to contribute to the improvement of turbocharged SI engines' performance as well as 1-D simulation capabilities.

Turbocharged engines suffer from poor gas exchange due to the high exhaust pressure created by the turbine. This results in power loss as well as high levels of residual gas, which makes the engine more prone to knock.

This thesis presents an alternative gas exchange concept, with the aim of removing the high exhaust pressure during the critical periods. This is done by splitting the two exhaust ports into two separate exhaust manifolds.

The alternative gas exchange study was performed by measurements as well as 1-D simulations. The link between measurements and simulations is very strong, and will be discussed in this thesis.

As mentioned, turbocharged engines are prone to knock. Hence, finding a method to model knock in 1-D engine simulations would improve the simulation capabilities. In this thesis a 0-D knock model, coupled to the 1-D engine model, is presented.

Keywords: spark ignited engines, 1-D flow simulation, knock, Divided Exhaust Period, turbocharged engines
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Of course many more have left a lasting impression during these years.
List of publications

Paper I

The disposition and writing of the paper was a co-operation between me and Fredrik Westin, who also performed the experimental data and most of the simulations.

Paper II

In this paper I focused on the background theories and simulations and Lindström on the experimental side.

Paper III

In this paper my focus was on the simulation part and building the knock model and Fredrik Lindström collected the experimental data.

All three papers are appended to the end of this thesis.
Abbreviations, symbols and subscripts

**Abbreviations:**
- SI: Spark Ignited
- CI: Compression Ignited
- TDC: Top Dead Center
- BDC: Bottom Dead Center
- IVO: Inlet Valve Opening
- IVC: Inlet Valve Closing
- EVO: Exhaust Valve Opening
- EVC: Exhaust Valve Closing
- CA: Crank Angle
- IMEP: Indicated Mean Effective Pressure
- PMEP: Pump Mean Effective Pressure
- BMEP: Brake Mean Effective Pressure
- FMEP: Friction Mean Effective Pressure
- t/c: turbocharged
- KO: knock onset
- KI: knock index

**Symbols:**
- A: cross-sectional area and scaling factor in ignition delay correlation
- B: temperature coefficient in ignition delay correlation
- C_f: friction loss coefficient
- C_p: pressure loss coefficient
- D: equivalent diameter
- H: total enthalpy
- I: inertia
- M: torque
- N: engine speed
- P: power
- Q_{ch}: heat release during combustion
- Q_{LHV}: fuel lower heating value
- \dot{Q}: heat flow
- T: temperature
U  mean fluid velocity
V_d  cylinder swept volume
W  work
a  scaling factor for the Wiebe function
c  speed of sound
c_p  specific heat
e  internal energy
h  enthalpy
h_g  heat transfer coefficient
h_c  heat transfer coefficient in the cylinder
m  mass and scaling factor for the Wiebe function
n  pressure exponent in ignition delay correlation
n_R  stroke factor, equals 2 for 4-stroke engines
p  pressure
t  time
x_b  percentage burned mass
\eta  efficiency
\rho  density
\lambda  relative air-fuel ratio
\gamma  specific heat ratio
\omega  rotational speed
\theta  crank angle
\tau  ignition delay

Subscripts:
a  air
b  brake
c  compressor
e  exit
f  fuel
i  inlet
T  turbine
u  unburned
v  volumetric
w  wall
n  net
g  gross
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Chapter 1 Introduction

Over a hundred years ago, transportation was revolutionised with the introduction of the internal combustion engine. Its promise of flexible and effective mobility was until then unheard of. In 1889 Daimler produced a four-wheeled passenger car with a maximum speed of 16 km/h [1]. Although this seemed powerful at the time, the quest for performance improvements is as old as the internal combustion engine itself. One way of improving engine performance is the use of a turbocharger. In 1925 a Swiss engineer named Alfred Büchi [2] managed to increase the engine power with 40% by using the energy of the exhaust gas for turbocharging. In the early days of turbocharging, the technology was mainly adopted to large diesel engines. As the technology improved, it became possible to develop small turbochargers to be fitted into small gasoline engines. In the beginning this was primarily adopted for race engines. The history of mass produced turbocharged gasoline engines is much influenced by the developments at Saab-Scania AB. Saab-Scania was founded in 1969 as a merge between the aircraft developer Saab (Svenska Aeroplan AB) and the truck developer Scania-Vabis. With the knowledge from developing turbocharged truck engines, the first Saab-Scania turbocharged SI engine could be introduced in 1977. It was a two litre engine fitted into the Saab 99 model.

Introduction of new technologies, regulations and customer demands all play a role in the development of the modern internal combustion engine. Today the
focus of research and development is on fuel efficiency improvements and emissions reductions. Turbocharging can play an important role in meeting future emissions legislation. Historically it was mainly used as a power boost and performance improver. However, lately turbocharging has gained popularity due to its positive effects on fuel consumption and emissions. Unfortunately, turbocharged engines suffer from some drawbacks not experienced in natural aspirated engines. Some of these drawbacks, such as the back-pressure caused by the turbine and knock, will be addressed in this thesis.

In order to make engine development more efficient and effective simulation tools are widely used. Simulation tools can offer flexibility not obtained with experimental studies. This should not be misinterpreted into believing that experimental studies are obsolete, just the opposite. Simulation tools, both 1-D and 3-D, are based on the conservation equations but also rely on empirical relationships and measurements to describe certain phenomenon. In this thesis the commercial software GT-Power has been used to model a turbocharged SI engine.

1.1 Objectives

The objectives of this thesis have been to contribute to the improvement of turbocharged SI engines’ performance as well as 1-D simulation capabilities. A new gas exchange system for turbocharged engines, called Divided Exhaust Period, has been investigated. The aim of this system is to improve the performance by reducing the efficiency losses associated with turbocharged engines. These losses are closely linked to the back pressure created by the turbine and the aggravated knock tendencies. Knock is a phenomenon that appears when the unburned gas in front of the propagating flame autoignites and triggers pressure oscillations in the combustion chamber. Being able to distinguish between non-knocking and knocking combustion will improve the simulation capabilities. Simulating knock accurately involves both chemical kinetics and 3-D flow; hence it will be very expensive in computational time.
Chapter 1 - Introduction

The objective in this thesis was to model knock in a 1-D simulation environment.

1.2 Thesis outline

First of all some general definitions, commonly used for comparing engines and validating engine models, are presented in chapter 2.

This is followed by chapter 3, a short introduction to 1-D gas dynamic modeling and the experimental set-up. These topics are linked by the model validation part that follows, which is also covered in Paper I.

Chapter 4 is an introduction and background to Paper II, in which a new gas exchange system for turbocharged engines is presented. The system is called Divided Exhaust Period due to its main feature, the splitting of the exhaust ports into two separate manifolds.

Chapter 5 is an introduction and background to Paper III. This chapter focuses on the simulation of combustion in general and specifically knocking combustion. Knock is a common phenomenon in SI-engines, generated by pressure waves propagating in the cylinder. It is highly undesirable due to its potentially damaging effects.

Chapter 6 contains a short conclusion.
Chapter 2 Spark ignition engine features

The names Spark or Compression Ignition enlightens the main difference between the two types of engines used in commercial vehicles. In a spark ignition (SI) engine the fuel and air mixture is ignited by a spark from an external energy source, i.e. a spark plug. In a compression ignition (CI) engine, on the other hand, the fuel autoignites due to high pressure and temperature caused by the moving piston. This work focuses on turbocharged SI engines. In this chapter, some general definitions of SI engine performance will be made. This is followed by a short introduction to turbocharging and engine combustion. The definitions will be used for validating simulation models as well as in the further discussion of engine performance.

2.1 Mean effective pressure

In an engine, work is produced by the combustion of fuel and the subsequent pressure increase and change of cylinder volume. Figure 1 shows a diagram of the pressure versus corresponding cylinder volume over the engine’s operating cycle.
The indicated work per cycle is obtained by the integration of cylinder pressure over the corresponding volume change:

\[ W = \int p \, dV \]  

where \( p \) is the cylinder pressure and \( V \) the cylinder volume. In a four stroke engine the piston is at its upper most and bottom most position twice per cycle. Some ambiguity about the cyclic integration can occur since the compression and expansion stroke define one cyclic motion and the intake and exhaust another. Hence two different definitions of indicated work exist. Gross indicated work is defined as the work delivered during the compression and expansion strokes while Net indicated work implies the work generated during the entire four-stroke cycle.

In order to simplify comparisons between engines of different size in terms of work output, a quantity independent of engine size has been defined and called mean effective pressure. In accordance with the previous definition of Net and Gross, the Net Indicated Mean Effective Pressure is defined as:
where $V_d$ is the swept cylinder volume and $p$ the cylinder pressure. Gross IMEP, on the other hand, is defined over the compression and expansion stroke:

$$IMEP_g = \frac{1}{V_d} \int_{-180}^{180} p dV$$  

The difference between net and gross IMEP is called Pump Mean Effective Pressure, PMEP or pump loss:

$$PMEP = IMEP_g - IMEP_n$$  

From eq. 2-4 it is evident that a positive PMEP means negative work on the crankshaft. PMEP and IMEP can also be expressed by the areas formed in the pressure versus volume diagram, see Figure 1:

$$PMEP = Area B + Area C$$  

$$IMEP_g = Area A + Area C$$  

The indicated power is the maximum available power during an engine cycle:

$$P = \frac{WN}{n_R}$$  

where $N$ is the engine speed and $n_R$ is the stroke factor (equals 2 for 4-stroke engines) which takes into account that the engine makes two revolutions for each power stroke. However some power is needed to overcome engine friction, to drive engine accessories and overcome the pumping power. What is left is the brake power:
\[ P_b = 2\pi NM \quad \text{eq. 2-8} \]

where \( N \) [rev/s] is the engine speed and \( M \) the engine torque. The term Brake refers to the usable power and is used for other quantities such as mean effective pressure. Hence Brake Mean Effective Pressure can be related to IMEP as:

\[ BMEP = IMEP_g - FMEP - PMEP \quad \text{eq. 2-9} \]

where FMEP is the friction mean effective pressure, which includes the work required to overcome engine friction and to drive engine accessories. BMEP can also be expressed as work per volume by using eq. 2-7 and 2-8:

\[ BMEP = \frac{W_b}{V_d} = \frac{P_b n_R}{N V_d} = \frac{2 m_R M}{V_d} \quad \text{eq. 2-10} \]

### 2.2 Power

The power was defined in eq. 2-7 as work per unit of time. Fuel and air are the main components for work generation in an engine. Hence, the power can be expressed in terms of how efficiently fuel is converted and air induced. The actual power of an engine can be expressed according to eq. 2-11, for derivation the reader is referred to any textbook on internal combustion engines, i.e. Heywood [4]:

\[ P = \frac{N}{n_R} \eta_f \eta_v \rho_{a,i} V_d Q_{LHV} \left( \frac{m_f}{m_a} \right) \quad \text{eq. 2-11} \]

where \( \eta_f \) is the fuel conversion efficiency, \( \eta_v \) the volumetric efficiency, \( \rho_{a,i} \) the density of the intake air, \( Q_{LHV} \) the fuel lower heating value and \( \left( \frac{m_f}{m_a} \right) \) the fuel-to-air ratio. Hence, in order to increase the power for a specific four-stroke
engine with a fixed fuel, the intake air density, engine speed, volumetric efficiency or fuel-to-air ratio or $\eta_f$ should be increased. In race car engines, it is common to increase the speed, but in commercial vehicles this is not considered an option, due to noise and durability. Variation in the fuel-to-air ratio is limited due to emission regulations and combustion stability. However, the density and volumetric efficiency can be changed.

The density can be increased by increasing the pressure or decreasing the temperature. The pressure can be increased by a compressor. Since the compressor also increases the temperature of the gas, cooling the gas can further increase the density. This is commonly done by a heat exchanger, a so-called intercooler, placed after the compressor exit.

The effectiveness of the intake system’s ability to supply the cylinder with fresh air is given by the volumetric efficiency. Generally the volumetric efficiency can be considered to be a relationship between the actual inducted amount of air into the cylinder at each cycle and the ideal displaced amount:

$$\eta_v = \frac{m_{a,i}}{\rho_a V_d}$$

where $m_{a,i}$ is the mass of air induced into the cylinder at each cycle and $\rho_a$ the density of air. The volumetric efficiency will increase by increasing the amount of air into the cylinder. This can be done by reducing the flow restrictions in the induction system i.e. inlet pipes, ports, valves and air filters. However, the volumetric efficiency can also be increased by improving the expulsion of exhaust gas. Due to negative pressure difference between cylinder and exhaust manifold at the end of the exhaust stroke some exhaust gas will be left in the cylinder. By decreasing this amount of gas, the volumetric efficiency can be improved since the actual induced amount of air can be increased.
2.3 Gas exchange

In a four-stroke engine a complete engine cycle is defined by 720 crank angle (CA) degrees. Engine operation can be distinguished by the four phases, compression, expansion, exhaust and intake, see Figure 1. The combination of exhaust and intake phases is called the Gas exchange. The purpose of the gas exchange is to empty the combustion chamber of exhaust and refill it with fresh air and fuel mixture. Figure 2 show the exhaust and intake valve lifts as a function of crank angle degrees. Some definitions are made; BDC, Bottom Dead Centre, is when the piston is at its lowest position, TDC, Top Dead Centre, is when the piston is at its highest position, EVO is the exhaust valve opening time, EVC the closing time and IVO the inlet valve opening time and IVC the corresponding closing time, the overlap is defined as the time between IVO and EVC.

![Exhaust and intake valve lifts for a conventional 4-stroke engine. EVO is the Exhaust Valve Opening, EVC the Exhaust Valve Closing, IVO and IVC are similarly defined for the intake valve.](image)

Slightly prior to BDC, the exhaust valves open. The piston moves towards TDC, meanwhile displacing the exhaust gas. Prior to TDC the intake valves open and as the piston moves towards BDC, fresh air-fuel mixture is sucked into the engine.
The timing of these valve lifts affects engine performance. The pressure difference between the cylinder and the manifolds at valve lift opening and closing depends on valve lift timing and affect the induction of air and expulsion of exhaust. The expansion and compression work depend on the timing of EVC and IVC respectively. Hence, valve lift timing is an optimization between; maximizing the amount of fresh air, minimizing the amount of burned gas left from the previous combustion, minimizing PMEP and maximizing IMEP.

In order to visualize some of the optimization difficulties a comparison between three different valve strategies will be made:

- Change of intake valve lift timing, keeping the exhaust valve lift fixed.
- Change of exhaust valve lift timing, keeping the intake valve lift fixed.
- Change of intake and exhaust valve lift timings, keeping the overlap period constant.

The results have been obtained by the use of a 1-D gas dynamic simulation code, GT-Power [3], which among other things enables a study of the mass flow through the intake and exhaust valves.
Figure 3 Effects of the change in intake valve lift timing (A) on P-V diagram (B) and mass flow (C) for 5000 rpm, full load. The grey curves correspond to the change in valve lift timing.

First of all, in Figure 3C the exhaust mass flow can be studied. The first exhaust pulse is called the blow-down pulse and the second the displacement pulse. The blow-down pulse is characterised by high enthalpy flow and peaks around BDC when the pressure difference between the cylinder and manifold is large. When the pressure in the cylinder decreases, the expulsion of exhaust gas is obtained by displacement from the piston.

Figure 3 shows the effects of phasing the intake valve. IVO and IVC occur 30 crank angle degrees later compared to the original case, see Figure 3A. As can be seen from the pressure versus volume diagram, Figure 3B, this results in a slight increase in PMEP. With later IVO timing the cylinder pressure initially decreases. As shown in Figure 3C the flow over the valve is quite small when the valve begins to open due to the ramping effects of the valve. Hence the
downwards motion of the piston creates a low pressure that cannot be compensated until the valve lift has increased and a greater inflow of air is possible. With the late timing of IVC the air and fuel mixture starts to be compressed when the intake valves are still open, which results in a back flow past the intake valves, as can be seen in Figure 3C.

In Figure 4 the effects of exhaust valve lift timing can be seen. The exhaust valve lift timing, as shown in Figure 4A, is advanced by 30 crank angle degrees. As can be seen in Figure 4B, this results in a decrease of PMEP. Due to the early timing of EVO, a large part of the exhaust pulse takes place before BDC, as can be seen in Figure 4C. This results in a more rapid decrease of the cylinder pressure when the exhaust valve opens, as can be seen in Figure 4B. On the other hand, the early timing of EVO has a negative effect on IMEP, since the
expansion work is interrupted at an earlier stage. During the overlap, an early timing of EVC has a slightly negative effect on PMEP. During the short overlap the cylinder gas is compressed by the piston, resulting in a pressure increase.

In Figure 5A both exhaust and intake valve lift timings are advanced while keeping the overlap constant. The early timing of EVO decreases IMEP and PMEP, the same affects as in Figure 4B. During the overlap period the pressure build-up that took place in Figure 4B is avoided. The back-flow over the intake valve, Figure 5C, increases when the overlap is increased compared to Figure 4C. This back-flow of exhaust into the intake system can be useful to reduce hydrocarbon emissions [5]. The last part of the exhaust can include high amounts of unburned hydrocarbons and when these are pushed into the intake
system and subsequently re-enters the cylinder and takes part in the following combustion event, the unburned hydrocarbon emissions can be reduced.

From the previous reasoning it can be concluded that valve lift phasing is not trivial, due to its combined effects on IMEP, PMEP, volumetric efficiency and hydrocarbon emissions. The optimization becomes even more complicated since the flow through the valves is dependent on engine speed. Conventional engines use fixed valve timings, i.e. the same opening and closing times independent of speed and load. This is inevitably a compromise between low and high-speed performance. Several variable valve lift timing systems have been developed, in order to overcome these problems.

2.4 Turbocharging

As mentioned previously, the inlet air density can be raised by the use of a compressor. The compressor needs to be supplied with work from an external source in order to increase the pressure of the gas. In a turbocharger the compressor is connected to a turbine through a rotating shaft. The basic idea behind the turbocharger is to drive the compressor with the power generated in the turbine. In turbocharged engines the turbine power is generated by the energy in the exhaust flow. Figure 6 gives a schematic view of the theories behind turbocharged engines.
Loop 1 to 4, in Figure 6 is the engine power stroke. The driving power of the turbine is the energy available at point 4 when the piston is at BDC. Area $A_T$ represents the available energy, which could be produced by expanding the exhaust gases to atmospheric pressure after the exhaust valves have opened. This energy is used to drive the compressor so that the intake pressure can be raised above the atmospheric pressure to the charging pressure, which requires work, area $A_p$. For the ideal case area $A_T$ equals area $A_p$. In other words, the power generated by the turbine will be used in the compressor. The power is related to the mass flow through the turbine, $\dot{m}$ and enthalpy, $h$, according to [6]:

$$P = \dot{m}(h_4 - h_3) = \dot{m}c_p(T_4 - T_3)$$ \hspace{1cm} \text{eq. 2-13}$$

where $T$ is the temperature and $c_p$ the specific heat. Figure 7, shows a schematic image of a turbocharger, with inlet and outlet conditions to the turbine and compressor.
Figure 7 A schematic image of a turbocharger with notation for inlet and outlet conditions, subscripts according to Figure 6.

Figure 8 An enthalpy–entropy diagram, where 7-8 is the compressor’s operation and 4-5 the turbine’s. s denotes the isentropic state.

Figure 8 shows the turbine and compressor operation in an enthalpy-entropy diagram. As illustrated by Figure 8, the basic idea is to compress the intake air to a higher pressure, moving from point 7 to 8. The turbine, on the other hand, expands the exhaust gas from point 4 to 5 in Figure 8.

Ideally the expansion and compression would be isentropic. However, due to mainly heat and flow losses the turbine and compressor efficiency, $\eta_T$ and $\eta_C$ are defined. They describe the relationship between the actual work and the isentropic work generated or required for an operation between the same pressures as the actual work transfer:

$$\eta_C = \frac{\text{isentropic work}}{\text{actual work}} \quad \text{eq. 2-14}$$

$$\eta_T = \frac{\text{actual work}}{\text{isentropic work}} \quad \text{eq. 2-15}$$

For isentropic compression and expansion the following is true:
\[ \frac{P_{in}}{P_{out}} = \left( \frac{T_{in}}{T_{out}} \right)^{\gamma - 1} \]  

eq. 2-16

where \( \gamma \) is the specific heat ratio. By utilising eq. 2-13, 2-14, 2-15 and 2-16 the power for the compressor and turbine can be expressed as follows:

\[-P_c = \frac{1}{\eta_C} \dot{m}c_p T_i \left( 1 - \left( \frac{P_8}{P_7} \right)^{\gamma - 1} \right) \]  

eq. 2-17

\[ P_T = \eta_T \dot{m}c_p T_i \left( 1 - \left( \frac{P_3}{P_4} \right)^{\gamma - 1} \right) \]  

eq. 2-18

where the subscripts correspond to the numbers in Figure 7. The power generation in a turbine can also be related to the turbine torque:

\[ P_T = \omega M \]  

eq. 2-19

where \( \omega \) is the rotational speed of the turbine wheel and \( M \) the torque.

The choice of turbine for a specific engine configuration is a compromise between low and high load performance of the engine. For commercial automotive purposes low load performance is most often preferred. In order to ensure sufficient pressure at the turbine inlet at low load, the inlet area to the turbine needs to be limited. However at high load a small inlet area will generate too high rotational speed of the turbine. Hence, it is necessary to limit the turbine rotational speed during high load due to mechanical constraints. This can be done by a wastegate, which is equivalent to a valve that leads part of the flow past the turbine, thereby decreasing the pressure at the turbine inlet.
2.5 Combustion

Transforming fuel to energy by combustion is of course an essential part of engine operation. Making this process as efficient as possible is a major area for development work. The actual combustion event is influenced by preceding as well as subsequent events.

In a port-injected engine the injection of fuel starts before the intake valves have opened. The fuel is injected towards the hot valves, ensuring fast evaporation. As the intake valves open a mixture of fuel vapours, droplets and air is sucked into the cylinder. The intake valves, ports and cylinder geometry are designed to enforce a certain motion and turbulence level of the flow in the cylinder. The actual combustion event can be characterized by three different phases, influenced by different factors [4]:

- **Flame development**: The time between spark discharge and when a small, detectable mass of charge has burned, often 10% of the mass.
- **Rapid burning**: The time it takes to burn most of the combustible mass, usually defined between 10% and 90% burned mass.
- **Overall burning**: The entire time for the burning process to take place.

Turbulence is an important factor for efficient combustion because it affects the mixing of air, fuel and residual gases. Residual gas refers to the gas left in the cylinder from the previous combustion. Hence, it is clear that local variations in mixture composition will exist throughout the combustion chamber. These local variations play an important role in flame development. The flame development, as defined previously, is primarily influenced by the mixture state, motion and composition near the spark plug. The following flame propagation is less influenced by local variations in mixture, at this stage the overall composition play a greater role. However, local variations of mixture composition between different cycles are one of the main causes of cycle-to-cycle variations. Three
main causes of these variations have been defined [4]: Local variation in mixture composition, mixture motion and air-fuel ratio.

\[\text{Figure 9 Cylinder pressure as a function of crank angle for nine different cycles at 1500 rpm.}\]

As can be seen from Figure 9, the cycle-to-cycle variations can be quite severe. Since engine operation will be limited by the worst cycle it is important to keep the variations to a minimum.
Chapter 3 Methodology

As mentioned previously, an aim of this thesis is to investigate a gas exchange system by predictive 1-D flow simulations. This chapter will give some general information on 1-D flow simulations and specific information on some simulation areas in need of special attention. The need for experimental data for validation purposes will also be discussed.

3.1 1-D engine simulation

Engine performance can be studied by analyzing the mass and energy flows between individual engine components and the heat and work transfers within each component. Figure 10 shows a simplified model of an engine, describing the interaction between different components.
Figure 10 Simplified turbocharged engine model, where C stands for the compressor and T for the turbine.

In the actual simulation model each pipe, bend etc needs to be represented. In Figure 11 the GT-Power model from the throttle to the cylinders is depicted. Each pipe element contains information of dimensions, surface roughness, temperature etc.
Figure 11 A view of the GT-Power model from throttle to cylinder.

In order to describe the entire engine a variety of input data is needed. Table 1 gives some examples of the required data.

Table 1 Summary of input data to a 1-D engine model

<table>
<thead>
<tr>
<th>Engine part</th>
<th>Type of information</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine characteristics</td>
<td>compression ratio, firing order</td>
</tr>
<tr>
<td>Cylinder geometry</td>
<td>bore, stroke, connecting rod length</td>
</tr>
<tr>
<td>Intake and exhaust system</td>
<td>geometry of manifolds</td>
</tr>
<tr>
<td>Throttles</td>
<td>location, discharge coefficients</td>
</tr>
<tr>
<td>Fuel injectors</td>
<td>location, fuel/air ratio</td>
</tr>
<tr>
<td>Intake and exhaust valves</td>
<td>valve diameter, lift profile, discharge coefficients</td>
</tr>
<tr>
<td>Turbocharger</td>
<td>performance maps</td>
</tr>
<tr>
<td>Ambient</td>
<td>pressure, temperature and humidity</td>
</tr>
</tbody>
</table>

Simulation of 1-D flow involves the solution of the conservation equations; mass-, energy- and momentum, in the direction of the mean flow [7]. Mass
conservation states that the rate of change in mass within a sub system is equal to the sum of mass flowing into and out from the system:

\[
\frac{dm}{dt} = \sum_i \dot{m}_i - \sum_e \dot{m}_e \quad \text{eq. 3-1}
\]

where subscript \( i \) denotes inlet and \( e \) exit. In 1-D flow the mass flow rate, \( \dot{m} \), is defined by:

\[
\dot{m} = \rho AU \quad \text{eq. 3-2}
\]

where \( \rho \) is the density, \( A \) the cross-sectional flow area and \( U \) the fluid velocity.

Energy conservation states that the rate of change of energy in a sub system is equal to the sum of energy transfer in and out of the system. The means of energy transfer are work, the energy accompanying the mass flow and heat:

\[
\frac{d(me)}{dt} = -\rho \frac{dV}{dt} + \sum_i \dot{m}_i H - \sum_e \dot{m}_e H - h_g A(T_{gas} - T_{wall}) + \frac{dQ_{ch}}{dt} \quad \text{eq. 3-3}
\]

where \( e \) is the internal energy, \( H \) the total enthalpy, \( h_g \) the heat transfer coefficient, \( T_{gas} \) and \( T_{wall} \) the temperature of the gas and wall respectively and \( Q_{ch} \) the energy released during combustion. The heat transfer from internal fluids to pipe and flowsplit walls is dependent on the heat transfer coefficient, the predicted fluid temperature and the internal wall temperature. The heat transfer coefficient, which is calculated at every time step, is a function of fluid velocity, thermo-physical properties and the wall surface finish [8]. The internal wall temperature is defined by the user.

Momentum conservation states that the net pressure forces and wall shear forces acting on a sub system are equal to the rate of change of momentum in the system plus the net flow of momentum out of the system:
\[
\frac{dm}{dt} = \frac{dpA + \sum_i m_i \dot{u} - \sum \dot{m}_p - 4C_f \rho u^2 dxA}{dx} - C_p \frac{1}{2} \rho u^2 A
\]  

eq. 3-4

where \( u \) is fluid velocity, \( C_f \) the friction loss coefficient, \( D \) the equivalent diameter, \( C_p \) the pressure loss coefficient and \( dx \) the element length. In order to obtain the correct pressure and friction loss coefficients the software uses empirical correlations to account for pipe curvature, surface roughness etc [8].

The whole system is discretized into smaller volumes, where a flow split is represented by a single volume and pipes are divided into several volumes. Scalar properties, such as temperature, pressure, density etc are assumed to be uniform for each volume. Accuracy improvements may be made by choosing finer discretization length. However, the discretization length is limited by the size of the time step. The time integration of the governing equations is explicit. The time step during this integration is limited by the Courant condition, which restricts the time step to be less than 0.8 of the time required for the pressure and flow to propagate across any discretized volume:

\[
\frac{\Delta t}{\Delta x} (u + c) \leq 0.8
\]  

eq. 3-5

where \( \Delta t \) is the time step, \( \Delta x \) the discretized length, \( u \) the fluid velocity and \( c \) the speed of sound. Due to the link between discretization length and time step, a small discretization length will generally result in slow execution by the software.

### 3.1.1 Combustion simulation

In order to simulate the engine system presented in Figure 10, the amount of chemical energy released during combustion, \( Q_{ch} \), needs to be simulated. By expressing the change in internal energy with thermodynamic relationships and neglecting the crevice effects, the heat release rate can be expressed as a function of cylinder pressure:
\[
\frac{dQ_{\text{th}}}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} + A h_c (T - T_w) \tag{eq. 3-6}
\]

where \( A \) is the combustion chamber surface area and \( h_c \) the heat transfer coefficient. The heat transfer coefficient is estimated according to a modified Woschni’s correlation \([8]\) \([4]\).

The deduced heat release rate can be expressed in a parametric form suggested by Wiebe \([9]\):

\[
x_b = 1 - \exp \left[ -a \left( \frac{\theta - \theta_b}{\Delta \theta} \right)^{m+1} \right] \tag{eq. 3-7}
\]

where \( x_b \) is the percentage of burned mass at crank angle \( \theta \), \( \theta_b \) is the crank angle at start of combustion, \( \Delta \theta \) the total combustion duration, \( a \) can be expressed as a function of the defined combustion duration and \( m \) is an adjustable parameter. The parameters \( \theta_b \), \( \Delta \theta \) and \( m \) can be adjusted so that the Wiebe function, eq. 3-7, resembles the accumulated heat release rate, which is based on measured cylinder pressure. In Figure 12 the effects of changing \( m \) and \( \Delta \theta \) can be seen.
Figure 12 The effect on the Wiebe curve from changing the parameters $m$ and $\Delta \theta$ with constant $\theta_0$ at 15° BTDC. Dotted line depicts $m=3$ and $\Delta \theta = 25^\circ$.

Figure 12a shows that increasing $m$ from 2 to 3 yields a steeper heat release and a long flame development time. Changing the combustion duration, $\Delta \theta$, affects the timing of combustion end, as shown in Figure 12b.

The heat release or Wiebe function are used as input data to the simulation model. Thus, cylinder pressure needs to be measured or in some way estimated, in order to calculate the heat release. The implication is that 1-D simulation using a heat release analysis for combustion simulation can never become truly predictive.

### 3.1.2 Turbocharger simulation

Turbocharger simulations are based on performance maps for the compressor and turbine. These performance maps are based on flow-bench measurements on the turbine and compressor [6]. The measurement data is then interpolated and extrapolated in order to cover the turbocharger operation range. In Figure 13 and Figure 14 the turbine and compressor maps are expressed as functions of reduced mass flow and speed, which removes the effects of the inlet conditions:
Chapter 3 - Methodology

Reduced mass flow: \[ \frac{\dot{m}_{\text{actual}} \sqrt{T_{\text{inlet}}}}{p_{\text{inlet}}} \] eq. 3-8

Reduced speed: \[ \frac{N_{\text{actual}}}{\sqrt{T_{\text{inlet}}}} \] eq. 3-9

where \( N \) is the turbine or compressor speed.

Figure 13 Pressure ratios for the compressor versus reduced mass flow rates. Equal colours are areas of constant reduced speed and efficiency respectively.

Figure 14 Reduced mass flow rates versus pressure ratios for the turbine. Equal colours are areas of constant reduced speed and efficiency respectively.

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Figure 13 and Figure 14 show typical performance maps for the compressor and turbine. The simulation procedure of the turbocharger will follow an iterative scheme:

1. The pressure ratio over the turbine and compressor are determined from the adjacent sub volumes immediately upstream and downstream of the compressor [8].
2. The turbine and compressor speed, which are the same, is known from the previous time step or the initial value.
3. With pressure ratio and turbine/compressor speed information, the mass flow and efficiency information can be found in the performance maps.
4. The power of the turbine and compressor can be calculated according to eq. 2-17 and 2-18. From the power, the torque can be calculated according to eq. 2-19.
5. Any torque imbalance between the turbine and compressor will lead to a change in turbine/compressor speed according to eq. 3-10:

\[
\Delta \omega = \frac{\Delta t}{I_{\text{rotor}}} \left( M_{\text{turbine}} - M_{\text{compressor}} - M_{\text{friction}} \right) \quad \text{eq. 3-10}
\]

where \( \Delta t \) is the calculation time step and \( I_{\text{rotor}} \) the inertia of the turbocharger rotor.
6. If the change in turbine/compressor speed exceeds the convergence criteria the calculation procedure will restart at 1.

Several areas of difficulty are involved in this handling of the turbine and compressor. First of all the measurements, on which the turbine and compressor maps are based, are performed with steady flow, although the actual flow in an engine is highly pulsating. Secondly, since interpolation and extrapolation is necessary in order to obtain the full maps. The effect of these simplifications can be studied in literature by Westin [10].
3.2 Experimental method

As mentioned previously, simulation models are highly dependent on experimental data as input as well as for model validation. Since the aim was to look at a new gas exchange system where the effects on for example combustion, mass flow and turbine performance were unknown, a prototype engine was built. The prototype engine used for Paper II was based on a standard port fuel injected turbocharged engine with a modified cylinder head. The same engine but with the standard cylinder head was used for Paper III. In Table 2 some specifics of the engines can be found.

<table>
<thead>
<tr>
<th>Table 2 Engine specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
</tr>
<tr>
<td>Stroke</td>
</tr>
<tr>
<td>Compression ratio</td>
</tr>
<tr>
<td>Number of valves</td>
</tr>
<tr>
<td>Total displacement</td>
</tr>
<tr>
<td>Cylinder head</td>
</tr>
<tr>
<td>Inlet valve diameter</td>
</tr>
<tr>
<td>Exhaust valve diameter,</td>
</tr>
<tr>
<td>standard head</td>
</tr>
<tr>
<td>Exhaust valve diameter,</td>
</tr>
<tr>
<td>prototype head</td>
</tr>
</tbody>
</table>

The level of model validation is dependent on what the model should be used for. In Paper I, a validation scheme for turbocharged SI engine is discussed. The scheme is presented in order of importance and will be presented here with information on how the measurements were performed, for more information about the measurements the reader is referred to Lindström [11].
1. **Average air mass flow rate** was estimated from fuel mass flow and air/fuel ratio:

\[
\dot{m}_{a,\text{actual}} = \lambda \dot{m}_{f,\text{actual}} \left( \frac{\dot{m}_a}{\dot{m}_f} \right)_{\text{stoichiometric}}
\]

where \( \lambda \) is the relationship between actual air/fuel ratio and the stoichiometric ratio, \( \left( \frac{\dot{m}_a}{\dot{m}_f} \right)_{\text{stoichiometric}} \) is the stoichiometric relation between air and fuel mass, specific for the fuel used.

2. **Air/fuel ratio** was measured with an ECM AFRecorder 2000A with an accuracy of \( \pm 0.008 \) for \( 0.8 < \lambda < 1.2 \) [12].

3. **IMEP** is used to check the model’s ability to predict engine output. It is calculated from measured cylinder pressure, eq. 2-2 and 2-3.

4. **Cylinder pressure trace**, a good match to measured data is necessary in order to correctly model IMEP, exhaust gas temperatures and pressure wave propagation in the exhaust and intake manifolds. Cylinder pressure was measured in all cylinders with nearly flush mounted AVL GM12D uncooled miniature piezoelectric transducers [13] and Kistler 5011 charge amplifiers [14]. The sampling rate used was 0.2 to 0.4 crank angle degrees. At least 200 cycles were recorded for each operating condition.

5. **Turbocharger speed** is crucial in order to find the correct compressor operation point, thereby modelling the mass flow rate and intake temperature correctly. It was measured with a Micro-Epsilon eddy current probe [15].

6. **Turbine inlet pressure trace** is needed in order to predict turbine power. It was measured with Kistler 4045A10 piezoresistive pressure transducers [14].

7. **Inlet manifold pressure trace** is used to check that the pressure pulsation in the inlet manifold is modelled correctly. These pressure measurements were performed with GEMS steel diaphragm gauge pressure transducers [16].

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8. **Inlet manifold average temperature;** a good fit to measured data ensures that the engine is fed with air of correct density. It was measured at several locations with shielded 3 mm type K thermocouples. The accuracy of class 1 type K thermocouples is the larger of $1.5^\circ$ and $0.004$ times the measured temperature [17].

9. **Turbine inlet average temperature** is needed for turbine power predictions. It was measured with similar equipment as above.

10. **Compressor outlet temperature** indicates if the compressor model predicts the correct efficiency. It was measured with similar thermocouples as previous temperature measurements.

11. **Turbine outlet pressure trace** is important in order to find the correct pressure ratio over the turbine. It was measured with similar equipment as the turbine inlet pressure trace.

Modelled quantities should be adjusted to be within the measurement range. Hence no model can be more accurate than the accuracy of the input and validation data.

### 3.3 Model validation

A simulation model can be validated at different levels; cycle averaged and crank angle resolved. The extent of validation depends on what the model will be used for. When studying gas exchange and flow pulsation it is necessary to validate the simulation model with crank angle resolved data. A comparison for measured and simulated IMEP and BMEP is found in Figure 15.
BMEP is set as the target for the simulations, hence no discrepancy between measured and simulated values exist. The slight inaccuracy in IMEP is due to the uncertainty in FMEP modelling, see eq. 2-9.

As mentioned it can also be important to validate the model to crank angle based data. In Figure 16 the simulated crank angle resolved pressure in the blow-down and scavenging systems, these are the two exhaust manifolds of the Divided Exhaust Period engine (see Figure 17), are compared to measured data. The valve lift data for the prototype engine are also shown.
Figure 16 Comparison between simulation and measurements; pressure in blow-down and scavenging system at 5500 rpm.

The model correlates fairly well with measurements and captures the pressure pulsation in the exhaust manifolds as can be seen in Figure 16.
Chapter 4 Divided Exhaust Period

This chapter summarizes Paper II. It starts with explaining the Divided Exhaust Period concept and finishes by presenting some results.

4.1 Divided Exhaust Period concept

In a patent from 1924 an alternative gas exchange concept for turbocharged engines was presented [18]. The Divided Exhaust Period (DEP) concept was developed to test the theories behind this concept and evaluate its usefulness for modern turbocharged engines. The basic idea with the DEP concept is to avoid the high pressure in the exhaust manifold created by the turbocharger. This is obtained by splitting the two exhaust ports from each cylinder into two separate exhaust manifolds.
Figure 17 Schematic image of the DEP engine.

Figure 17 shows a schematic image of the DEP engine. The Exhaust blow-down system is connected to the turbine while the Exhaust scavenging system by-pass the turbine and lead the exhaust directly to a close coupled catalyst. A trapping valve is introduced downstream the close coupled catalyst, in order to control the pressure in the system and thereby control the amount of fresh charge flowing directly from the intake to the exhaust. However, the physical splitting of the valves is not the only difference from a standard engine. Figure 18 show the valve lift curves for the DEP engine.
Figure 18 Valve lift profiles for the DEP engine.

The exhaust valve lifts are separated in phasing and lift profile. The reasoning behind the separation of the exhaust mass flow can be explained by studying the exhaust mass flow rate as a function of crank angle degrees for a standard turbocharged engine, see Figure 19.

Figure 19 Simulated normalized cumulative mass and mass flow rate for a turbocharged engine at 5500 rpm.
From Figure 19 it can be concluded that at 270 crank angle degrees, when the blow-down pulse has been replaced by the displacement pulse, approximately 70% of the total mass flow has passed through the exhaust valve. In the DEP concept the idea is to separate the two exhaust pulses into separated manifolds. The enthalpy rich blow-down pulse is used in the turbine while the displacement pulse is evacuated into the scavenging system. By evacuating into a system with a lower pressure, as the scavenging system in the DEP concept, the positive pressure difference between cylinder and exhaust manifold should theoretically be maintained. This would result in improved PMEP values.

The positive pressure difference over the engine, obtained with the DEP concept, has several other benefits as well. The cylinder evacuation is improved; hence the volumetric efficiency increases and knock resistance improves when the amount of residual gas decreases [19].

Cylinder evacuation is also affected by pulse interaction in the exhaust manifold. In a standard four cylinder turbocharged engine with a single turbine, the blow-down pulse from one cylinder is reflected and can interfere with the previous cylinder in firing order during its exhaust-intake overlap. In the DEP concept however this can be avoided by only allowing Exhaust blow-down valve timings slightly longer than 180 crank angle degrees.

Furthermore, the DEP concept can benefit from having a close coupled catalyst. In turbocharged engines the turbine acts as a heat sink, hence delaying the time for when the catalyst has reached its working temperature. By not opening the Exhaust blow-down valve during cold start, so that all the exhaust passes directly to the closed coupled catalyst, a decrease of emissions will be obtained.

4.2 Results

The effects of changing the valve lift phasing for the two exhaust valves were investigated both with measurements and simulations. The valve lift profiles tested on the prototype engine are shown in Figure 20.
By changing the EVO timing of the Exhaust scavenging valve, see Figure 20 the amount of exhaust directed to the turbine was varied. A critical aspect of the Divided Exhaust Period concept is the potential for PMEP improvements. Figure 21 shows PMEP as a function of speed and valve lift profile.

As can be seen from Figure 21 the DEP engine exhibits negative PMEP, that is positive work according to eq. 2-4, over an extended speed range compared to a standard turbocharged engine. In addition, high speed positive PMEP is reduced. 
compared to the standard turbocharged engine. Advancing the scavenging exhaust valve opening 15 crank angle degrees, denoted early scavenging 1 in Figure 20, extends the range of positive pumping work and decreases high speed pumping losses even further compared to the DEP reference camshaft. A very early scavenging exhaust valve opening, denoted early scavenging 2 in Figure 20, results in pumping work similar to the standard turbocharged engine at low speed. However at high engine speeds the pumping losses are significantly reduced. The power improvement from the reduced pumping losses in the DEP engine is up to 10 kW at 5000 rpm, which is about 6% of the engine rated power.

Figure 22 Obtained brake mean effective pressure with different exhaust valve lift curves compared to the standard engine. The solid markers indicate closed trapping valve in the scavenging exhaust system.

Looking at BMEP obtained for the tested DEP engine, see Figure 22, reveals that the DEP engine reached slightly higher power than the standard engine in the tests. However, low speed BMEP was in fact below the standard engine when the trapping valve, see Figure 17, was open. The reason for the low BMEP can be found in the DEP concept. At low engine speeds all of the energy in the exhaust has to be transferred via the turbine to the compressor in order to reach
the boost pressure target. However, with the DEP concept a part of the exhaust is bypassed the turbine and as a consequence it is difficult to reach the boost target. With the DEP engine concept it would be possible to use a smaller turbine without suffering from increased residuals, as the amount of residual gas is not directly related to the exhaust pressure before the turbine. Even with a considerably smaller turbine it would be very difficult to reach the target with some of the exhaust energy lost through the scavenging system. A thorough matching of the turbocharger was not carried out and the tests were performed with a standard turbocharger with a smaller inlet diameter.

![Graph showing comparison between DEP engine and standard turbocharged engine residual gas content at IVC and DEP improvement over speed range.](image)

**Figure 23** Comparison simulated values of residual gas content in the cylinder for the DEP engine versus the standard turbocharged engine.

In Figure 23 simulated residual gas content for a standard turbocharged engine is compared to the DEP engine. An improvement can be found over the entire speed range. The improvement is as much as 60% at 2000 rpm, which implies improvements in knock resistance. However, a refined method to study knock in 1-D engine simulations would improve the simulation capabilities. Such a method is presented in the next chapter.
Chapter 5 0-D modelling of knock

This chapter summarizes Paper III. First, an introduction to knock is presented, which is followed by a presentation of the 0-D knock model and finally some results.

5.1 Introduction to knock

SI engines, especially turbocharged ones, suffer from an undesired combustion phenomenon called knock. The term knock originates from the noise created by pressure oscillations originating from spontaneous ignition of the unburned mixture, the so called “end-gas”, ahead of the advancing flame front [4].

![Figure 24 Autoignition centre in the combustion chamber.](image)

As the end-gas is compressed by the piston and the propagating flame front, the pressure and temperature increase. In some cases the pressure and temperature combination can lead to autoignition, centred round one or more points in the
end-gas, see Figure 24. The autoignition triggers very fast combustion which can lead to pressure oscillations in the combustion chamber. These pressure oscillations can be severe enough to cause major engine failure. The severity of the pressure oscillations depends, among other things, on the location of the autoignition points and the amount and rate of the heat release. The location of the autoignition points depend on local inhomogeneities in composition and temperature [20].

5.2 Knock model

Due to the potentially damaging effects, it is clearly important to consider knock not only in an experimental but also in a simulation environment. Simulations that are performed with a combustion, which in reality would knock, are not of much use. In the experimental environment, knock can be detected by cylinder pressure transducers. These pressure transducers detect the pressure oscillations caused by autoignition. Figure 25 is an example of how knock can be detected by measuring the cylinder pressure [21].

![Figure 25 Cylinder pressure from a knocking cycle and the filtered cylinder pressure.](image)

Figure 25 Cylinder pressure from a knocking cycle and the filtered cylinder pressure.
In order to compare different knocking cycles two definitions were made. Knock onset, KO, gives the crank angle degrees for when knock starts to occur; knock index, KI, give a value of the strength of the pressure oscillations.

In the simulation environment knock is not so easily detected. It is a stochastic phenomenon highly dependent on local composition in the end-gas. However, in order to perform simulations that are not only a copy of an experiment a method to simulate knock is desirable. An empirical model based on the theories of Livengood and Wu [22] was implemented in the 1-D software model. Livengood and Wu proposed that autoignition occurs when:

\[
\int_{t_{\text{start}}}^{t_{\text{auto}}} \frac{1}{\tau} \, dt = 1 \quad \text{eq. 5-1}
\]

where \( \tau \) represents the ignition delay time and \( t \) is the elapsed time between start of compression to autoignition. The ignition delay time, \( \tau \), can be determined by fitting experimental data to an Arrhenius type function:

\[
\tau = Ap^{-n} \exp \left( \frac{B}{T_u} \right) \quad \text{eq. 5-2}
\]

where \( p \) is cylinder pressure, \( T_u \) the unburned zone temperature and \( A, n \) and \( B \) constants. In order to find values for \( A, n \) and \( B \) the error of the Livengood-Wu integral at measured knock onset was minimized by using the least squares method as suggested by Douad and Eyzat [23]. eq. 5-3 gives the equation to be minimized:
where $k$ is the number of individual cycles. In Paper III only $A$ was optimised and the values of $n$ and $B$ were set to 1.7 and 3800 respectively, according to the recommendations of Douad and Eyzat [23]. The optimization was performed using pressures and KO values from measurements and temperatures of the unburned gas from the engine simulation. In later studies [24] [25], optimization of the three constants was performed, the results are presented in Table 3.

*Table 3 Summary of values for $A$, $n$ and $B$.*

<table>
<thead>
<tr>
<th></th>
<th>$A$</th>
<th>$n$</th>
<th>$B$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Values used in Paper III</td>
<td>0.021</td>
<td>1.7</td>
<td>3800</td>
</tr>
<tr>
<td>New optimization</td>
<td>0.0071</td>
<td>1.325</td>
<td>3296</td>
</tr>
</tbody>
</table>

The knock model is integrated into the engine model, schematically shown in Figure 26.
Chapter 5 - 0-D modelling of knock

In the engine model the combustion event is described by a heat release or Wiebe analysis, see eq. 3-6 and 3-7. Crank angle based information of the cylinder pressure, unburned zone temperature and percentage burned mass are sent to the knock model. The cylinder pressure and unburned zone temperature are used to calculate the Livengood and Wu integral, eq. 5-1.

The unburned zone temperature is obtained through simulations using a two zone combustion approach. In a two zone analysis of the combustion event the gas in the combustion chamber is split into unburned and burned gas. Prior to combustion the pressure, temperature and density of the gas mixture change due to compression by the piston. So far the gas mixture can be considered homogenous. As soon as the spark is discharged and combustion starts taking place, a difference of state between the burned and unburned mixture occur. The density of the burned gas decreases and the following gas expansion compresses the unburned gas mixture ahead of the moving flame front. By setting up the equations for conservation of mass and energy as well as considering the thermodynamic properties for the unburned and burned zones and considering

Figure 26 Schematic image of the interaction between the engine and knock model.

In the engine model the combustion event is described by a heat release or Wiebe analysis, see eq. 3-6 and 3-7. Crank angle based information of the cylinder pressure, unburned zone temperature and percentage burned mass are sent to the knock model. The cylinder pressure and unburned zone temperature are used to calculate the Livengood and Wu integral, eq. 5-1.
the pressure to be uniform across the combustion chamber [4], the state in each zone can be evaluated separately.

![Figure 27 Burned and unburned zone temperature and mass fraction burned as a function of crank angle degrees.](image)

In Figure 27 the temperature of the unburned and burned zones as well as the ratio of burned mass can be studied as a function of crank angle degrees. The burned temperature increase drastically as combustion starts. The unburned zone temperature has a comparably moderate temperature increase due to compression by piston and flame front. Maximum temperature for the burned and unburned zone can be found around the timing for 50 % burned mass. The following decrease in temperature is a result of the decrease in heat release rate, due to the diminishing amount of burnable mass and the expansion caused by the piston movement.

The Livengood and Wu integral will keep increasing unless some additional constraints are set for the integral. Cowart et. al. [26] proposes that unburned mass at knock onset is about 10 to 20%. Hence, it was found appropriate to add the restriction that the integral must become equal to one before approximately 90% of the mass is burned for the simulation to be considered operating at the knock limit. The knock model will then give the user information about crank angle degrees for KO and percentage burned mass at KO.
5.3 Results

The knock model was tested for a variety of speeds and $\lambda$. In Table 4 the results have been summarized, showing that the average difference between simulated and measured KO was at worst 2.4 crank angle degrees.

Table 4 Measured and simulated knock onset for different operating conditions.

<table>
<thead>
<tr>
<th>Case [rpm]</th>
<th>2500</th>
<th>3000, lean</th>
<th>3000, stoich.</th>
<th>3000, rich</th>
<th>3500</th>
</tr>
</thead>
<tbody>
<tr>
<td>Measured average KO</td>
<td>14.7</td>
<td>13.8</td>
<td>14.8</td>
<td>12.3</td>
<td>8.6</td>
</tr>
<tr>
<td>Simulated average KO</td>
<td>14.5</td>
<td>13.5</td>
<td>17.3</td>
<td>12.2</td>
<td>8.9</td>
</tr>
<tr>
<td>Difference</td>
<td>0.16</td>
<td>0.28</td>
<td>-2.42</td>
<td>-0.08</td>
<td>-0.32</td>
</tr>
</tbody>
</table>

The accuracy presented in Table 4 is very good considering the accuracy of the combustion simulation and KO measurements.
Chapter 6 Conclusions

1-D simulation is a powerful tool for engine development. It is a cost and time efficient development tool. Perhaps of greater importance is that information, which can be hard to acquire through measurements, can be given by the simulation model. However, as discussed in this thesis 1-D simulation models require additional information from measurements or estimations. 1-D simulations will never be truly predictive as long as the combustion model relies on measured data. However, qualified estimations can be made for known combustion systems.

Studies of unknown systems such as the Divided Exhaust Period concept are more dependent on measurement data, since the concept affects the combustion extensively by reducing residual gases. Through measurements and simulations it can be shown that the DEP concept succeeds in reducing some of the negative aspects of turbocharged engines. The residual gases as well as the pumping losses can be decreased. The decrease in residual gas content will improve knock resistance. The pulse interaction is avoided by splitting the exhaust flow.

Simulations of unknown systems, where qualified estimations of the combustion needs to be made are limited by, among other things, knock. These pressure oscillations cannot be modeled in a 1-D simulation code. In this thesis a method for distinguishing between non- and knocking cycles as well as detecting the
angle for knock onset is developed. The empirical knock model is able to predict knock onset by approximately two crank angles. Due to the difficulty in measuring knock onset, this is seen as a very good result. The information given by this knock model will aid the user in making qualified assessment of the severity of knock and whether the specific operation will be possible in reality or not.

In conclusion this thesis presents a scheme for validating 1-D simulation models, how these models can be utilized in engine development, in this case the Divided Exhaust Period concept and an empirical knock model, which will improve the modeling capabilities of 1-D softwares.
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[18] British patent no. 12,227/22; Improvements in or relating to Internal Combustion Engines.


