Limits of mixture dilution in gas engines

PROEFSCHRIFT

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Voor mijn vader,

die het verschijnen van dit boekje niet meer meemaken mocht
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Summary

Natural gas engines find application in transport as well as for stationary power generation. These engines have a lower efficiency compared to the most widely used power plant, the diesel engine, however engines running on natural gas also have some distinct advantages. Gas engines that are optimized to the same level as their diesel counterparts can achieve lower specific $CO_2$ emissions due to the higher $H/C$ ratio of the fuel. This is of great interest since $CO_2$ is a greenhouse gas. A further motivation for the use of natural gas as a fuel is the need for diversification of fuel supply. Heavy duty gas engines in particular are predominantly used with a fuel lean mixture. This mixture dilution can be done with either excess air, these are called ”lean burn engines” or with recycled exhaust gases, called ”lambda 1 + EGR engines”. When using mixture dilution at low engine load, engine efficiency is increased by reducing throttling losses. Due to the mixture dilution, peak combustion temperatures decrease and the formation of $NO$ is reduced.

Mixture dilution however can only be applied to a certain extent. When the mixture is diluted excessively, the emission of unburned hydrocarbons increases sharply due to engine misfires, partial burns and other combustion instabilities. When the mixture is diluted, its burning velocity decreases. Lean burn engines therefore typically apply a combustion system (combustion chamber shape) which increases turbulence to boost the overall burning rate. The generally accepted ‘model’ of the combustion process is one where flames locally propagate with the laminar burning velocity and where the overall burning rate can be found by multiplying the flame area with the burning velocity.

Developing dedicated gas engines with help of engine simulation codes (e.g. GT-power, AVL Boost, Ricardo Wave and others) requires knowledge on the flame propagation process to predict the burning rate or heat release. Generally a basic assumption of a spherical flame front which propagates outward from the spark plug is used. The propagation speed of the flame front (relative to the unburned mixture) or turbulent burning velocity, $S_T$, in that case is taken proportional to the turbulent velocity fluctuation which, in turn, is assumed to scale with engine speed. The underlying assumption here is again that combustion occurs in flamelets that locally propagate with the laminar burning velocity and that flame surface is created by turbulent velocity fluctuations. For lean mixtures however this assumption is less validated. The predictive capabilities of engine simulation codes for lean burn engines are therefore less.

In the study presented in this thesis, the limits of lean combustion in a heavy-duty gas engine were explored with the aim of increasing understanding of the combustion process of these lean mixtures. This knowledge can then be used to improve the predictive capabilities of engine modeling codes. The questions to be answered here were

- Does the turbulent burning velocity still scale with engine speed for very lean mixtures?
• Is the combustion regime that is applicable for very lean mixtures is still 'classical' flame propagation?

• Is the assumption of spherical propagation justified for lean mixtures in gas engines?

The investigation was split in a number of separate parts which targeted different aspects of the combustion process and its interaction with the flow phenomena. A single cylinder test engine featuring optical access to the combustion chamber was designed and constructed for this purpose. Engine tests in the same engine, only with the transparent parts replaced by full metal ones, were conducted to explore the limits of its operating range. The heat release and mass burning rate were determined from in-cylinder pressure measurements. The turbulent burning velocity $S_T$ was determined using the classical assumption of a spherically propagating flame front. The laminar burning velocity $S_L$ for the same mixtures was measured using the technique of constant volume combustion in the EHPC (Eindhoven High Pressure Cell). This resulted in a correlation for the laminar burning velocity $S_L$ as function of pressure, temperature and mixture equivalence ratio. A similar, but less extensive correlation was determined for mixtures diluted with inert components. The diluent in this case was a mixture of nitrogen and carbon dioxide with similar thermal properties as recycled exhaust gas for a stoichiometric gas engine. These correlations were used to determine the ratio of turbulent to laminar burning velocity for various engine operating conditions. It was shown that the burning velocity ratio for lean mixtures scales approximately with engine speed, confirming the general model.

To evaluate the applicable combustion regime for lean mixtures, both the process of flame propagation and also the properties of the turbulent flow inside the engine cylinder were investigated. The applicable combustion regime can be derived from typical velocities and spatial scales of the combustion process. The flow inside the engine cylinder was measured using Particle Image Velocimetry. To enable this the engine was equipped with a window in the piston crown and also the upper part of the engine liner was replaced by a transparent piece. The turbulent velocity fluctuations were determined from ensembles of images recorded at the same crank position but in different engine cycles. The cycle-to-cycle fluctuation was separated from the in-cycle turbulent fluctuations using a spatial filtering technique. Spatial autocorrelation functions were used to determine the integral length scale of the flow. The applicable combustion regime was demonstrated in a Borghi diagram that was constructed from the laminar burning velocity $S_L$, the turbulent velocity fluctuation $u'$, the integral length scale of the flow and an estimate for the flame thickness. The combustion in this engine was typically found in the 'corrugated flamelets regime' for stoichiometric mixtures. For the combustion process of very lean mixtures the 'thin reaction zones regime' was more applicable. This finding confirmed that combustion in a gas engine for these lean mixtures still occurs in the form of 'classical' flame propagation. The experiments with optical access were then extended with visualizations of the combustion process using Mie scattering on the same oil droplets that were used for the velocity measurements. Since oil droplets are not visible in the burned zone, the contour of a two-dimensional slice of the inflamed volume could be made visible. These experiments showed that the assumption of a spherically propagating flame is largely valid for stoichiometric mixtures. For lean mixtures however it was found that the shape of the inflamed volume is far from spherical. For many images islands of unburned mixture were
found in the burned zone and islands and peninsulas of burned products were seen in the
unburned mixture ahead of the flame. This indicates a very irregular, highly corrugated
three dimensional structure. The contour of the inflamed zone was investigated further
with help of fractal analysis. An attempt was made to quantify the flame surface from
the images and to compare this to the surface needed to justify the mass burning rate
found from the pressure signal analysis and laminar burning velocity. Here it was found
that the surface determined from the fractal analysis was a factor 2-5 too small. This was
attributed to the limited information on the three dimensional structure of the inflamed
volume and also to the fact that deformations larger than the integral length scale of the
flow were not accounted for by the fractal analysis.
Chapter 1

Introduction

1.1 Background

In Europe, a growing number of automotive engines are diesel engines. These engines have a higher efficiency compared to conventional (stoichiometric) SI engines, but - without after treatment - they have considerably larger emissions (i.e. of $NO_x$ and particle emission). For a long time, engines on natural gas have been proposed as a clean alternative to these diesel engines (especially in the medium and heavy-duty range). More recently, natural gas is being proposed as an alternative fuel producing relatively lower $CO_2$ emissions. Implicit to stoichiometric SI combustion is of course a lower (i.e. part-load) efficiency. That means that to achieve better than diesel $CO_2$ emission levels, the thermal efficiency of the natural gas engine should improve. A very effective way for this is to run these engines on a so-called lean mixture ($\lambda > 1$). Due to reduced throttling at part load, efficiency increases. At the same time, as the mixture is made fuel leaner, the peak combustion temperature is lowered and therefore the production of (thermal) $NO_x$ decreases. With such engines very low engine-out emissions levels have been demonstrated. These low emissions levels imply a much reduced need for after treatment. For realizing the very low emissions expected in the future however, it is necessary to shift towards even higher air-fuel ratios. There is however a maximum to the amount of mixture dilution an engine tolerates.

A consequence of leaning the mixture is that its laminar burning velocity is reduced. In an engine, there is only limited time available to burn all the mixture; therefore the lower burning velocity has to be compensated to achieve a sufficiently high overall burning rate. A common means to do this is to increase the mixture turbulence by changing the shape of the combustion chamber or inlet ports. For too high turbulence levels however it becomes very difficult to ignite the charge and also incomplete combustion occurs. As a consequence, engine efficiency decreases and the emission of unburned fuel ($HC$ emissions) increases sharply.

Next to this, in modern engine development the use of simulation and modeling techniques continuously increases. The software tools employed use mostly one-dimensional gas dynamics and thermodynamics calculations. The combustion process is then modeled using a prescribed burning rate or, in slightly more advanced codes, as a turbulent combustion process where the influence of variables like engine speed is accounted for. For gasoline and stoichiometric mixtures, this approach is successful to some extent. For lean mixtures however the combustion process is more complex and is influenced by engine
2 Introduction

and operating parameters in a more complicated fashion. For instance, simple scaling of the burning rate with engine speed is in some cases not justified anymore. These engine simulation codes would therefore benefit from a better description of the turbulent combustion process of lean mixtures.

1.2 Objectives

This thesis is the result of a study into the turbulent combustion process as it occurs in a lean burn natural gas engine. This study included both theoretical work as well as the construction of a test set-up. Various measurements were performed and the theory of turbulent combustion was tested against these measurements. This thesis therefore aims

1. To offer a description of the experimental work involved that was needed to investigate the combustion process in the engine used. This includes the construction and operation of an optically accessible test engine, a description of the measurement techniques used together with details of their application in the engine, and the description and application of various data processing techniques and methods.

2. To present relevant theoretical aspects of turbulent combustion and offer a comparison of this theory with measurements of the combustion process in the engine. This leads to understanding of the combustion process and the problems occurring in the combustion of fuel-lean mixtures in engines.

3. To provide insight in the aspects of the combustion process that need to be taken into account when developing predictive engine simulation codes for lean burn engines. These aspects include the change in fundamental properties of the mixture (i.e. laminar burning velocity) with increasing mixture dilution but also the interaction of turbulence and flame structures.

1.3 Outline of the thesis

This thesis starts with a literature survey into lean-burn engines. This survey gives an overview of various design aspects as well as the dilution limits that are observed in practical heavy-duty gas engines. Next, several aspects of the theory for turbulent combustion are presented. This chapter aims to present the concepts that are used in the end of this thesis when the interaction between combustion and turbulence is described. Chapter 4 introduces the experimental engine set-up that was constructed as part of this work. Key aspects of its construction are described and also various problems and difficulties that were encountered during its operation are explained. The engine was designed for optical access into the combustion chamber to allow the visualization of the combustion process. However, reference measurements without optical access were also performed. These measurements are the subject of chapter 5. In this chapter the results are presented for engine operation near the dilution limit. Measured emissions levels as well as heat release calculations are presented. These measurements form the basis for the selection of the engine operating conditions that were studied further using optical access. Chapter 6 describes the measurement of the velocities and flow structures in the combustion chamber of the engine. The measurement technique used, Particle Image Velocimetry is
introduced and results are presented. Next to information on the turbulence levels and flow structures, also the laminar burning velocity of the fuel/air mixture is required for an understanding of the combustion process. This laminar burning velocity was measured using the constant volume technique in the Eindhoven High Pressure Cell (EHPC). The techniques and methods used for this are the subject of chapter 7. The result of these measurements is a correlation for the laminar burning velocity of a methane/air mixture which captures the effects of equivalence ratio, pressure and temperature. Also, a similar correlation is presented for mixtures that are diluted with ’simulated’ residual products instead of excess air. Finally, in chapter 8, the results from this work are combined in an attempt to describe the turbulent combustion process in the engine. The combustion process is characterized by locating it in a combustion diagram. Also, visualizations of the combustion process that were conducted as an extension of the flow measurements are presented. These visualizations employ Mie scattering to visualize the shape of the turbulent flame envelope. To describe the relations between mass burning rate, laminar burning velocity and flame surface increase due to turbulence, fractal theory was employed in an attempt to describe the actual flame surface. The main findings are presented at the end of each chapter. Finally, in chapter 9 the findings and conclusions are summarized and recommendations are made for further research.
Chapter 2

Literature Survey

To define the scope of this PhD study, a literature survey was conducted. In this chapter the main results of this survey will be presented.

2.1 General

The engines that are used for transport purposes these days can be divided in two categories, spark-ignition (SI, otto-cycle) engines and compression-ignition engines (CI, diesel-cycle). For passenger car applications, SI engines are applied mostly. However, since small diesel engines nowadays have reached a power and torque level that exceeds that of SI engines, these engines are rapidly increasing their market share for passenger cars, especially in Europe.

For heavy duty engines (engines that deliver a high power and torque for a continued period) like those applied in trucks and buses, CI engines are the dominant engine type. The reason for this is that diesel engines offer a higher efficiency and lower fuel consumption compared to SI engines. Both engine types however are subject to legislative limitations; i.e. for transport purposes, governments limit the emission of polluting gases from traffic. The amounts of the various polluting gases emitted differ by engine type. For SI engines, the main polluting gases are nitrogen oxides ($NO$ and $NO_2$ are generally referred to as $NO_x$, the relative amounts however differ for SI and diesel engines), unburnt hydrocarbons (referred to as $HC$ emissions) and carbon monoxide ($CO$). For diesel engines, emissions originating from unburnt fuel like $HC$ and $CO$ are less of a problem, for these engines the $NO_x$ emissions together with the formation of particulate matter (soot) form a problem. Especially in urban areas, the emission of $NO_x$, $HC$’s and soot has a big impact on the environment. For instance, the $NO_x$ together with water forms acid rain, and $NO_x$ together with $HC$ can form a photochemical smog under the influence of sunlight. In some very densely populated areas a thick brown layer of this ’smog’ can be seen in the air above a city. These emissions originate mainly from automotive transportation.

2.2 Emissions sources

In this section the origin of the various emissions sources is reviewed. In this respect there are differences between spark-ignition and diesel engines. Since this thesis is on
combustion in SI engines using gaseous fuels, the focus will be on these engines. Where
differences between SI and diesel engines are relevant, these will be addressed.

2.2.1 \( NO_x \)

Nitrogen oxides are formed under the high temperatures occurring during the combus-
tion process. There are basically three mechanisms through which \( NO_x \) emissions can be
formed, thermal \( NO_x \), prompt \( NO_x \) and fuel \( NO_x \). Which mechanism prevails depends
on the conditions. For typical combustion in gas engines, thermal \( NO_x \) is responsible for
most of the \( NO_x \) emissions.

With thermal \( NO_x \), nitrogen from the air reacts directly with oxygen under high tem-
peratures in the post flame zone to form \( NO \) through the so called extended Zeldovich
mechanism [40]:

\[
\begin{align*}
O + N_2 & \rightleftharpoons NO + N \\
N + O_2 & \rightleftharpoons NO + O \\
N + OH & \rightleftharpoons NO + H
\end{align*}
\]

(2.1)  (2.2)  (2.3)

This mechanism for \( NO \) formation is very temperature dependent and has a lower tem-
perature limit of about 1700-1800 K. Important here is that the formation of \( NO \) through
this mechanism is not only controlled by the presence of high temperature and the avail-
ability of oxygen, but also time is very important. If there would be enough time, the \( NO \)
concentration would react to its equilibrium value. In an engine, during combustion the
\( NO \) concentration rapidly increases. During expansion the temperature decreases fast
so the reverse reactions do not have enough time to proceed. One could say that the
concentration of \( NO \) is “frozen” above the equilibrium concentration that corresponds to
the temperatures during expansion and exhaust.

The maximum temperatures occur with an air/fuel ratio around stoichiometric. Due to
the need for oxygen however, the maximum \( NO \) formation rate occurs at slightly leaner
mixtures.

To decrease the formation rate of \( NO \), one should reduce the necessary conditions. Since
the composition of air is fixed, reduction of the nitrogen concentration is not an option.
Also, the time available for combustion and expansion is determined by the engine speed.
The formation rate of \( NO \) can however be influenced by lowering the temperature level
during combustion. To sustain flame propagation at a certain speed however, a certain
minimum local temperature exists [25]. This implies that there also exists a certain min-
imum amount of \( NO \).

With prompt \( NO \), also called Fenimore \( NO \), the \( NO \) is formed in the flame front it-
self. This process is less temperature dependent, but is influenced by the air/fuel ratio.
At lower air/fuel ratios (rich mixture), more \( NO \) is formed. This mechanism is therefore
less relevant for lean combustion as it occurs in natural gas engines.

If nitrogen is present in the fuel itself in chemically bound form, most of this nitrogen is
converted to \( NO \). This process is referred to as fuel \( NO \). The contribution of this me-
chanism of course depends on the fuel used. For practical gaseous fuels, this mechanism is
not important.

2.2.2 Unburned hydrocarbons

Hydrocarbon emissions from SI internal combustion engines originate from a number of sources. The most important are crevice volumes, wall quenching, partial burns and misfires.

When the mixture in a SI engine is compressed, mixture is pushed into the crevice volumes (e.g. the space between piston and cylinder). Since flame propagation cannot take place in these crevices, this fuel is not burnt. When the pressure decreases during the expansion stroke, the unburnt mixture escapes from these crevices. If the temperature is still high enough, most of the fuel will oxidize but some will remain unburnt and escape through the exhaust. Crevice emissions do not play a role in diesel engines since in those engines the gases being compressed do not contain fuel.

During combustion a layer of mixture close to the relatively cold surfaces of the cylinder, cylinder head and piston cannot burn due to quenching effects. The thickness of this layer depends on the air/fuel ratio, temperature and the burning velocity of the mixture and increases with leaner mixtures.

If combustion is very slow, it can happen that the combustion process is not fully completed. The combustion process then essentially stops when the pressure and temperature decrease on opening of the exhaust valve or even already during the expansion stroke. This phenomenon is called bulk quenching or partial burn. The remaining fuel partly oxidizes in the exhaust but some escapes unburnt.

If the ignition fails to create a self-sustaining flame kernel, or if there is no spark at all, all the fuel in the mixture remains unburnt. This condition is called misfire.

2.2.3 Carbon monoxide

Due to imperfect mixing of air and fuel, pockets of rich mixture can occur. In this rich combustion, there is insufficient oxygen to oxidize the CO to CO$_2$ and CO can be found in the exhaust gases. Also the occurrence of colder areas during combustion can lead to the formation of CO. In modern SI engines, CO emission is no real problem anymore.

2.2.4 Particulate matter

The emission of particulate matter is not an issue with homogeneous charge SI engines (engines that use a fully premixed charge). In light-duty engines, there currently is a trend towards direct injection strategies. These engines do emit some fine particulate matter due to the locally inhomogeneous mixture. Direct injection engines however are not the main subject of this research. For (heavy-duty) diesel engines the emission of soot limits the air/fuel ratio that can be used.
2.2.5 Non-regulated emissions

Several gases that escape the engine in large or very small quantities are not regulated. An example of a non-regulated emission is the emission of carbon dioxide ($CO_2$). Despite always being present in the atmosphere, it is receiving a lot of attention since it is a so-called greenhouse gas. Reduction of $CO_2$ emission can be achieved in a number of ways, for instance by using a fuel with a higher $H/C$ ratio so less $CO_2$ is produced for a given output power. The fuel with the most extreme $H/C$ ratio would of course be hydrogen. Another method for reducing $CO_2$ emission is to increase engine efficiency, so less fuel is used for a given output power.

For obvious reasons, it is highly desired to develop better engines for transportation purposes that emit less polluting gases and have a higher efficiency. A further motivation for the development of alternatives is that the availability of crude oil is getting more difficult due to technical and political reasons and therefore the dependency on these oil-based fuels should be reduced by introducing more fuel diversity.

2.3 Natural gas engines

A possible alternative would be the use of SI engines using natural gas as a fuel. Since natural gas as an engine fuel has a much lower tendency to knock, these engines can achieve a higher thermal efficiency than SI engines running on gasoline by utilizing a higher compression ratio. By simply replacing the fuel however, a natural-gas engine still has the problem of relatively high $NO_x$ emission levels due to the high combustion temperatures associated with a stoichiometric mixture. This stoichiometric combustion enables the use of a three-way catalyst, so these $NO_x$ emissions can be reduced to a very low level. Stoichiometric combustion however is generally undesired for a heavy-duty engine as this operation mode is characterized by a lower thermal efficiency and high fuel consumption and a high thermal load on the engine components.

In practice, stoichiometric operation as in a light-duty gasoline engine is not used for heavy-duty engines. Instead these engines run on a diluted (fuel-lean) mixture. This has some important implications for the efficiency and emissions of these engines. First, it has to be stated that a fuel-lean mixture can be achieved in two different ways: “lean-burn” combustion and ”$\lambda=1+EGR$”. Of course, mixed forms also are possible, but for the moment a clear separation will be made.

2.3.1 Lean-burn engines

In lean-burn engines, the air/fuel mixture is diluted using excess air. The relative dilution is expressed using the air excess factor $\lambda$.

$$\lambda = \frac{(A/F)_{\text{actual}}}{(A/F)_{\text{stoichiometric}}} \quad (2.4)$$

Here, $(A/F)_{\text{actual}}$ indicates the actual air/fuel ratio, $(A/F)_{\text{stoichiometric}}$ indicates the stoichiometric air/fuel ratio which is of course fuel dependent. A stoichiometric engine
Natural gas engines operates at $\lambda = 1$, whereas a lean burn engine typically operates at $\lambda = 1.4 \ldots 1.7$. The higher limit is determined by the characteristics of the engine and the fuel. For increasing air excess factors, the combustion process starts to show variability in flame development and burn duration. These variations in the combustion process lead to variations in the amount of work per engine cycle which is experienced as a variation in engine torque. In the higher limit, the so called "lean limit", cyclic variations rapidly increase and so does the emission of $HC$ and $CO$. The energy of the spark may be insufficient to initiate a self-sustaining flame kernel as the minimum ignition energy rises with a leaner mixture. This phenomenon is known as misfire. Also the combustion becomes very slow so that there is insufficient time to complete the combustion. This phenomenon is generally called partial burn. In vehicle applications, the variability in engine torque (also called driveability) limits the use of the engine at higher air excess factors.

Typical advantages of lean-burn engines are:

- A diluted mixture has a lower tendency to knock, therefore a higher compression ratio can be used. This higher compression ratio leads to a higher thermal efficiency, since the thermal efficiency of the (ideal) otto-cycle is defined as:

$$\eta_{th} = 1 - \frac{1}{r_c^{-\gamma - 1}}$$  \hspace{1cm} (2.5)

in which $r_c$ is the compression ratio and $\gamma$ is the ratio of specific heats $c_p/c_v$.

- The average composition of the mixture has a higher ratio of specific heats ($c_p/c_v$), therefore more expansion work can be extracted from the cycle.

- For a given energy input, the mixture volume is larger. This means that, in throttled operation, the engine throttle valve can open further for a given output power giving less pumping work during the intake stroke. At lower engine load, this is the main advantage.

- Since the mixture is diluted, the heat from the combustion is adsorbed by a larger mass of gases. Therefore the temperatures during combustion are lower. Since the formation of $NO_x$ is largely temperature-controlled, much less $NO_x$ is formed at higher $\lambda$’s. For lean-burn combustion however, there is a peak in $NO_x$ production at slightly lean ($\lambda = 1.15$) air/fuel ratios. This is due to the simultaneous presence of high temperatures and of sufficient oxygen. For even leaner mixtures the $NO_x$ production steadily decreases.

- Since the gas temperatures are lower, less heat is transferred to the walls of the combustion chamber, cylinder and piston crown. Lower heat loss leads to a higher thermal efficiency.

- Lower gas temperatures put a lower thermal load on the exhaust system which is beneficial since most heavy-duty natural gas engines are modified versions of their diesel counterparts.

Of course, there are also disadvantages:
- Since the combustion is not stoichiometric, there is no balance between \( HC \)'s, \( CO \) and \( NO_x \) so a three-way catalyst cannot be used. Only an oxidation catalyst can be used to oxidize the \( HC \) and \( CO \). These catalysts have a much lower conversion efficiency (partly due to the low exhaust gas temperature of a lean-burn engine), and also the \( HC \) emissions from a natural gas engine mainly consist of methane which is a very stable molecule and therefore hard to catalytically oxidize.

- Leaning the mixture lowers its laminar burning velocity, this has to be compensated by increasing engine turbulence levels (by so called "fast burning combustion chambers"). Increasing engine turbulence generally leads to a higher pressure drop over the inlet valves and also the heat transfer to the combustion chamber walls is increased.

2.3.2 \( \lambda=1+EGR \) engines

In these engines, recycled exhaust gases are used to dilute the mixture. The fresh mixture itself has a stoichiometric air/fuel ratio. The overall air/fuel ratio remains stoichiometric and a three-way catalyst can be used. The mixture dilution is usually expressed as

\[
x_{EGR} = \frac{\text{mass of recycled exhaust gases}}{\text{total inlet mass}}
\]

Some of the benefits mentioned for lean mixtures in the previous section also apply here but there are also some differences:

Advantages:
- The higher allowable compression ratio and lower combustion temperature also apply here. The reduction of the combustion temperature is even more effective since exhaust gas has a higher specific heat than air. This is even more effective if the recycled exhaust gases are cooled prior to recirculation.
- A three-way catalyst can be used which results in very low emission levels
- \( NO_x \) production starts decreasing already with small dilution levels, there is no initial peak.

Typical disadvantages for these engines are:
- Unless exhaust gas cooling is applied, the temperature levels in the engine are higher than with lean burn.
- The addition of exhaust gases has a larger impact on the combustion, therefore the dilution level the combustion process can tolerate is usually much lower than with lean mixtures. Typically an engine tolerates some 60-70\% excess air, whereas only 25-30\% of exhaust gases can be used with acceptable engine stability.
- Due to the lower dilution, the advantage of an increased mixture volume is less.
- Exhaust gas recirculation and cooling leads to a more complex and therefore more expensive engine.
Development of lean burn and $\lambda=1+EGR$ engines

Lumsden [51] states that, just by diluting the mixture, very little net emissions advantages can be gained. Reduction of $NO_x$ by lowering combustion temperatures inevitably leads to an increase in $HC$ emissions. To have an overall advantage in the trade-off between $NO_x$ and $HC$ with respect to stoichiometric engines, after treatment of the exhaust gases is necessary. In that view, the $\lambda=1+EGR$ strategy has an advantage since a three-way catalyst can be used. The emissions trends are illustrated in figure 2.1 [42].

2.4 Development of lean burn and $\lambda=1+EGR$ engines

Economical and environmental considerations have led to a development towards cleaner and more efficient engines ever since the internal combustion engine was invented. However, in the last decades the main influencing factor in this development is the politics which imposes laws against 'dirty' vehicles and the increasing awareness that the world’s oil reserves are limited, which reflects in the oil price.

In the past decades, numerous engines have been converted to a fuel-lean technology (either lean-burn of $\lambda=1+EGR$) with varying success. This literature survey attempts to give a short impression of the challenges and limitations of such an engine conversion.

Most heavy-duty lean-burn engines are basically diesel engines that have been converted to SI operation. The main changes are a lower compression ratio, a different fuel supply (and storage) system, an ignition system, the introduction of a throttle valve to be able to control engine power for low loads, a change in valve timing and the placement of a
suitable catalyst. Because the number of lean-burn engines is rather low, the investment cost of engine modifications is kept to a minimum. The most important modifications are reviewed in the next sections. Also, some concepts and definitions are introduced which will be used in the course of this thesis.

2.4.1 Combustion chamber

A diesel engine uses a combustion chamber with a bowl-in-piston geometry. The compression ratio for a turbocharged diesel engine typically is about 1:16 - 1:18. For a lean burn SI engine the compression ratio is limited by engine knock, therefore the compression ratio is reduced to 1:10 - 1:14, depending on the intended fuel and air/fuel ratio. When using a diluted mixture, the burning velocity of the mixture itself (e.g. the laminar burning velocity of the mixture at a given air/fuel ratio) is much lower. As there is limited time available to burn the entire cylinder charge, the combustion locally has to be accelerated to increase the overall mass burning rate. This is usually accomplished by increasing the turbulence level of the mixture. The total mass burning rate in this situation is dominated by the turbulence. The burning velocity of the mixture itself does play a very modest role here. The turbulence level of the mixture in the cylinder during combustion is determined by the initial turbulence level of the mixture entering the cylinder during the intake stroke, and is increased further by the piston motion and the geometry of the combustion chamber. Engine designers have used this fact to influence (increase) the turbulence level by changing the shape of the diesel bowl-in-piston for a more complicated design. Examples of piston shapes are the 'Nebula' combustion chamber, developed by Ricardo [39], the 'Tri-Flow' combustion chamber, developed by AVL [13], or the 'Quartette' combustion chamber, developed by Einewall [20]. All these combustion chambers are designed to have high levels of small-scale turbulence, and have a flow pattern that has been shaped to give low flow velocities at the spark plug to favor ignition conditions. A number of combustion chambers is displayed in figure 2.2. The thermal efficiency is affected by the burn duration. The maximum possible efficiency is that of the ideal otto-cycle where all heat addition occurs instantaneously at top dead center. In reality, the combustion takes time, so less work can be extracted from the cycle. Combustion that occurs later in the cycle gives less work.

In literature, several examples of practical engines can be found. In figure 2.3 the efficiency of several engines has been plotted in chronological order. Where possible, also the combustion chamber layout is displayed. It is seen that the achieved thermal efficiency levels are pretty similar at about 40%. Unfortunately, only few references for $\lambda=1+EGR$ engines could be found.

2.4.2 Fuel system

For natural gas engines, the fuel system can be designed in different ways. Mostly, the air and fuel are mixed in the engine intake manifold using a venturi for gas metering and mixing. With this strategy, the mixture in the combustion chamber is homogeneous (the air/fuel ratio is the same everywhere). A second option is port fuel injection. In this case the fuel is injected in the inlet manifold runners for each cylinder. The available time for the fuel and air to mix is shorter and the mixture is less homogeneous. An advantage
Figure 2.2: Natural gas engine combustion chambers

Figure 2.3: Efficiencies lean-burn and λ 1+EGR engines
here is the possibility of air/fuel ratio tuning per cylinder. A third option is to inject the fuel in the cylinder itself ("direct injection"). With this strategy, even more flexibility is possible. Also, so called "stratified combustion" is possible. With stratified combustion, the overall air/fuel ratio is lean but by timing the moment of fuel injection with respect to the valve and ignition events locally richer conditions are possible to improve ignition conditions. Due to the higher cost of port-fueling and direct injection systems these are until now rarely applied for heavy-duty natural gas engines. The engines considered in the present study therefore use a homogeneous mixture.

2.4.3 Ignition system

When the mixture is diluted, the energy needed for a successful ignition increases (figure 2.4). This energy also depends on the heat losses from the ignition source, which are partly due to heat conduction to spark electrodes and partly due to convective losses to the mixture flowing along the ignition source. For most lean burn engines, a conventional spark ignition system is used. These ignition systems can ignite mixtures up to approximately $\lambda=1.7\ldots1.8$. When an even leaner mixture is used, sometimes pre-chamber ignition systems are applied. With these systems, the actual spark ignition takes place in a richer mixture, after which this rich mixture ignites the bulk of the very lean mixture. Also, on some engines the lean air/fuel mixture is ignited by a small injection of diesel fuel (pilot injection system). The latter two ignition systems are much more complex, and therefore more expensive. Also, for very low emission limits, the emissions from a pre-chamber or pilot injection become a significant part of the total engine emissions.

2.4.4 Miscellaneous effects

There are of course many more factors in the design and operation of lean-burn and $\lambda=1+EGR$ engines that play a role in the efficiency and emission levels that can be achieved. The case-studies found in literature unfortunately are often poorly documented. On a number of influencing factors, too little data can be found to indicate trends. The
Development of lean burn and $\lambda=1+$EGR engines

The following section presents some general trends, based on the few references found.

### Fuel composition effects

Most lean-burn SI engines use natural gas as a fuel, however the composition of natural gas is not the same everywhere. The composition varies between almost 100% $CH_4$ to fuels which contain a substantial fraction of higher hydrocarbons and/or inert components. Some typical compositions are indicated in table 2.1 [80]. One of the most important properties of a gaseous fuel is its knocking sensitivity. The knocking sensitivity of a natural gas is determined by the composition and expressed by the Methane Number (MN). The methane number is 100 for methane and zero for hydrogen. For natural gas compositions, methane numbers range from about 60 to 100. Heavier hydrocarbons reduce the methane number (making the air/fuel mixture more sensitive to knock), inert species like $N_2$ increase the methane number. When an engine is designed for a fuel with a high methane number, the compression ratio can be chosen higher which leads to a higher efficiency.

### Compression ratio

As already mentioned, a higher compression ratio gives a higher theoretical thermal efficiency. Also, with a higher compression ratio the end-compression temperature is higher and ignition of the mixture will be easier. At very high end-compression pressures, the voltage demands on the ignition system increase which often puts a practical limit on the boosting pressure.

### Engine speed

No significant effects of engine speed were found. The engines documented in literature had similar size, and also the turbulence in an engine scales with piston speed. This piston speed is limited by the materials of the piston and liner, and is very similar among different engines with different engine speeds.

### Engine load

When increasing engine load, the pressures and temperatures increase. This increases the amount of $NO_x$ that is formed. On the other side, with higher exhaust gas temperatures the conversion efficiency of an after treatment system also increases. The amount of residual gas decreases due to the larger flow, which increases the mixture dilution tolerance.

<table>
<thead>
<tr>
<th>Origin</th>
<th>$CH_4$</th>
<th>$C_2H_6$</th>
<th>$C_3H_8$</th>
<th>$C_4H_{10}$</th>
<th>$C_5H_{12}$</th>
<th>$N_2$</th>
<th>$CO_2$</th>
</tr>
</thead>
<tbody>
<tr>
<td>Netherlands</td>
<td>83.5</td>
<td>3.6</td>
<td>0.7</td>
<td>0.2</td>
<td>0.1</td>
<td>10.8</td>
<td>1.1</td>
</tr>
<tr>
<td>Norway</td>
<td>88.7</td>
<td>4.9</td>
<td>1.1</td>
<td>0.4</td>
<td>0.1</td>
<td>3.4</td>
<td>1.4</td>
</tr>
<tr>
<td>Russia</td>
<td>96.2</td>
<td>1.2</td>
<td>0.3</td>
<td>0.2</td>
<td>0.0</td>
<td>1.8</td>
<td>0.3</td>
</tr>
<tr>
<td>Algeria</td>
<td>88.6</td>
<td>8.2</td>
<td>2.0</td>
<td>0.7</td>
<td>0.0</td>
<td>0.6</td>
<td>0.0</td>
</tr>
<tr>
<td>France</td>
<td>97.3</td>
<td>2.1</td>
<td>0.2</td>
<td>0.2</td>
<td>0.0</td>
<td>0.3</td>
<td>0.0</td>
</tr>
<tr>
<td>Abu Dhabi</td>
<td>82.1</td>
<td>15.9</td>
<td>1.9</td>
<td>0.2</td>
<td>0.0</td>
<td>0.1</td>
<td>0.0</td>
</tr>
</tbody>
</table>

Table 2.1: Volumetric composition of natural gases
Engine size
The investigated engines had similar sizes, no clear trends could be found here. However, with larger engines the power loss due to friction is relatively smaller. Also the ratio of combustion chamber surface area to volume is smaller which gives less heat loss and a therefore a higher efficiency.

2.4.5 Conclusions

Engine design
In figure 2.3, two different efficiencies have been plotted, the first is the overall maximum efficiency (closed symbols). The maximum efficiency mostly occurs at a medium engine load. The other is the nominal efficiency (open symbols), which is the efficiency at the nominal maximum operating point the engine was designed for. As can be seen, the maximum thermal efficiency is limited to approximately 40%. The single engine that reached 42% efficiency was a specific co-generation $\lambda=1+EGR$ engine, designed for only one engine speed and load. Applications for vehicle use have to account for load and speed variations and transient behavior. Compromises have to be made, so the overall efficiency is less optimal. The nominal efficiency is typically a few percent lower than the maximum efficiency. The reason for this is that the air/fuel ratio is typically somewhat lower at maximum load and the volumetric efficiency is also a bit lower.

If one looks at the various combustion chambers that have been plotted besides the efficiency points, it turns out that special combustion chamber shapes have been popular for some time but that a similar efficiency can be achieved with a much simpler combustion chamber shape (which is more similar in shape to the diesel piston). The benefits do not outweigh the extra cost of a complex piston design.

Dilution limitations
Apart from the construction of lean burn engines, it turns out there are a number of fundamental limitations to the combustion process in this engine type. The mixture dilution in practical engines does not exceed $\lambda=1.6...1.8$. Beyond this point the mixture is very difficult to ignite, and also the combustion gets very slow. Ultimately, the occurrence of unburnt fuel ($HC$ emission) and combustion variations limit the operation. The dilution limit for $\lambda=1+EGR$ engines is around 25-30% dilution (mass fraction). At this point an increase of $HC$ emissions and cyclic variations limits engine operation. Exhaust gases have -especially when cooled before recirculation- a greater impact on the reduction of $NO_x$ emission than excess air because there is no initial peak in $NO_x$ production with low dilution levels. As there is no excess oxygen present with this dilution type, incomplete combustion occurs with lower dilution levels than with lean burn.

Efficiency
In general the maximum possible efficiency of lean burn engines is somewhat higher than that of $\lambda=1+EGR$ engines. However, for lean burn engines the mixture dilution has to be maximized to meet the $NO_x$ emission regulation with minimal after treatment. At these high $\lambda$ values, the efficiency already decreases again due to a slow combustion process and also $HC$ emissions form a problem.
Which engine strategy is best is not a simple choice. There are fundamental differences in the combustion process of mixtures diluted with air and with exhaust gases. The explanation of these differences is one of the objectives of this research, together with the influence of fuel composition on the combustion process in lean burn and $\lambda=1+EGR$ engines.

2.5 Discussion

In this chapter, a literature survey was presented which attempts to indicate the scope of this research. The effects of mixture dilution in SI engines have been described, and also the differences regarding engine efficiency and emissions. It was shown that there is a clear limit to the amount of dilution as engine operation becomes difficult. Also, simply diluting or leaning the mixture to the limit is not the way to go for an optimal engine design. More knowledge is needed to understand the details of the combustion process in order to predict the effects of a change in for instance engine geometry, engine speed or fuel composition on the combustion.

In this respect, simulation tools find more and more application. These tools however do not incorporate the effects of mixture dilution fully. For stoichiometric combustion simple burning rate relations ("Wiebe functions") can be used. For very diluted mixtures however these simple approaches do not capture the specific effects of high mixture dilution.
Chapter 3

Combustion theory

In this chapter, the theory of premixed combustion which is needed to understand the combustion phenomena in lean burn and $\lambda=1$+EGR engines is presented. As this combustion process is quite complex, this chapter will start with the principles of chemistry and heat release. Combustion in SI engines is a turbulent combustion process. It is generally assumed that this combustion process, on a smaller scale, can be seen as an array of so called flamelets in which locally laminar combustion occurs. Therefore, laminar combustion and the laminar burning velocity will be reviewed. In a turbulent combustion process, there is a strong interaction between the combustion chemistry and the motion of the gases. The characteristics of the flow (velocities and spatial scales) together with the local properties of the flame can be used to characterize the combustion in a diagram. Using this knowledge the engine combustion process will be reviewed in more detail. The problems occurring with combustion of diluted mixtures will be explained using the combustion theory presented.

3.1 Combustion chemistry

A combustion process is a process in which a fuel reacts with oxygen. In this reaction, the reactants are converted to products, thereby releasing a certain amount of energy in the form of heat.

\[
\text{Fuel} + \text{Oxygen} \rightarrow \text{Products} + \text{Heat}
\]  

(3.1)

Fuels used in internal combustion engines typically consist of a mixture of hydrocarbons ($C_xH_y$). When assuming complete combustion, the reaction products are carbon dioxide and water. The general chemical reaction equation between a hydrocarbon and oxygen can be written (in molar terms) as

\[
C_xH_y + (x + \frac{y}{4})O_2 \rightarrow xCO_2 + \frac{y}{2}H_2O
\]  

(3.2)

A similar formulation in terms of mass can be found by multiplying with the molar mass $M$ of each species:

\[
(xM_c + yM_H)C_xH_y + (x + \frac{y}{4})M_{O_2}O_2 \rightarrow xM_{CO_2}CO_2 + \frac{y}{2}M_{H_2O}H_2O
\]  

(3.3)
Since the fuel is not mixed with only oxygen but with air which contains roughly 21% (by volume) oxygen, the reaction becomes

\[ (xM_c + yM_H)C_xH_y + (x + \frac{y}{4})(M_O_2O_2 + 3.76M_N_2N_2) \rightarrow \]

\[ xM_{CO_2}CO_2 + \frac{y}{2}M_{H_2O}H_2O + 3.76(x + \frac{y}{4})M_N_2N_2 \] \hspace{1cm}(3.4)

Reaction 3.4 is valid for stoichiometric (\( \phi = 1 \)) combustion only. For leaner air/fuel mixtures (\( \lambda > 1 \)), there is excess air present so reaction 3.4 can be extended to

\[ (xM_c + yM_H)C_xH_y + \lambda(x + \frac{y}{4})(M_O_2O_2 + 3.76M_N_2N_2) \rightarrow \]

\[ xM_{CO_2}CO_2 + \frac{y}{2}M_{H_2O}H_2O + (\lambda - 1)(x + \frac{y}{4})M_O_2O_2 + \]

\[ + \lambda 3.76(x + \frac{y}{4})M_N_2N_2 \] \hspace{1cm}(3.5)

For \( \lambda=1+EGR \) engines, there is no excess air, instead products are mixed with the reactants. With the fraction of recycled exhaust gases \( x_{EGR} \) defined as in equation 2.6

\[ x_{EGR} = \frac{\text{mass of recycled exhaust gases}}{\text{total inlet mass}} \]

the reaction can be written as

\[ (1 - x_{EGR})((xM_c + yM_H)C_xH_y + (x + \frac{y}{4})(M_O_2O_2 + 3.76M_N_2N_2)) + \]

\[ + x_{EGR}(xM_{CO_2}CO_2 + \frac{y}{2}M_{H_2O}H_2O + 3.76(x + \frac{y}{4})M_N_2N_2) \rightarrow \]

\[ xM_{CO_2}CO_2 + \frac{y}{2}M_{H_2O}H_2O + 3.76(x + \frac{y}{4})M_N_2N_2 \] \hspace{1cm}(3.6)

The heat released by the reaction is used to raise the temperature of the products. As a result of this, the pressure rises which can be used to extract work. On a more detailed level, the chemical reactions are much more complex than suggested in equation 3.2. For instance, the reaction for the combustion of methane in a single step would be

\[ CH_4 + 2O_2 \rightarrow CO_2 + H_2O \] \hspace{1cm}(3.7)

In reality, dozens of intermediate reactions play a role. For the combustion of methane, some of these reactions are

\[ CH_4 \rightarrow CH_3 + H \quad \text{(chain initiation)} \]
\[ CH_3 + O_2 \rightarrow CH_2O + OH \quad \text{(chain propagation)} \]
\[ OH + CH_4 \rightarrow H_2O + CH_3 \quad \text{(chain propagation)} \]
\[ O_2 + H \rightarrow O + OH \quad \text{(chain branching)} \]
\[ OH + H \rightarrow H_2O \quad \text{(chain breaking)} \]

These intermediate species can all be found in small quantities in the flame front.

## 3.2 Laminar premixed combustion

### 3.2.1 Laminar burning velocity

The term flame is most often used in association with something visible. In this context, a flame is the reaction front in which reactants are converted into products (figure...
3.1. This flame can be subdivided in separate zones, the pre-heat zone and the reaction layer (figure 3.2). In the pre-heat zone, the unburnt mixture is heated up to reaction temperature. The heat needed for this comes from the chemical reactions in the reaction layer. A more detailed view of the internal flame structure for a $CH_4$ flame, including the concentration of some species can be seen in figure 3.3.

The burning velocity $S_L$ is defined as the velocity at which the reaction front propagates into a quiescent mixture. This is illustrated in figure 3.4. The burning velocity is a property of the fuel, mixture composition (equivalence ratio and presence of inert diluents), pressure and temperature. In figure 3.5, the burning velocity of a number of fuels as function of equivalence ratio is shown [40]. The burning velocity is roughly determined by two factors: the reaction time scale $t_r$ and the diffusivity of heat and mass $D$. The reaction time scales are determined by the activation energy of the reaction $E_a$ and can be approximated by

$$t_r \approx \frac{1}{A \rho_u} e^{\frac{E_a}{RT}}$$

in which $A$ is a frequency factor for the overall reaction and $\rho_u$ is the density of the unburnt mixture. Reactions only start when the temperature of the unburnt mixture
is close to the adiabatic burning temperature $T_b$. This is also seen in the reaction time scale. Due to the large activation energy $E_a$, the reaction time scale is very large unless the temperature $T$ approaches $T_b$.

The second important factor, the diffusivity $D$, determines the transport of heat from the reaction zone to the unburnt mixture. The larger the diffusion rate, the faster the unburnt mixture reaches a temperature at which reactions start. The flame temperature $T_b$ is influenced by the temperature of the unburnt mixture $T_u$. At a higher flame temperature the mixture ahead of the flame reaches the reaction temperature sooner, so the burning velocity is higher with a higher $T_u$. With the parameters $t_r$ and $D$, the following relation [16] can be written for the adiabatic burning velocity

$$S^0_L = \sqrt{\frac{D}{t_r}} = \sqrt{\frac{\rho_u A D e^{-E_a/RT_b}}{}}$$ (3.9)

The superscript '0' in equation 3.9 indicates unstretched conditions. This will be explained in a separate section.

The burning velocity also depends on the pressure. From equation 3.9, it could be expected that the burning velocity is not pressure dependent since $\rho_u$ scales with pressure while the diffusivity $D$ scales with the inverse. However when pressure rises, chain-breaking reactions, which slow down the combustion process, become more important. This leads (for methane/air combustion) to a pressure dependence of $S^0_L \sim p^{-0.5}$. The mass burning rate $m_b^0 = S^0_L \rho_u$ however does increase with pressure $m_b^0 \sim p^{0.5}$.

**Correlations for the laminar burning velocity**

The reference data for the laminar burning velocity that can be found in literature have mostly been determined at ambient conditions (293K, 1 atm). In an SI engine however, the pressure and temperature levels are far from ambient. The pressure can reach values
up to 100 bars, the temperature of the unburnt mixture can reach up to \(\pm 1000\text{K}\) due to compression of the unburnt mixture. In this research, correlations for the burning velocity as function of pressure, temperature etc. will be used. These generally have the form

\[
S_L = f(S_{L,0}, p, T_u, \phi, D)
\]  

(3.10)

in which \(S_{L,0}\) is a reference burning velocity, \(\phi\) the mixture equivalence ratio and \(D\) is the fraction of inert diluents in the unburnt mixture. In a later chapter these correlations will be treated in more detail. In chapter 2, the air excess factor \(\lambda\) was introduced. In combustion literature, it is more common to use the mixture equivalence ratio \(\phi\). The mixture equivalence ratio is defined as

\[
\phi = \frac{(F/A)_{\text{actual}}}{(F/A)_{\text{stoichiometric}}} = \frac{1}{\lambda}.
\]  

(3.11)

In this thesis, both terms are used. For engine related cases \(\lambda\) is mostly used, for more fundamental combustion related cases \(\phi\) is often preferred.

### 3.2.2 Flame thickness

The thickness of the reaction front is of the order \(O(0.1\text{mm})\), and depends on the properties of the mixture (burning velocity) and pressure. For leaner mixtures the flame thickness is larger, for higher pressures the flame thickness reduces. In literature, several definitions for the flame thickness \(\delta_f\) can be found [28]. These definitions are all based on some kind of diffusion.

\[
\delta_f = \frac{k_u}{\rho_u c_p S_L} \quad (\text{thermal diffusivity})
\]  

(3.12)

\[
\delta_f = \frac{D}{S_L} \quad (\text{mass diffusivity})
\]  

(3.13)

\[
\delta_f = \frac{\nu}{S_L} \quad (\text{viscosity})
\]  

(3.14)

### 3.2.3 Flame temperature

The temperature of an adiabatic flame \(T_b\) can be estimated using conservation of enthalpy [64]. During combustion, the pressure is assumed constant. The specific enthalpy of the products then equals that of the reactants.

\[
h_u = \sum_{i=1}^{n} Y_{i,u} h_{i,u} = \sum_{i=1}^{n} Y_{i,b} h_{i,b} = h_b
\]  

(3.15)

\(Y_i\) is the mass fraction of species \(i\). From

\[
\sum_{i=1}^{n} Y_{i,u} h_{i,\text{ref}} + \int_{T_{\text{ref}}}^{T_u} c_{p,u} dT = \sum_{i=1}^{n} Y_{i,b} h_{i,\text{ref}} + \int_{T_{\text{ref}}}^{T_b} c_{p,b} dT
\]  

(3.16)

in which

\[
c_{p,u} = \sum_{i=1}^{n} Y_{i,u} c_{p,i}(T), \quad c_{p,b} = \sum_{i=1}^{n} Y_{i,b} c_{p,i}(T)
\]  

(3.17)

the adiabatic flame temperature \(T_b\) can be determined.
3.2.4 Flame stretch

The burning velocity $S_L^0$ is the burning velocity under unstretched conditions. In this ideal situation the flame front is a constant surface. This gives a balance between the diffusion of species, radicals and heat which leads to a certain propagation velocity $S_L^0$ of the flame front. In more realistic situations, the flame front is affected by disturbances like the flow of the unburnt mixture. In general, the flame surface area and shape are not constant then. This phenomenon is generally known as flame stretch. The flame stretch rate $K$ is defined as the relative change in flame surface area

$$K = \frac{1}{A} \frac{dA}{dt}$$

(3.18)

Stretch can come from two mechanisms, strain and curvature. With strain, the change in surface area comes from velocity gradients in the area of the flame front. These velocity gradients occur when the flow is not perpendicular to the flame front together with velocity gradients in the flow field, or if the flame front is curved. With curvature, the change in flame area comes from the propagation of a curved flame front. For example an expanding spherical flame experiences an increasing flame surface ($K > 0$). The curvature however decreases as the flame expands and so does the flame stretch. Flame stretch is best explained using the Landau-Darrieus instability. Without the effects of flame stretch all premixed flames would be inherently unstable. Consider a flame front with an initial disturbance (figure 3.6 [16]). In the upper bulge, due to thermal expansion in the flame front, the streamlines on the burnt side are bent towards the local normal on the flame front. This means that on the unburnt side there is a diverging flow ($u_a$ is the local flow velocity of the unburnt mixture). If the local burning velocity $S_L$ would be constant, this would mean that $u_a < -S_L$ and the disturbance would grow. The same holds for the lower bulge, where $u_a > -S_L$. Flame stretch generally has a decreasing effect on $S_L$, so these disturbances will be damped.

The effect flame stretch has on the burning velocity is determined by the Lewis number

\[ u_a = -s_L \]
Le. The Lewis number is defined as the ratio between thermal- and mass diffusivity

\[ Le = \frac{\text{thermal diffusivity}}{\text{mass diffusivity}} = \frac{\alpha}{D} = \frac{k_u}{\rho_u c_{p,u} D} \] (3.19)

If \( Le > 1 \), positive stretch \( K > 0 \) has a decreasing effect on the flame temperature and the burning velocity. If \( Le < 1 \), positive flame stretch leads to a higher flame temperature and burning velocity. This may lead to unstable flames. Whether this happens depends on the value of the Lewis number and on the heat loss the flame experiences. There exists a certain critical Lewis number \( Le_c \). This critical Lewis number is smaller than 1 for zero heat loss, but increases towards \( Le_c = 1 \) when the heat loss increases. For flames with \( Le > Le_c \), hydrodynamic disturbances will be damped, flames for which \( Le < Le_c \) are inherently unstable. An example are hydrogen flames which have a low Lewis number and develop cellular structures.

When looking at things in more detail, it is possible to relate the reduction in flame temperature and burning velocity due to stretch to certain properties of the system. First, the flame stretch \( K \) is expressed as a Karlovitz number \( Ka_a \) (dimensionless flame stretch) using the flame thickness \( \delta_f = k_u/\rho_u c_{p,u} S_L \) (equation 3.12) and burning velocity \( S_L \)

\[ Ka_a = \frac{K \delta_f}{S_L} = \frac{K k_u}{\rho_u c_{p,u} S_L^2} = \frac{\rho_u K k_u}{c_{p,u}(m_b)^2} \] (3.20)

Next, a quantity called the Markstein number \( M \) is introduced. This Markstein number indicates the sensitivity to stretch and is proportional to \( Le - Le_c \). Using these quantities, the stretched flame temperature \( T_b \) can be expressed as

\[ T_b = T_b^0 [1 - \epsilon Ka_a(Le - 1)] \] (3.21)

The parameter \( \epsilon \) depends on the activation energy \( E_a \). The change in the flame temperature has an effect on the burning velocity which can expressed as

\[ S_L = S_L^0 [1 - M Ka_a] \] (3.22)

The above analysis is valid for weak stretch, at the moment it is unknown if it is also applicable to the stretch rates occurring in turbulent engine combustion.

### 3.3 Turbulent combustion

Combustion in an internal combustion engine takes place in a turbulent flow field, therefore turbulent combustion phenomena will be treated in this section. The principles explained in the previous section also apply for turbulent combustion. The general assumption (for engine combustion) is that a turbulent flame essentially is an array of (pieces of) laminar flamelets. The turbulence merely deforms the laminar flame to a certain extent. First, some basic concepts will be explained.

#### 3.3.1 Turbulent flow field

A turbulent flow field is characterized by velocity fluctuations in space and time. The flow field contains various spatial and temporal scales. The velocity \( u \) at a certain point
in space $x$ can be written as \[47\]
\[u(x, t) = \bar{u}(x) + u'(x, t)\] (3.23)
in which $\bar{u}(x)$ is the mean velocity and $u'(x, t)$ is the turbulent velocity fluctuation about the mean. The mean velocity is defined as
\[
\bar{u}(x) = \lim_{\Delta t \to \infty} \frac{1}{\Delta t} \int_{t_0}^{t_0+\Delta t} u(x, t) dt
\] (3.24)
The r.m.s. velocity fluctuation $u'_{rms}(x)$ can be written as
\[
u_{rms}(x) = \lim_{\Delta t \to \infty} \left( \frac{1}{\Delta t} \int_{t_0}^{t_0+\Delta t} (u(x, t) - \bar{u}(x))^2 dt \right)^{1/2}
\] (3.25)
The spatial scales in a turbulent flow field range from the integral length scale $L_i$, determined by the dimensions of the volume, through the Taylor scale $L_T$ to the smallest Kolmogorov scale $\eta$ which is determined by the turbulence level and the properties of the fluid. Turbulence is characterized by the Reynolds number
\[
Re = \frac{u L_i}{\nu}
\] (3.26)
$\nu$ is the kinematic viscosity of the fluid. Sometimes a turbulent Reynolds number
\[
Re_t = \frac{u' L_i}{\nu}
\] (3.27)
is used. The integral length scale $L_i$ is a typical dimension of the volume, the Kolmogorov scale
\[
\eta = \left( \frac{\nu^3}{\varepsilon} \right)^{1/4}
\] (3.28)
is the scale at which the smallest eddies in the fluid dissipate their energy through viscous dissipation. $\varepsilon$ is the energy dissipation rate ($\varepsilon \approx u'^3 / L_i$). Time- and velocity scales at the smallest level are $\tau_{\eta} = (\nu/\varepsilon)^{1/2}$ and $u_{\eta} = (\nu \varepsilon)^{1/4}$. The largest and smallest scale are related by
\[
\frac{\eta}{L_i} \approx \left( \frac{u' L_i}{\nu} \right)^{-3/4} = Re^{-3/4}\]
(3.29)
The Taylor scale $L_T$ is the distance over which strain occurs in a turbulent flow
\[
L_T = \frac{L_i \nu}{u'} = \frac{L_i^2}{Re_t}
\] (3.30)
To give some indication of these spatial scales for a typical internal combustion engine flow \[40\]: the integral length scale is of the order of the valve lift, but decreases as the piston approaches TDC and the mixture is compressed. The resulting integral length scale is $L_i=10...20$ mm, the Taylor scale $L_T$ is $O(1)$ mm and the Kolmogorov scale $\eta$ is as small as $0.01$ mm. The latter two scales depend on the turbulence level and therefore on the engine speed.
3.3.2 Chemistry time scales

Using the flame thickness $\delta_f$ and burning velocity $S_L$ as introduced in 3.2.1 and 3.2.2, some characteristic time scales for the combustion can be defined. The flame thickness and burning velocity can be combined in a chemical time scale, or flame residence time $\tau_c = \delta_f / S_L$. Similar, using the turbulent velocity fluctuation $u'$ and integral length scale $L_i$, a turbulent mixing time $\tau_m = L_i / u'$ can be defined. Now a non-dimensional number called the Damkohler number can be derived

$$Da = \frac{\tau_m}{\tau_c} = \frac{L_i S_L}{u' \delta_f}$$  \hspace{1cm} (3.31)

This Damkohler number indicates the speed of the chemical reactions relative to the convective mixing time. A large $Da$ indicates fast chemistry. In this case, thin flame fronts are present. A small $Da$ indicates that the mixing is faster than the chemistry. In terms of flames this means that there are no flame fronts present but instead the reactions take place simultaneously in multiple reaction zones. The Damkohler number, together with the turbulent Reynolds number can be used to display the various modes of combustion in a diagram (figure 3.7 [2]). In this diagram, lines indicating constant $u'/S_L$, constant $\eta/\delta_f$, and constant $L_i/\delta_f$ can be drawn. $u'/S_L$ is a measure of the relative strength of the turbulence. The ratio $\eta/\delta_f$ is a measure of the local distortion the flame experiences due to the turbulence. For conditions for which thin flames exist, $L_i/\delta_f \gg 1$. According to Abraham [2], the hatched region in figure 3.7 indicates the conditions where normal SI engine combustion takes place.

Nowadays, another kind of combustion diagram called the Borghi diagram finds more use. Peters [65] introduced a modified version of this diagram. In this diagram (figure 3.8),

Figure 3.7: $Da - Re_T$ combustion diagram [2]
the parameters which make up the Damkohler number $Da$ are re-arranged.

$$Da = \frac{\tau_m}{\tau_c} = \frac{L_i S_L}{u' \delta_f} = \left( \frac{S_L}{u'} \right) \cdot \left( \frac{L_i}{\delta_f} \right)$$  \hspace{1cm} (3.32)

A further useful non-dimensional number is the turbulent Karlovitz number $Ka$ (not to be confused with the dimensionless flame stretch $Ka_a$). This number can be regarded as the inverse of the Damkohler number, based on the Kolmogorov length scale [66]

$$Ka = \frac{1}{Da(\eta)} = \frac{\tau_F}{\tau_\eta} = \frac{\delta_f^2}{\eta^2} = \frac{u_\eta^2}{S_L^2}$$  \hspace{1cm} (3.33)

The Karlovitz number $Ka$ measures the flame scales in terms of Kolmogorov scales and indicates whether the smallest eddies are capable of influencing the internal flame structure.

In the diagram, next to the Karlovitz number as defined in equation 3.33, a second Karlovitz number $Ka_\delta$ based on the inner layer thickness $\delta_\eta$

$$Ka_\delta = \frac{\tau_\delta}{\tau_\eta} = \frac{\delta_\delta^2}{\eta^2} = \delta^2 Ka$$  \hspace{1cm} (3.34)

is defined [66]. The ratio of velocity scales $S_L/u'$ is then plotted against the length scales ratio $L_i/\delta_f$. In the resulting diagram, a number of regimes can be differentiated.

In the region labeled 'laminar', velocities are so low that the Reynolds number $Re < 1$ so flames have the shape of continuous, almost flat sheets. In the wrinkled flamelets regime, the flame sheets are deformed by the turbulent flow field, however, these sheets are still continuous. In the corrugated flamelets regime the flames are curved to such
extent that these can influence each other. Further up in the diagram, the thin reaction zones regime can be found. Here the smallest eddies can penetrate into the pre-heat zone and influence the flame structure. In the top regime, labeled ‘broken reaction zones’, flames cannot be found. The turbulence is so strong that the normal flame structure is broken apart and reactions are occurring in multiple places simultaneously. As an illustration, the interaction of the turbulent eddies with the flames is shown in figure 3.9. In the left figure, the smallest eddies are still so large that these can only deform the flame sheet, but cannot enter the flame structure (wrinkled/corrugated flamelet regime). In the right figure, the eddies at Kolmogorov scale are small enough to penetrate into the flame structure and influence the diffusion processes in the flame (thin reaction zones regime). At the separation line between the corrugated flamelet and thin reaction zones regimes, the Kolmogorov scale $\eta$ is of the same order as the total flame thickness $\delta_f$. This means that the turbulent eddies will influence the pre-heat zone of the flame in such a way that the flame behavior is affected. The pre-heating of the unburnt mixture is accelerated by ‘turbulent’ mixing in the pre-heat zone. The pre-heat zone will thicken, but the reaction layer remains unaffected. At the line between the thin reaction zones and broken reaction zones, the turbulent eddies are comparable to the inner layer thickness $\delta_l$. These eddies can enter the inner flame structure and break the flame apart. At that moment, the turbulent mixing causes reactions to happen in multiple places and a laminar flame cannot be found anymore.
3.3.3 Turbulent burning velocity

As explained in the previous section, in moderate to strong turbulence intensities (from the wrinkled flamelet up to the thin reaction zones regime) the effect of turbulence is that it deforms the flames without affecting them very much. This is the basis for most expressions for the turbulent burning velocity $S_T$. A turbulent flame is assumed to locally propagate with the laminar burning velocity $S_L$ (figure 3.10). The wrinkling of the flame leads to an increase in the flame surface area so that the total mass burning rate can be estimated using the flame surface increase.

$$ S_T \cdot A_T \cdot \rho_u = \dot{m}_b = S_L \cdot A_L \cdot \rho_u $$

A turbulent flame appears much thicker than a laminar flame due to the wrinkling of the flame surface. The turbulent flame surface $A_T$ is the surface that surrounds the turbulent flame brush. The increase in flame surface area is often related to the turbulence intensity. Most correlations for the turbulent burning velocity $S_T$ are empirical relations and have the following form

$$ \frac{S_T}{S_L} = 1 + C \left( \frac{u'}{S_L} \right)^n $$

These correlations are then fitted to measurements to determine the $C$ and $n$. More complicated expressions may also account for the effects of flame stretch.

3.4 Combustion in SI engines

As already mentioned earlier, combustion in SI engines is a turbulent combustion process. The combustion process is initiated by the energy from an electrical spark. This energy creates an initial volume in which the temperature is high enough to initiate combustion. Also, due to the very high temperature at the spark, an initial radical pool is formed. From this ignition source, a small spherical flame kernel is formed. Initially, the flame kernel is small compared to the flow scales and its propagation is essentially laminar. When the flame kernel grows, it is affected by the turbulent eddies. Initially, only the smallest eddies can make the flame surface wrinkle. The larger eddies can only kick the flame kernel around. Due to the cascade in eddy sizes, the flame is affected by eddies of increasing size as the flame grows. When the flame kernel reaches a size comparable to the integral length scale, the combustion can be considered fully turbulent.

During the following combustion process, the air/fuel mixture is consumed. During the combustion process, the burnt products expand. This, together with the volume change, compresses the unburnt mixture ahead of the flame front. This compression increases the pressure and temperature of the unburnt mixture. This implies that when the combustion process would be characterized using the regimes explained in 3.3.2, it may shift from one regime to another.

When considering time scales for a SI engine, intervals are mostly expressed in crank angle degrees instead of absolute time units. The total burning time of the mixture is then divided in two parts (figure 3.11). The first part is the flame development angle $\Delta \theta_d$. In this interval -expressed in crank angle degrees- the initial spark from the ignition develops into a self-sustaining flame kernel. The second part is called the rapid burning angle $\Delta \theta_b$. 
In this part, the major part of the fuel is burned. There is no clear separation between the intervals. Mostly, the point where a certain fraction (5-10%) of the charge mass has been burnt ($x_b = 5..10\%$) is taken as the separation between the flame development angle and rapid burning angle.

3.5 Theoretical dilution limit

One of the most important questions in this study is: what limits the mixture dilution in SI engines? The answer to this question lies in the fundamentals of the combustion process and may be related to flame temperatures that are too low in combination with heat loss, high stretch levels or a combination of these. Peters [64] states, for the laminar case, that flames cannot exist unless the adiabatic flame temperature $T_b$ is sufficiently larger than the inner layer temperature $T^0$ (see also figure 3.2). If the adiabatic flame temperature $T_b$ is too low, there will be too little diffusion of heat towards the preheat zone to increase the temperature to the inner layer temperature $T^0$ and reactions cannot take place. The limiting temperature in this case is somewhat higher as extinction occurs at a finite value of the burning velocity [64]. The actual flame temperature often is less than the adiabatic flame temperature due to heat loss from the reaction zone. Heat loss mechanisms may include radiation, mixing with colder gases etc. This makes the transport of heat to the pre-heat zone even more critical.

In turbulent combustion, the effects of flame stretch may also be of importance. Of course, this depends on the fuel used. Stretch influences the flame temperature through the Lewis (or Markstein) number as explained in equation 3.21.

3.6 Discussion

In this section, the theory for understanding premixed laminar and turbulent combustion was presented. Factors influencing the laminar burning velocity were discussed. Also for turbulent combustion a number of non-dimensional numbers was presented with which a
particular combustion process can be characterized. One of the main questions for this study, the existence and understanding of a mixture dilution limit, was discussed shortly. Mechanisms that limit combustion (heat loss, stretch) all have in common that these lower the flame temperature which in turn results in a heat transfer from the reaction zone to the pre-heat zone that is insufficient for sustaining flame propagation. The combustion limit in SI engines is also related to these issues. A first step in identifying the mechanism behind the dilution limit is to determine the regime in which the combustion takes place.
Chapter 4

Experimental engine set-up

4.1 Engine design

For the research presented in this thesis, a special research engine set-up was realized. The engine was initially designed for performing flow measurements in a different study. The initial design therefore was not part of this work. The engine, in its initial configuration, was not actually used prior to the start of this work. For performing combustion research, the engine was adapted to enable fired operation on natural gas. The contribution in this work consisted of a heavy modification and a re-design of sections of the engine to make it suitable for the experiments foreseen in this study. The development of the engine up to a sufficient level of reliability was also part of the process. In this chapter, this experimental set-up will be introduced. Some design features specific to this engine will be described, and also some critical limitations encountered during operation of this engine will be analyzed. The engine set-up was designed with multiple purposes in mind. With this engine, it should be possible to perform 'normal', (continuously) fired operation, useable for characterizing engine combustion from the measured in-cylinder pressure signal. Next to this, the engine was designed to provide optical access to the combustion chamber to facilitate the recording of the combustion phenomena through a window in the piston crown and also to allow the measurement of the flow characteristics in the combustion chamber using either PIV or LDA.

4.1.1 Characteristic dimensions

The experimental set-up consists of a single-cylinder SI engine which was specially adapted for natural gas operation. As this research is aimed at Heavy-Duty engines (engines that run at a high specific load for a prolonged period), the construction of the test engine should reflect the geometry of typical Heavy-Duty engines. In the world there is a limited number of natural gas engines in operation. This number is too small to justify the large expenses of developing a dedicated engine. Therefore, heavy-duty natural gas engines are mostly conversions of existing diesel engines. This implies that the dimensions and geometry of these engines are similar to diesel engines. For this engine set-up, it is therefore a logical choice to use an existing diesel engine as a start. The engine base or bottom end was derived from a single cylinder Heavy-Duty diesel research engine (type "Proteus", designed by Ricardo). By using this engine base, peak cylinder pressures up to 150 bars present no problem for the construction. The stroke of
this engine is 156 mm. The cylinder bore dimension was set at 130 mm which is a typical value for a heavy-duty engine of this size. The resulting displaced volume is 2.07 liters. This geometry is also typical for modern Heavy-Duty engines (in which there is a trend towards higher specific output rather than a larger engine).

4.1.2 Mechanical design features

Because of the special demands for this research, i.e. optical access, flexibility in operating conditions, easy maintenance, the base engine was adapted in a number of ways. In the following section, a number of design features unique to this engine will be described in more detail.

Hydraulic force transfer

The set-up was designed in a 'modular' way, the lay-out is such that it is possible to use different cylinder heads with only minor modifications to the construction. In this engine, the cylinder head is not bolted to the engine block. Instead it is bolted to an extension of the engine base. The upper liner is suspended in a 'hydraulic piston' and hydraulic pressure is used to force this upper liner against the cylinder head to provide a stiff construction to guide the piston (figure 4.1). An O-ring system is used as a gasket against the cylinder pressure between the sliding liner and the cylinder head. An additional advantage of this construction is the possibility to inspect or clean the top part of the liner and/or windows or perform work on the piston rings without having to remove the cylinder head. The side plates of the engine support the hydraulic forces. These side plates have been constructed very heavy, so that minimal extension of the construction results from the combustion load. Also, the inertia of the large mass helps to dampen residual vibrations.

For this research, the cylinder head from a DAF MX heavy-duty diesel engine was used. This cylinder head consists of a single casting for six cylinders which are fed through a common intake manifold which is incorporated into the cylinder head casting. With the current set-up, the first cylinder from this cylinder head is used, the remaining inlet runners have been closed at the valve seats.

Heavy-duty diesel engines typically employ a swirl geometry for the inlet runner design. In the current cylinder head, one of the runners acts as a swirl port, while the other functions as a tangential port. These ports were designed to generate a moderate amount of swirl (typical swirl number RS 0.5). Theoretically it would be possible to influence the amount of swirl by manipulating the inlet runners. However, as the focus of this research is on combustion this was not done.

”Bowditch” construction for optical access

To provide optical access to the combustion chamber, a Bowditch construction using an elongated piston and extended liner has been designed on top of the base engine (figure 4.1). Due to the construction with a hydraulic piston and cylinder, it was necessary to divide the extended liner in two parts. The lower liner is fixed to the crankcase. The upper liner is positioned in line with the lower liner by the hydraulic piston and cylinder. As usual in a Bowditch construction, a 45° mirror provides optical access to the combustion
Figure 4.1: Engine cross-section, full-metal configuration
chamber through a window in the piston crown (figure 4.11). The top part of the upper liner is designed as a separate piece. As a second means of optical access, this part can be replaced by a transparent liner. Using this transparent liner, it is possible to view the combustion chamber from the side, for the part of the cycle that the piston crown does not obstruct the view through the liner window. A second application of the liner window is the measurement of the in-cylinder flow phenomena. Using the PIV technique, a planar laser sheet can enter through the liner window. Light, scattered from particles in this laser sheet can then be recorded in the perpendicular direction through the piston window. For measurements which do not require optical access, both the piston window and the top part of the upper liner can be replaced by full-metal parts. This will be further explained in sections 4.2 and 4.3.

Overhead camshaft

In most heavy-duty diesel engines, the valves are actuated using pushrods and rockers. As pushrods tend to be rather heavy and elastic for large lengths, this is not a very practical mechanism for an engine that uses an elongated piston and liner. Therefore it was chosen to design a valve actuating mechanism using an overhead camshaft. The camshaft is driven from the crankshaft using a belt. This construction gives the user the freedom to change the valve timing by changing the position of the belt-tensioner wheels. For the experiments described in this thesis, the standard valve timing from the diesel engine was used.

Engine balancing

The original version of the base engine used a large mass to decrease vibrations. For this study, where the positioning of a laser sheet and the camera field-of-view should be as accurate as possible, the reduction of vibrations is of great importance. To achieve this, the engine base was modified by adding a set of balancing shafts (see figure 4.2) which rotate at the crank speed, but in a direction opposite to the crankshaft itself. These balancing shafts, together with the crankshaft counterweights compensate for the inertia forces of the piston motion in the vertical plane (figure 4.2(a)). As the crankshaft and the balancing shafts counter-rotate, the inertia forces in the horizontal plane cancel out as indicated in figure 4.2(b). With this set of balancing shafts, the engine is fully compensated for first order unbalance. This, together with the large mass, results in an engine oscillating motion of no more than $O(1 \text{mm})$. Only at a limited number of engine speeds, a resonance can be detected which propagates through the floor of the engine test cells.

Integration of dynamometer and electric motor

The engine set-up was integrated with both a dynamometer (Schenck W450 eddy current brake) and an electric motor. When the engine is motored only, the engine speed is kept constant with a frequency controlled inverter. When the engine itself is running, the engine speed (or load) is kept constant with the dynamometer. During the initial motoring tests, it was found that the motoring torque was substantially larger than expected. This would mean that there exists a certain minimum engine load for fired operation, as the engine should at least produce the torque to turn itself over. To circumvent this problem, a
A different frequency inverter was installed which made it possible to use the electric motor in a constant-torque mode, additional to the constant-speed mode. The electric motor would then apply a torque larger than what is required to motor the engine. The excess torque is dissipated by the dynamometer. To calculate the generated power when the engine starts firing, one can simply use the additional torque which is measured by the dynamometer. An added benefit of this mode of operation is that transients in engine load do not lead to large variations in engine speed as the relative change in torque the dynamometer has to adapt to is less. One variable that is not accounted for is the change in engine friction as the engine starts firing. Values for typical engine friction can be found in literature ([72]), however these were considered not realistic due to the different construction of this engine. In practice, it turned out to be more convenient and also more accurate to set the engine load using the indicated load (IMEP) instead of the load measured by the dynamometer. This IMEP-based load setting was used for all measurements in this thesis.

### Engine friction and lubrication

In a normal engine, a single piston fulfils both the function of sealing the combustion chamber against gas leakage, as well as supporting the thrust forces. With a Bowditch construction however, the piston is extended so these functions are fulfilled by different parts of the piston. When constructing an optically accessible engine with this construction, one of the challenges is the lubrication of the piston/liner contact. In a normal engine, there is an oil film on the liner wall which prevents direct contact of piston and liner. This oil film also helps the piston rings to seal the combustion chamber. To prevent the oil from entering the combustion chamber, oil control rings are used which scrape the oil from the liner wall and return it to the engine sump. These oil control rings are assisted by high pressure blow-by gases which help to transport the oil to the sump. With this extended piston construction however the situation is different. The upper and lower part of the piston both present a number of problems. The lower part of the piston which supports the thrust forces is lubricated in a conventional manner through oil that slings from the crankshaft and its counterweights. An additional oil jet is directed at the small-end bearing. The original piston was a one piece design which used a bronze sleeve on the
The lower part to support the thrust forces of the piston (figure 4.4(a)). The lower part of the piston was in addition provided with a set of two conventional oil control rings. It was found however that, for this engine, the amount of oil on the lower liner wall is so large that the oil control rings cannot keep up with it and oil passes these rings. This was partly due to the excessive oil deposition on the liner wall below the piston, but also due to the absence of blow-by gases flowing downward along the lower piston part. As the space in the extended piston should remain as oil-free as possible to prevent contamination of the mirror with oil, the oil has to be removed before it reaches the space between the upper and lower parts of the piston. A vacuum pump was used for this, which draws an air stream through a narrow slit which is formed by an additional ring mounted on top of the lower liner. This construction is shown in figure 4.3(a). Any oil that passes the oil control rings is removed this way. Using this oil vacuum system, it was possible to operate the engine for approximately one hour without cleaning the mirror.

The upper part of the piston presents a different problem. The contact of the piston with the upper liner should be lubricated also, however there is no oil there. To circumvent the use of an oil based lubricant, the piston is supported against the upper liner by a ring made of graphite-filled sinter bronze (CuSn8413/12PF-3mot3) which provides dry lubrication using graphite. Also the two compression rings are constructed from this material. Behind these compression rings, an expander spring is mounted which increases the contact pressure between the compression ring and liner surface. The surface of both the upper and lower liner was coated with PowerSeal® (a brand name for Nicasil), which consists of a nickel layer with silicon carbide particles in it. This layer is very hard and wear resistant.

As already mentioned in the previous section, the torque required to motor the engine was larger than expected. To determine the cause of this friction, motoring tests were performed to measure the contribution of each component (crank assembly, camshaft drive, piston rings, compression work etc.) to the total engine motoring torque. Here it was
found that the friction of the crank assembly was rather large. Unfortunately, no direct cause could be found. At a later stage, when the engine was firing under load, the heat resulting from friction at the lower piston-liner contact caused the lower part of the piston to expand excessively and to seize in the lower cylinder. In an attempt to reduce friction and to increase the hydrodynamic pressure in the lubricated contact, the bronze sleeve on the lower piston part was modified to a barrel shape alike a standard piston skirt and more testing was done. After a third seizure it was found that the lower part of the piston was deformed too much by the excessive forces and temperatures that occur during a seizure; this made the piston unusable for further work. It was therefore decided to re-design the piston. The new construction employed the skirt of an existing modern steel truck piston, with the extended part of the original construction bolted onto it (figure 4.4(b)).

During the seizures, also the surface layer of the lower cylinder was damaged due to the excessive forces. This lower cylinder was therefore modified so a standard cast-iron truck sleeve could be fitted. A further reason for this modification was the better match in running characteristics of the surfaces of the truck piston skirt and the cast iron liner as these came from the same engine type. Due to the slightly different shape of the standard truck cylinder liner, also the oil vacuum system had to be modified. The oil collecting ring on top of the lower liner was removed and replaced by a series of holes in the liner wall. These holes were positioned at the top-most position of the oil control rings (figure 4.3(b)). Locating the holes at that location maximizes the period available for removing the oil from the oil control rings as the piston speed is relatively low at TDC.

An additional problem was the lubrication of the small-end bearing. Due to oil starvation resulting from an early attempt to reduce the flow of oil across the oil control rings, the original bearing showed excessive wear resulting in free play between bearing and wrist pin, ultimately leading to contact between the piston crown and the cylinder head. To prevent this from happening again, the clearance height was increased and an oil jet was installed to provide adequate oil supply for the small-end bearing and also to remove heat from the lower piston part and bearing. The small-end bearing itself was also re-designed to improve the distribution of oil in the bearing. A standard truck engine small-end bearing was used as an example for the dimensions and geometry of the oil distribution grooves in the bearing inner surface.

After these modifications, tests showed that the motoring torque had reduced from about 60 Nm to approximately 40 Nm. This value is still larger than what is found in standard truck engines but since there are more, and also heavier parts moving and rotating for the same engine capacity this friction torque was considered acceptable.

When at a later stage the engine load and -speed were steadily increased, problems returned in the form of a combined seizure of the lower piston/liner contact and the small-end bearing. The simultaneous occurrence of these problems made it very difficult to find their cause. From the type of damage it could not be determined unambiguously which lubricated contact generated the heat leading to failure of both components. It was most likely that heat from the seizing piston-liner contact overheated the wrist pin causing it to expand excessively and press away the oil film in the small-end bearing. Alternatively, the small-end bearing could generate heat causing the skirt to expand excessively. As the piston skirt and lower liner were not damaged beyond repair, a modified small-end bearing with a slightly larger clearance was installed. Only after extensive fired operation at high load, it was found -again through seizure of the lower piston skirt- that the overall shape of the piston was deformed too much. The entire piston had deformed leading
to a misalignment of the upper and lower part of the piston. The cause of this seizure was attributed to a difference in operating temperature between the truck engine that donated the piston skirt and lower liner and this engine. Although the lower part of the piston does not run very hot, due to the way the oil vacuum system is incorporated in the lower cylinder, this lower liner has a low temperature leading to a reduced running clearance. After a final modification in which the piston was rebuilt using a new skirt for the lower part and where also the clearance between the piston skirt and lower liner was increased by increasing the bore diameter of the lower liner, no more problems occurred. The motoring torque was unfortunately not measured again due to time limitations.

4.1.3 Intake, fueling and exhaust system

In this section the systems used to provide air and fuel are described. Also the layout of the exhaust system is shown. A schematic of the engine setup is shown in figure 4.5.

**Air delivery**

The test engine has been designed for intake pressures up to 3 bars. As there is no turbocharger present, the air supply for the engine is provided by a separate compressor. To provide a constant supply pressure, a large vessel (2 cubic meters) is located between the compressor and the engine. The pressure in this vessel is set somewhat above the desired intake pressure. An electronically controlled throttle valve then adjusts the flow to the engine such that the intake pressure is kept at a constant value even when the flow increases or decreases due to changing engine working conditions. The air mass flow
Figure 4.5: Schematic view of test stand

is measured using a coriolis type mass-flow meter located in between the settling tank and the throttle valve. The pressure and temperature of the intake air are measured at the entrance to the cylinder head. As an extra means of controlling the engine intake conditions, an electric heater is installed in the intake section. This heater however was suffering from operational problems and was not used for the experiments described in this thesis.

Mixture formation

As this research is on the phenomena occurring in highly diluted premixed combustion, the mixture that enters the engine cylinder should be as homogeneous as possible. Bearing in mind that the current trend in fuel metering is towards fuel injection per cylinder (to have more control over the air/fuel ratio), the fuel metering system chosen for this engine is sequential gaseous fuel injection. This makes a possible (future) switch to a port fueling strategy easy to implement. For the experiments described however, in which a homogeneous mixture is used, 6 fuel injectors have been located at a single location upstream of the intake manifold. In between the fuel injector body and the engine, a static mixer and a mixing tank are provided. The feeding pressure for the injectors was chosen at 5.5 bars, so that for intake pressures up to 3 bars (absolute pressure) sonic flow occurs in the fuel injectors. The fuel injectors are dimensioned such that the gas injection pulses of these 6 injectors follow up or even overlap as all 6 injectors inject once every 2 engine revolutions. In this way, a quasi-continuous gas flow is created. The injection duration is determined by the engine control unit (MoTec M8 ECU). The actual (average) fuel flow is measured using a coriolis type mass-flow meter which is located in between the high and low pressure reducers. This location was chosen so as not to create an additional pressure
drop due to the flow meter. This ensures that the feeding pressure for the injectors is independent of the fuel flow rate. This injection pressure was also monitored.

**Exhaust system**

As only one cylinder is used, the engine exhaust system consists of a single pipe that connects to the cylinder head. The exhaust gases are collected in a mixing tank, mounted on top of the engine before leaving the test cell. This mixing tank has a dual purpose. Firstly, it allows emissions measurements of the time averaged composition of the exhaust gases (see section 5.4). Secondly, when using mixture dilution with EGR instead of air in a follow-up investigation, the necessary exhaust gases can be extracted from the mixing tank. The pressure and temperature of the exhaust flow are measured at the entrance of the mixing tank. At the exit of the mixing tank, a butterfly valve is placed. The opening angle of this valve is controlled by an electric actuator. As this valve throttles the flow of exhaust gases from the mixing tank, it can be used to set the engine back pressure. For engine working points in which the intake manifold pressure is below atmospheric pressure, the back-pressure valve is fully open. For working points in which the intake manifold pressure is increased, the engine back-pressure is kept equal to the intake manifold pressure to simulate the presence of a turbocharger and to provide operating conditions as close to a normal engine as possible.

### 4.2 Design and operation of the 'full-metal' version

#### 4.2.1 Combustion chamber

As already pointed out in chapter 2, SI engines for natural gas can utilize a higher compression ratio than gasoline engines due to the higher knock resistance of natural gas. As found in the literature, the compression ratio of natural gas engines varies roughly between 10 and 14, depending on the fuel composition. For this research engine, the compression ratio can be adjusted by modifying the size of the TDC compression volume. This volume can be adjusted either by changing the piston bowl or the TDC clearance distance. As an intermediate value, a compression ratio of 12.5 was chosen. This choice fixes the size of the TDC compression volume. The compression volume can still be shaped in a number of ways, which has a large effect on the flow and turbulence of the mixture. The shape of the compression volume can be as simple as a flat top (‘pancake’ combustion chamber), or very complex with a carefully shaped flow pattern. For the research described in this thesis it was decided to keep the piston bowl relatively simple, but at the same time to utilize the squish effect to create additional turbulence. As mentioned earlier, gas engines are often derived from diesel engines. Therefore, the shape of the piston bowl was derived from a ‘diesel bowl’ which was made deeper to lower the compression ratio to 12.5. This choice resulted in a rather shallow bowl which has a squish ratio (defined as the ratio of the squish area to the bore area) of 28%. This design compression ratio was lowered to 11.25 in a later stage however by increasing the clearance height. At that stage, also the bowl was redesigned to provide better piston cooling. This redesign increased the squish ratio to 50% (see section 4.2.2). Further information on this can be found in chapter 5.
During the measurements in which optical access was used, the compression ratio was reduced further to 9.5. This is described in chapter 8.

4.2.2 Piston cooling system

Due to their special construction, engines with optical access are often severely limited in their maximum load. Limited engine cooling, or no cooling at all leads to an excessive temperature of the piston top and cylinder walls when the engine is fired continuously. These high temperatures are not realistic (i.e. the temperature levels are not representative for a normal engine), and also cause material and lubrication problems. In practice, when studying combustion in such an engine, its operation mode is limited to skip fire operation. After each combustion period, which can be as short as a single cycle, the engine is given time to cool down again. Usually, a separate, more standard engine is then used for continuously fired operation. However, when these engines are not identical, the combustion process studied using the optically accessible engine can be different from the process in the continuously fired engine. In this research, the same engine is used both for continuously fired operation and for operation using optical access.

For continuously fired operation, the piston window is replaced by a metal bowl with the same combustion chamber geometry. Also, the transparent upper part of the liner is replaced by a water cooled version. To remove the heat from combustion, a method for cooling the piston is needed. Several designs for piston coolers have been constructed and tested. In the following subsection, these designs will be treated briefly. After that, for the final design, the theoretical cooling performance will be estimated and compared to the measured performance. Finally, the practical implications of this on the set-up will be evaluated.

Designs tested

The design of a piston cooler was initiated by what seemed to be self ignition of the mixture inside the combustion chamber. Later however, this turned out to be caused by a spark plug firing control error in the engine management system. Still, despite no self ignition of the mixture would occur, the piston temperatures measured were higher than in a normal engine and a lower temperature level was highly desirable.

The first bowl design that was fitted on top of the piston was a steel bowl in which only the bowl depth was increased to lower the compression ratio from the diesel value (16) to a more modest value of 12.5. Essentially, a metal version of the window was used which was constructed strong enough to withstand the pressure and to have the same mass as the intended optical window so the engine balancing would not be disturbed (figure 4.6(a)). The discoloration of the bowl material induced during firing indicated that the surface temperature (very likely) exceeded that of a normal piston. Therefore the underside of the piston bowl was cooled with a jet of compressed air and at a later stage with a combination of compressed air and a water spray. In this set-up, an infrared thermometer was mounted at the location of the mirror, facing the underside of the piston bowl. This thermometer measured the surface temperature of the center area of the underside of the bowl (see figure 4.7). It turned out that these cooling methods provided insufficient cooling to the piston bowl (the bottom surface reached temperatures exceeding 300 °C, indicating an even higher temperature for the top surface and upper edge of the bowl).
A second steel bowl was then made with an increased number of cooling fins to provide a larger cooling surface (figure 4.6(b)). Temperature measurements showed a somewhat lower temperature for the underside of this bowl at the same engine load but the temperature level was still too high.

It is hardly possible to model this cooling process, since the heat transfer of an impinging jet of air and water is not very well known. At this stage, to limit the number of trials, a case study using FEM modeling was conducted to find out which parameters have the most influence on the flow of heat from the bowl upper surface which is exposed to combustion to the cooling medium. This study showed that the heat transfer coefficient at the bottom surface of the bowl does not influence the total heat transfer very much. The limiting factor in this process is the relatively low heat conduction of the stainless steel that the bowl was made from. Therefore it was decided to design a piston bowl from aluminum. Because of the lower strength of this material, the design is different from a steel bowl. Several design variations were modeled to find an optimum in terms of sufficient strength and stiffness and low heat resistance. These design variations employed different patterns of cooling channels from the bottom surface into the bowl material. From these calculations it was found that it is advantageous to drill channels up to the hottest place, which is the edge of the bowl. To make this possible, the bowl shape was changed to a deeper bowl with a smaller diameter, which led to a higher squish ratio of 50%. Together with this material change, it was also decided to change to a 'closed' cooling system using a liquid cooling medium. Two different liquid cooled designs were tested. The first of these (figure 4.8) consisted of an aluminum piston bowl which had holes drilled from the bottom upward around the actual bowl so as to allow the liquid to access the hottest places (aluminum piston bowl I, CR=11.25). Under this bowl in the space inside the piston, a closed aluminum pipe was mounted which functions as a shaker in which the liquid splashes up and down as the piston oscillates. The liquid splashes into the holes in the bottom of the bowl (see enlarged section in figure 4.8) where it is heated up and, depending on the temperature, partly evaporates. This hot liquid is then cooled.

Figure 4.6: Piston bowls, steel designs
Design and operation of the ‘full-metal’ version

Figure 4.7: Temperature measurement on steel piston bowls; steel piston bowl II

again at the walls of the aluminum pipe which in turn is cooled by a cold air stream. The working principle here is to bring a cooling medium with a larger heat capacity to the hottest places and to provide a larger cooling surface to the surroundings. However, it was found that the viscous dissipation occurring in this fluid shaker was heating up the liquid even when the engine was only motored and not receiving heat from the piston bowl. Therefore a second, more complicated design was realized. A bundle of thin walled pipes was installed in the space inside the elongated piston. These pipes are connected at the top and bottom by a chamber. In this bundle (figure 4.9), fluid is circulated. The fluid circulation is induced by the inertia of the fluid itself as the entire heat exchanger moves up and down with the piston, assisted by non-return valves in the pipes. The fluid travels upward through the outer layer of pipes and downward through the inner pipes. The heat from the piston bowl is transferred to the fluid which splashes into the holes in the bottom of the bowl. The fluid is cooled again when it flows through the pipes. These pipes have a ribbed outer surface and are cooled by a cold air stream which is forced through the pipe bundle. Also, the drilling pattern in the aluminum piston bowl was modified in such a way that channels allow the fluid to reach just below the upper surface of the bowl edge (aluminum piston bowl II, CR=11.25). This is shown in the enlarged sections of figure 4.9.

Cooling performance

The development of the various piston bowls as described in section 4.2.2 was performed more or less on a ‘trial and error’ basis. However, since the final design seemed promising,
Figure 4.8: First version piston cooling system; aluminum piston bowl I, CR=11.25

Figure 4.9: Final version piston cooling system; aluminum piston bowl II, CR=11.25
its cooling performance was investigated in more detail. It was assumed that the heat transfer from the pipes to the air flowing through the cooler is limiting the total heat transferred. The temperature of the fluid in the cooling system is also limited. As this fluid is mostly water, boiling of the fluid and a subsequent pressure rise in the cooling system results. An attempt was made to calculate the amount of heat transferred using Nusselt relations for flow through a cylinder array. But, as this configuration is far from standard, only a fairly rough estimation could be made. Assuming a surface temperature for the pipes of 100 °C, the heat transfer was estimated at O(10 kW). More information on this can be found in appendix G.

During engine operation, the actual heat transferred was calculated from measurements of the cooling air flow rate and the temperature difference between entry and exit of the cooling section. From the measurements, a maximum heat transfer rate of 11 kW was found.

A careful series of experiments in which the engine load and speed were varied have shown that the amount of heat transferred to the piston cooling system increases with the thermal power of the engine (i.e. the power based on the calculated IMEP value). This is shown in figure 4.10(a). The surface temperature of the pipes increases with the engine thermal power and also with increasing engine speed. This is shown in figure 4.10(b). At higher engine speed, the transfer efficiency increases. A larger fraction of the thermal power is then transferred to the piston cooling system. This increased transfer efficiency can be attributed to the higher flow velocity of the cold air stream over the outside of the pipes due to piston motion as well as a more effective 'pumping' of the fluid through the pipes as the inertia forces increase with the square of engine speed.

Maximum engine load and firing strategy

The maximum heat transfer as explained in section 4.2.2 implies that there is a practical limit to the engine load that can be used continuously. During testing it was found that the maximum allowable engine power (thermal, based on IMEP) that could be used safely
was about 20 kW, during continuous firing. At this power rating, the surface temperature of the cooling pipes increases to values above 90 °C (figure 4.10(b)). This translates into a lower specific load at higher engine speed. When a higher specific engine load is desired, the firing strategy must be changed to skip-fire or burst-fire mode. With skip-fire, the engine fires one cycle, followed by a number of non-fired cycles to allow the piston top and combustion chamber to cool down. With burst-fire, the engine is fired for a certain amount of cycles in succession (typically $O(100)$ cycles). During this firing period, the walls of the combustion chamber and the piston surface will heat up. The limit for this firing strategy is still the temperature of the fluid in the piston cooler. The temperature of the fluid lags the heat input from combustion by a small amount of time, due to the instationary heat conduction in the aluminum piston bowl.

When using skip- or burst-fire, the measurement of quantities like emissions becomes more difficult. With skip-fire, the measurement of emissions is hardly possible, since the exhaust gases are contaminated with unburnt fuel from the cycles that were skipped. This is unavoidable since the fuel and air are fully premixed upon entering the engine. For the burst-fire strategy, the measurement of emissions data can only be started when the engine has stabilized its working point (engine speed, intake pressure etc.). This means that a firing period should be somewhat longer than required for the measurement time.

### 4.3 Design and operation of the 'optical' version

#### 4.3.1 Piston window and mirror

As already mentioned in section 4.1.2, the engine features a Bowditch-type elongated piston that provides optical access to the combustion chamber through a window in the piston crown. This window forms the bottom of the piston bowl, so that the geometry of the combustion chamber remains the same as in the full-metal version. The material choice for this window depends on the measuring technique used. For recording flame images, any material that is optically transparent for visible light will do. However, when using Laser Induced Fluorescence sometimes transparency for UV light is needed. This limits the material choice to quartz glass. The different optical materials have different strengths and construction limitations (sensitivity to cracking), so the thickness of the window has to be adjusted to the material choice. For the measurements described in this thesis where UV transparency was not required, a sapphire window with a thickness of 25mm was used.

#### 4.3.2 Liner window

As a second means of optical access, a part of the cylinder liner can be replaced by a transparent ring. Due to the more stressful conditions for the liner, sapphire was chosen for this part because of the higher strength. This transparent liner can be used either to view the processes inside the combustion chamber, or to introduce a laser light-sheet for use with LIF or PIV. Also, the application of LDA is possible through this transparent liner. Due to high costs the height of this transparent part was limited to 25 millimeters. This dimension is more than sufficient for introducing a laser sheet, and also an LDA probe can reach the complete combustion chamber. If there is a need to introduce a laser sheet at a position other than the upper 25 mm of the combustion chamber, the
Figure 4.11: Cross section showing mirror placement, piston window and stacking options for the upper liner part.
Experimental engine set-up

4.3.3 Engine load limitation

As already mentioned in section 4.2.2, the amount of cycles the engine can fire continuously is limited by the amount of heat that can be accumulated by and/or removed from the piston crown. Whereas with the full-metal version there is a cooling system which cools the top of the piston, this cooling system is not present when using optical access. When using optical access with a piston window and/or liner window, the accumulation of heat in these windows becomes limiting. As the window materials have a poor heat conduction (sapphire: 40 W/mK) compared to aluminum alloys (150 W/mK), the heat input from combustion will introduce a more pronounced transient temperature profile in the window. On top of the temperature profile, close to the window surface even cyclic variations in temperature occur. These temperature variations give rise to thermal stress in the window material which limits the amount and duration of heat input that is allowed. Sapphire is the preferable material for this application (compared to quartz) due to its combination of higher thermal conductivity and allowable stress.

4.4 Instrumentation, data acquisition and engine control

Instrumentation

The condition of the engine is monitored using a number of sensors. These sensors monitor a number of slowly varying signals like the average intake- and exhaust pressure, the temperatures in the intake- and exhaust system, average engine speed and dynamometer torque. Dedicated sensors are used for monitoring fast changing quantities like the cylinder pressure.

Data acquisition

The test set-up is equipped with 2 separate data acquisition systems. The first, a TueDacs system, takes care of slowly varying signals. This system monitors the complete set-up and registers variables like temperatures and average pressures of the intake and exhaust system and the various flow rates. The second acquisition system, a Fevis 3.0 engine indicating system, takes care of the recording of signals that vary within an engine cycle. This system registers signals on a crank angle basis with a resolution of 0.1 degrees. The timing pulses needed for this are provided by an encoder mounted on the crankshaft. The most important quantity to be registered is the cylinder pressure signal. Further signals that are registered by the Fevis system are the various emission components from the fast response emission sampling system (see section 5.4) and the intake manifold pressure which is used for cylinder pressure referencing.
Engine control

The engine working condition is controlled partly manual, with the assistance of an engine management system (MoTec M8). As already mentioned in sections 4.1.3, the intake pressure is controlled by an electronically actuated throttle valve with feedback from the actual intake pressure. The amount of fuel is determined using a lookup table (map) in the engine ECU which uses the engine speed and intake pressure as parameters. Manual control of the air/fuel ratio is possible by specifying a correction factor to this map.

4.5 Emissions measurement

Because this engine set-up is more limited in its operating strategies and operating duration than a 'standard' engine set-up, the measurement of the various emission components is also more difficult. The set-up is equipped with a fast response analyzer (Horiba MEXA 1400FR). This fast response analyzer features a very small sampling cell, which makes the sampling time very short ($O(\text{ms})$). If the exhaust gases are sampled at a location close to the exhaust valves, emissions concentrations can be separated per engine cycle. The emission sampling system consists of 3 separate analyzers:

- The FID analyzer measures the concentration of hydrocarbons in the exhaust gases by burning these in a hydrogen flame. This process generates some ions. Since the flame is placed between two electrodes, the ions generate a small electric current which is a direct measure for the amount of ions generated. This process effectively counts C-atoms and therefore cannot distinguish between lighter and heavier hydrocarbon compounds. The output signal is the C1 equivalent of the hydrocarbon concentration.

- The Chemiluminescence analyzer (CLA) measures the concentration of nitrogen oxide ($\text{NO}$). The nitrogen oxides from a SI engine consist mostly of $\text{NO}$.

- The Non-Dispersive Infrared analyzer (NDIR) measures the concentration of carbon dioxide ($\text{CO}_2$), carbon monoxide ($\text{CO}$) and hydrocarbons. The physical process employed is the adsorption of infrared light by these species.

Some additional information on the fast response analyzer set-up can be found in appendix A. The presence of a sampling line introduces an additional delay in response time which has to be accounted for when matching emissions measurements with engine cycles. This however introduces no problems, as long as the exhaust gases sampled at different engine cycles do not mix in the sampling line. Due to the thin sampling lines, care has to be taken to avoid condensation of the water vapor from the exhaust gas inside the sampling lines as certain gases can dissolve in this condensed water. The sampling line is therefore heated electrically and is kept at a temperature of $190^\circ\text{C}$.

As an alternative to the measurement of emissions per engine cycle, a second sampling point is provided downstream of the exhaust mixing tank. Using this sampling point, the same equipment can be used to measure the time-averaged emissions concentration. Besides this fast response analyzer, also a Hartmann & Braun analyzer rack was present. This rack contained a paramagnetic oxygen analyzer (Magnos 3) and a combined $\text{CO}/\text{CO}_2$ NDIR analyzer (Uras 3K). These analyzers are much slower than the fast response analyzer and are only suitable to measure the time-averaged concentration. The measured
oxygen concentration is used in the calculation of the equivalence ratio when an oxygen balance method is used. Additional to emissions analyzers, a UEGO sensor (Universal Exhaust Gas Oxygen sensor) is mounted in the exhaust downstream of the mixing tank. This sensor measures the oxygen concentration in the exhaust gases from which the mixture equivalence ratio can be calculated. The readout of this sensor is used as a visual aid to adjust the fueling correction of the engine management system and also as a check for the equivalence ratio as calculated from the air and fuel mass flows.

4.6 Discussion

The experimental engine set-up described in this chapter presents a useful tool in the investigation of lean combustion in gas engines. It features a realistic heavy-duty engine geometry which is the same for the full-metal version and the version providing optical access. The engine load that can be used is representative for a heavy-duty gas engine at medium load. The maximum load that can be used in continuous firing mode is -for an engine featuring a Bowditch construction- fairly high. The optical access provided is sufficient for imaging a reasonable part of the combustion chamber (80 millimeters) and can be used for imaging the combustion as well as for performing velocity and/or turbulence measurements. The emissions measurement system provides the possibility to measure the average composition as well as to evaluate the emissions per cycle using a fast response emissions analyzer.
Chapter 5

Engine tests using lean mixtures

5.1 Introduction

The combustion characteristics of the engine were investigated using a number of fired engine experiments. For these experiments, the engine was equipped with a cooled (upper) liner and cooled piston without optical access, as described in section 4.2. For this test series, the mixture was made leaner until combustion instabilities occurred. The onset of these instabilities was marked by a sharp rise in the emission of unburned hydrocarbons as well as an increase in the IMEP variation coefficient (COVimep). In this chapter, the results of this test series will be presented. First, an overview is given of the test conditions (working points, fuel etc.). After that, the methods and techniques used to transform the raw measurement data into meaningful results will be explained.

5.2 Test conditions

The composition of natural gas is not the same in every location in the Netherlands, also there are variations in time. For this work, methane was therefore chosen as a reference fuel. However, first a full series was conducted with DNG (Dutch Natural Gas) as a fuel. The test matrix consisted of three engine speeds and three engine loads as indicated in table 5.1. The tests with 8 and 12 bars IMEP were not performed for the lowest engine speed, since this combination does not occur in normal engine operation for engines of this size. All tests were conducted with full continuous fired operation. After completion, the same test series was repeated with pure methane as a fuel. Due to a piston seizure at full load during the DNG series, it was decided to perform the high load/speed tests with methane in a burst fire mode. In this mode, the engine was run in a 1:1 skip-fire mode to keep it at a reasonable operating temperature and was then run continuously for a short period in which the measurements were taken. This strategy and its implications for emissions measurement will be treated in more detail later. Also, for the methane series the highest engine speed was reduced to 1350 rpm for reasons of safety as a resonance in the engine was detected at 1400 rpm.

Ideally, to generate consistent data, one would adjust the ignition timing to MBT (Maximum Brake Torque). As this is a single cylinder engine running at low speed it is not easy to find this setting because the engine torque fluctuates continuously with every cycle. As an alternative, the ignition timing was adjusted such that the maximum
cylinder pressure occurred around 8 degrees ATDC.

### 5.3 Engine set-up for fired operation

The engine set-up as used for the fired engine tests was described already in chapter 4. A schematic representation of the setup can be found in figure 4.5. The following text therefore refers to figure 4.5. In this figure, the main engine instrumentation is shown. A Fevis data acquisition system was used to log several signals on a crank angle basis. These signals are the cylinder pressure, the various emission components, the manifold pressure and the momentary engine speed (which is an internal channel and is calculated as the derivative of the encoder signal). Next to this, all signals relevant for monitoring the engine are recorded by a slower system (TueDacs) at a frequency of 2 Hertz.

### 5.4 Emissions measurement

For the measurement of engine-out emission levels, a fast-response analyzer was available (Horiba Mexa 1400FR). This instrument facilitates the measurement of emissions on a cycle-to-cycle basis. In this way, the increased emission levels from misfires or slow burning engine cycles can be distinguished. This possibility was used for some engine working points. The fast response analyzer consists of 3 modules for the measurement of unburned hydrocarbons (HC), nitrogen oxide (NO) and carbon monoxide (CO) and carbon dioxide (CO$_2$) respectively. Next to this fast response analyzer, a H&B (Hartmann & Braun) paramagnetic analyzer (Uras 3) was present which was used to measure the oxygen concentration in the exhaust gases. This analyzer also featured a module for carbon monoxide/carbon dioxide. Due to the slower response of this analyzer, it could not distinguish individual cycles.

With these restrictions in mind, the measurement of exhaust emissions could be performed in two modes:

- Measurement of $HC$, $NO$, $CO$, $CO_2$ an $O_2$ concentrations from the sample point after the exhaust back-pressure valve. The concentration is the time averaged concentration due to the mixing that takes place in the exhaust mixing tank and individual cycles cannot be distinguished. The measurement of the $CO$ and $CO_2$ concentration can be performed using either the Horiba or the H&B analyzer. These analyzers measured nearly identical $CO_2$ concentrations. The latter proved more suitable for the measurement of $CO$ as it provided a measurement range that was more suited to the very low levels of carbon monoxide from the lean combustion. The oxygen concentration was measured using the paramagnetic module in the H&B analyzer.
In this way, the equivalence ratio could be determined using either carbon- or oxygen balance techniques (see appendix B).

- Measurement of $HC$, $NO$, $CO$ and $CO_2$ concentrations in the exhaust port using the Horiba analyzer. With this method, individual cycles can be distinguished regarding the mentioned species. This method was used for the engine working points at which continuous operation was not possible due to limitations in the engine cooling and a burst fire mode had to be used. The measurement of the oxygen concentration can be performed simultaneously from the sample point after the back-pressure valve, but it is not necessarily representative for the engine operation at that particular moment due to the slow response of that analyzer. During the short sampling period ($O(100$ cycles)) the measured oxygen concentration has not reached a stable value yet. Determination of the equivalence ratio based on emission concentrations is therefore only possible using a carbon balance since this does not require the oxygen concentration.

For most engine working points, the more accurate determination of the equivalence ratio using both carbon and oxygen balance was considered more important than the emission concentration from individual cycles and all sampling was performed after the exhaust back-pressure valve. The series with DNG as a fuel were all performed using continuously fired engine operation and all emissions were sampled after the exhaust back-pressure valve. Only for the high load measurements with methane which were performed in burst-fire mode a switch was made to sampling in the exhaust port. A comparison between both modes was done at an intermediate engine load (8 bar IMEP) to check if both modes yielded comparable results. This proved to be the case.

As an illustration, the response times for the various emission analyzers are shown in figure 5.1(a). For this situation, all analyzers were installed at the sampling point downstream of the back-pressure valve. It can be seen that there is a delay time for the exhaust gases to reach the sampling point due to the traveling time in the sample line and the mixing in the exhaust mixing tank. The fast response analyzer shows the arrival of the exhaust gases relatively fast, after about 4 engine cycles the first rise in concentration is detected. After that, the concentration of $CO_2$ steadily increases whereas the concentration of unburned hydrocarbons decreases. The hydrocarbon analyzer was saturated as the motored engine operation was achieved by cutting the ignition and retaining the fuel flow. The concentration of $NO$ shows a sharp increase, after which it keeps rising but at a slower rate. As the mixing behavior of the various components is the same as they follow the same path to the analyzers, this is most likely caused by the larger amount of $NO$ that the first fired cycle generates. This first fired cycle contains no residual exhaust gases from the cycle preceding it and consequently reaches a higher temperature. Even when both $CO_2$ and hydrocarbons have reached a stable value (after about 30 cycles), the $NO$ concentration keeps rising at a slow rate. This may be caused by the steadily increasing temperature of the gases in the combustion chamber as the walls heat up with every burning cycle. In the figure, the oxygen concentration first shows no response at all. The indicated oxygen concentration is that of the unburned mixture that was flowing through the engine prior to switching on the ignition. The oxygen concentration only starts to decrease after about 60 engine cycles (corresponding to about 7 seconds at the current engine speed). This is caused both by the slower analyzer as well as by the long sampling line. It shows clearly that the measurement of the oxygen concentration is not
representative for the exhaust gases from the burst of fired cycles. Also, looking at the readings from the fast response analyzer, any calculation of air-fuel ratio can first take place after about 30 engine cycles and the carbon balance is the only method that can be used.

In figure 5.1(b), a similar graph is shown for the case when the fast response probe is installed in the exhaust port. In this case, there is only the delay of the shorter sample-line, and an additional delay for each separate analyzer. The measured concentrations rise rapidly after switching on the ignition and have stabilized already after 3 engine cycles. The delays between the emission of $HC$, $NO$, $CO_2$ and $CO$ and their respective detection by the fast response analyzer is a function of the length of the sample line as these analyzers use a constant volumetric flow rate for sampling. This flow rate is however partly driven by the exhaust pressure, therefore the delay resulting from the sample line is slightly less at higher exhaust pressures. The constructional details of the fast response sampling system can be found in appendix A. The additional delay of the measurement process in the respective analyzers is also a constant. This means that the measured signal can simply be shifted back in time to link the measured emission concentration to the engine cycle it originated from. The specifications for this analyzer give the response time as $t_{90}$ which is the time it takes to reach 90% of the correct reading. This time $t_{90}$ is
< 30 ms for all analyzers. Further, the manufacturer states that the sampling line traveling time $t_D$ is also $\leq 100$ ms for sampling lines with a length $\leq 500$ mm. The sampling line used is about 1000 mm, so a traveling time between 100 and 200 ms can be expected. The total delay, written as the sum of $t_{90}$ and $t_D$ can be estimated by shifting the signals such that the change in concentration occurs shortly after the exhaust valve opens. In figure 5.2, this process is illustrated. Figure 5.2(a) shows the raw concentration measured for a situation in which the engine was motored for 25 cycles and then fired for the same amount of cycles. In figure 5.2(b), these signals have been shifted by the correct amount. The vertical line indicates the moment the exhaust valve opens in the first fired cycle.

For the continuously fired engine working points, the specific emission concentrations were determined from the measured species concentrations using the average concentration over the interval considered (n cycles). For the engine working points that were performed in burst-fire mode, the species concentration was time shifted as described previously. Since the measured species concentration fluctuates even in one engine cycle, the species concentration was taken from the period that the exhaust valve is open (EVC-EVO) only. In reality, a blowdown phase occurs when the exhaust valve opens which lasts until the cylinder pressure is approximately equal to the pressure in the exhaust port. After this blowdown phase, the flow of exhaust gases is determined by the movement of the piston as the cylinder volume decreases. After the exhaust valve closes, the last gases that have been expelled from the cylinder remain in the exhaust port. Since the fast response analyzer samples using a constant flow rate, the measured species concentrations were weighted using the instantaneous exhaust flow rate, which was determined using an outflow model [44]. The species concentrations vary as the exhaust process takes place. Especially later in the exhaust stroke, unburned fuel emerges from the crevices in the cylinder and this fuel may burn or fail to do so since the temperature has decreased to a low level by that stage. Since these gases remain in the exhaust port after the exhaust valve closes, these would be oversampled when the species concentration is averaged over the entire cycle. The specific emissions concentration can be determined from the species concentration and the engine power. The engine power as calculated from the full-cycle IMEP was used for this as the actual brake power is not representative due to the high internal friction in this engine. The specific emissions concentration is therefore expressed as $[g/kWh_i]$. The air-fuel ratio can also be determined from the measured emission concentrations. In total, five different methods were used for the determination of the air-fuel ratio, or air excess ratio $\lambda$. Firstly, since both fuel and air flow rates were measured, $\lambda_{AF}$ can be determined by dividing the ratio of air and fuel mass flows by the stoichiometric mass ratio. From the species concentrations, the air excess ratio was determined using three accepted methods, the carbon balance, the oxygen balance and the Brettschneider method (which is an extension of the oxygen balance method, accounting for water in the fuel and/or air). The formulae for all methods used are rather lengthy and are therefore summarized in appendix B. The last method was the reading of the UEGO sensor that was mounted in the exhaust, just after the back-pressure valve. The air excess ratio indicated by this sensor was used as a quick aid for setting the engine working point. For air excess ratios that were not too high, the reading of this sensor was fairly accurate, but as the mixture was made leaner, the presence of unburned fuel contaminated this sensor which resulted in an erroneous reading for some time after each engine misfire. Also, this sensor is not suitable for use in a pressurized environment, as was observed in an earlier experiment.
where the sensor was mounted upstream of the back pressure valve. The differences between the three emissions-based methods were typically only a few percent. The oxygen balance and Brettschneider methods are very close. This is to be expected since the difference is only the water in the intake air which is reduced through the use of a freeze-dryer after the air-compressor (the resulting relative humidity was $\pm 10\%$ at $20^\circ C$). The carbon balance method was often found some 2-3% higher than the oxygen and Brettschneider methods. The massflow-based $\lambda_{AF}$ was also close but often about 2% below the oxygen and Brettschneider methods. Finally, the UEGO reading was close to the emissions-based air excess ratios when the mixture was not too lean but it indicated a too low value (>5%) for really lean mixtures. For the further processing of the data, the air excess ratio using the Brettschneider method was used except for the cases where the oxygen concentration was not available (i.e. the burst-fired sessions).

5.4.1 Emissions results

The main objective of the emissions measurements that were performed was the selection of engine working points for further investigation using more advanced methods. For each engine working point the air excess ratio $\lambda$ was increased until incomplete combustion or complete misfires indicated a certain limit. This limit is marked by a sharp rise in the concentration of unburned hydrocarbons, an increase in IMEP variation and a decrease in engine efficiency. Since problems like spark plug fouling and contamination of the spark plug lead with oil proved to be a returning problem, these had a small but unavoidable influence on the ignition of the mixture which caused some variation in ignition performance and as a result also in the limiting lambda values. As an example, the specific hydrocarbon emission for 8 bar IMEP and 1000 rpm is shown in figure 5.3(a). At stoichiometric combustion, the specific unburned hydrocarbon concentration is already 10 g/kWh, this value doubles for $\lambda = 1.5$ and keeps increasing as the air excess ratio $\lambda$ increases. This observation shows one of the consequences of using a ‘research’ engine for these base measurements. Since this engine has large crevices, the base hydrocarbon emission is very high compared to modern standards (about 20 g/kWh). A likely explanation is that the fuel that emerges from the crevices with decreasing cylinder pressure encounters a relatively cold environment and may fail to oxidize. The increase at $\lambda \approx 1.70$ indicates the start of incomplete combustion. This limiting air excess ratio is rather universal for these conditions and depends little on engine speed and load. The limiting lambda values at which engine operation became irregular have been selected for each working point as the leanest mixture prior to the increase of the IMEP variation. These values have been summarized in table 5.2 for methane and in table 5.3 for natural gas. The values are all in the range $\lambda = 1.65 - 1.75$ for both fuels. When looking closer at the data sets with a larger IMEP variation it shows that this is caused by the occurrence of complete misfires most of the time, however very slow burning cycles also occur from time to time. The limits as displayed in tables 5.2 and 5.3 have been used as setpoint for the combustion visualizations described in chapter 8. In figure 5.3(b), the emission of $NO$ for the same engine working point is displayed. As expected, since lambda increases, peak combustion temperatures decrease and so does the concentration of $NO$. For this engine working point, the minimum specific $NO$ concentration is approximately 1 g/kWh. Even lower values require an even leaner mixture which inevitably also leads to higher hydrocarbon emissions. This $NO$ emission is already quite low, a normal engine with less crevices
may allow a higher air excess ratio for the same $HC$ emission resulting in even lower $NO$ emissions.

### 5.4.2 Efficiency

In figure 5.4(a), the cycle efficiency for 1000 rpm and 8 bar IMEP is displayed. This cycle efficiency is based on the IMEP value calculated over the entire cycle. In this case, this is more representative than using the actual brake power due to the high friction losses. As expected, the efficiency increases as the mixture is made leaner and decreases fast when misfires occur. In figure 5.4(b), an overview is plotted for all working points with $CH_4$ fuel. Here, the lambda values that have been used for the x-axis are those based on the measured air- and fuel mass flow rates, so differences in emissions measurement strategy (section 5.4) can be ruled out. Although there is a lot of scatter due to the limited number of data points, some trends are visible. The cycle efficiency increases when the engine speed is increased. This is due to reduced heat losses (reduced time for heat transfer) and to a lesser extent a higher volumetric efficiency, as the valve timing is optimized for a medium speed. For this specific engine also the reduced blow-by losses contribute to a higher efficiency for higher engine speed. Another observation is that the efficiency increases as the engine load increases. For the shift from 4 to 8 bar IMEP at 600rpm, reduced throttling losses are responsible for the efficiency increase. A likely cause for the shifts from 8 bar to 12 bar IMEP is a relatively lower heat loss due to the higher mass and increased temperature of the walls. The efficiency for 12 bar IMEP is higher than at 8 bar, but also displays a maximum for a lower lambda value. The absolute values for the efficiency are a bit low when compared with literature. In figure 2.3, the maximum efficiencies are plotted for several engines. These efficiencies are a few percent higher on
average. This can be explained by the different operating conditions. For a typical HD engine, the maximum efficiency is found for a higher load and speed than the load and speed at which this engine was operated. The major reason for the difference however is the different construction of this engine. The blow-by fraction is rather high for this engine; this is explained further in the next section. Also the wall temperatures are lower which leads to a higher heat loss.

5.5 Analysis of cylinder pressure signal

The cylinder pressures were recorded with the intention of applying a calculation model to extract burning rate information. Therefore considerable attention must be paid to the process of recording and evaluating the signal quality. While this seems standard practice, there are several pitfalls that one can run into.

The cylinder pressure was recorded using an uncooled piezo-electric pressure transducer, type AVL GU21C. The pressure transducer was mounted in the cylinder head, between 2 valves. The radial position was just above the edge of the piston bowl. The signal was amplified using a charge amplifier, type Kistler 5011B. The output signal was recorded with a resolution of 0.1° CA by the Fevis indicating system.

When studying the cylinder pressure signal, a number of difficulties was encountered which required a more detailed study. As an engine with an elongated piston construction is different from a normal engine regarding constructional details and temperature levels, differences in compression behavior could be expected. However, from the calculation of the compression polytropic index (which indicates the deviation from a fully adiabatic process) it was found that this polytropic index showed a strange behavior. In this section, an analysis of these phenomena will be presented. The analysis is divided in the following parts: pressure signal phasing errors, pegging of the absolute pressure level, compression ratio effects, blow-by losses from the cylinder and the presence of crevices. First however, some information on the polytropic index is given.
Analysis of cylinder pressure signal

\[ dQ = \frac{n}{C_0} \frac{c}{C_0} \frac{1}{p} dV (3) \]

Eqs. (1) and (3) are equivalent, and both of them only depend on the instantaneous values of pressure and volume, these variables being known or directly measured during a motored engine test [4].

If the cylinder walls are initially (at the IVC) at the same temperature as the intake air, the adiabatic and polytropic coefficients are equal at the beginning of the cycle, and, when both coefficients start to separate (Fig. 3), heat begins to be transferred from the internal gas to the surroundings, as deduced from Eq. (3). However, if the walls are initially hot as a consequence of previous cycles, then the polytropic coefficient starts the compression cycle above the adiabatic one, they both cross each other in a certain instant of the compression stroke and they cross back again in a certain point of the expansion stroke. It can be observed that the polytropic coefficient finally ends with lower values.

\[ n = \frac{-dp/p}{dV/V} \]  (5.1)

The polytropic index \( n \) would equal the ratio of specific heats \( \gamma \) if the compression and expansion processes were adiabatic. However, in reality heat transfer occurs between the gases and the walls of the combustion chamber. In figure 5.5(a) [22], the general behavior of the polytropic index for a 'normal' engine is displayed. The ratio of specific heats \( \gamma \) is shown as the thinner line. This ratio of specific heats is not constant but shows a minimum around TDC since the specific heats change with temperature. At the start of the compression stroke the fresh gases have a lower temperature than the walls and there is heat transfer from the warm walls to the gas. The polytropic index is therefore higher than the adiabatic one. As the temperature of the gas increases due to compression, at some point (labeled \( \dot{Q} = 0 \)), the direction of heat flow reverses and the polytropic index will be lower than the adiabatic one. Close to TDC, calculation of \( n \) from a measured pressure signal becomes increasingly difficult due to the small volume changes and noise on the pressure signal. In the expansion stroke a similar process occurs. At first there is flow of heat from the gas to the walls and the polytropic index is higher than the adiabatic one. Later in the expansion stroke the direction of heat flow reverses and the polytropic index drops below the adiabatic one.

In figure 5.5(b) an early polytropic index plot for the single cylinder engine is shown. In this plot, a number of differences can be observed with respect to figure 5.5(a). Firstly, it shows that the overall polytropic exponent is lower than the adiabatic one. A crossing of the polytropic and adiabatic lines does not occur. Secondly, in the part of the compression process preceding TDC the polytropic curve drops sharply. The theoretical polytropic curve (figure 5.5(a)), also shows a drop, only closer to TDC. In the expansion part, a deep drop after TDC is also observed. These differences originate partly from deficiencies in the measurement and/or conditioning of the pressure signal and partly from engine parameters that have not been determined or estimated accurately enough.

5.5.1 Polytropic index

The polytropic index \( n \) shows the relation between changes in volume and pressure and is defined as

\[ n = \frac{-dp/p}{dV/V} \]  (5.1)

The polytropic index \( n \) would equal the ratio of specific heats \( \gamma \) if the compression and expansion processes were adiabatic. However, in reality heat transfer occurs between the gases and the walls of the combustion chamber. In figure 5.5(a) [22], the general behavior of the polytropic index for a 'normal' engine is displayed. The ratio of specific heats \( \gamma \) is shown as the thinner line. This ratio of specific heats is not constant but shows a minimum around TDC since the specific heats change with temperature. At the start of the compression stroke the fresh gases have a lower temperature than the walls and there is heat transfer from the warm walls to the gas. The polytropic index is therefore higher than the adiabatic one. As the temperature of the gas increases due to compression, at some point (labeled \( \dot{Q} = 0 \)), the direction of heat flow reverses and the polytropic index will be lower than the adiabatic one. Close to TDC, calculation of \( n \) from a measured pressure signal becomes increasingly difficult due to the small volume changes and noise on the pressure signal. In the expansion stroke a similar process occurs. At first there is flow of heat from the gas to the walls and the polytropic index is higher than the adiabatic one. Later in the expansion stroke the direction of heat flow reverses and the polytropic index drops below the adiabatic one.

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5.5.2 Pressure signal phasing errors

In a mechanical piston compression machine like an internal combustion engine, the maximum cylinder pressure occurs almost simultaneously with the minimum volume. In practical engines the maximum pressure occurs just before the crank angle with the minimum volume. This difference is generally known as the thermodynamic loss angle and has a magnitude of $\mathcal{O}(0.5\,\text{degrees})$. The exact value depends both on the ability of the piston rings to seal the combustion chamber and on the heat transfer from the hot gases to the walls of the combustion chamber (and thus on the temperature of the chamber walls). In this specific engine, the exact location of TDC was first determined statically with a cold engine by accurately measuring the piston height at two crank positions on both sides at equal distance from TDC. This distance should not be taken too small as piston motion is virtually zero around TDC. At a later stage, the location of TDC was determined dynamically using a capacitive sensor that indicates the distance between the piston surface and the sensor. This resulted in an almost negligible correction. As the piston position is determined by the crank mechanism, the location of TDC could be determined using the crankshaft encoder to an accuracy of $0.1^\circ\text{CA}$. Correcting the location of TDC relative to the pressure curve eliminated the drop during expansion to a large degree.

5.5.3 Pegging of the absolute pressure level

The pressure transducers used for measuring cylinder pressures can only detect pressure changes. These sensors work by shifting an electric charge as function of the force acting on an piezo crystal. This charge is then converted to a voltage that can be recorded. Despite the use of special high resistance cabling, some of the charge will leak away in time, therefore these sensors cannot measure constant pressures. This means that although the pressure change during the engine cycle will be recorded, the absolute pressure level is not exactly known. The procedure used for correcting the absolute pressure level is known as pegging. Several procedures for pegging the pressure signal have been in common use for engine research, but each of them has its advantages and disadvantages. One method to correct the absolute pressure level is to shift the measured pressure trace so that the cylinder pressure equals a known pressure at some point in the engine cycle. This point is usually taken around or just after BDC. When there is no volume change and therefore a negligible flow across the inlet valves, the cylinder pressure should equal the inlet manifold pressure. A problem with this method is that the crank angle at which these pressures are equal depends on the engine speed and inlet flow dynamics and therefore a wrong choice results in an incorrect pressure correction. Another way of correcting the absolute pressure level is to use the pressure-volume relations and adjust the absolute pressure level so that the polytropic index matches some expected value during a part of the compression process. The problem here is that this requires knowledge on heat and mass losses during the compression process. As this is a far from standard engine, these losses are larger than usual and the choice of a suitable polytropic index is non-trivial. Eventually, for the pegging of the cylinder pressure signal a different method was used. This method uses the measured inlet mass flow as reference. During the intake process, a pressure difference between the inlet manifold and cylinder drives the flow of fresh mixture into the cylinder. The cylinder pressure was iteratively adjusted such that the integral of the inlet flow equals the measured fresh inlet mass as found by dividing the mass flows by the
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engine speed. The method has also been described in [74].

\[(\dot{m}_{\text{air}} + \dot{m}_{\text{fuel}}) \frac{120}{n} = \int_{\theta=IVO}^{\theta=IVC} \dot{m}_{\text{in}} d\theta\] (5.2)

This method, as well as the direct referencing to intake manifold pressure, assumes that the measured intake manifold pressure is the pressure level just upstream of the intake valves. Measuring the pressure at this location is limited by constructional difficulties and the pressure is normally measured at a location further upstream. This can potentially introduce errors in the method. One important issue is the presence of pressure waves in the intake manifold. These pressure waves can cause substantial differences in the actual intake mass, depending on the dimensions of the intake system and the engine speed. Since the intake system is relatively large for this engine as the dimensions are for 6 cylinders, it was assumed that pressure waves are weak and do not cause large errors here. This assumption was supported by the crank angle resolved measurement of the intake manifold pressure; this showed only weak pressure variations. A further difficulty is the time shift of the pressure signal that is caused by the distance between sensor location and the intake valves. This time shift was accounted for using the speed of sound at the conditions and composition in the intake manifold. The correction that resulted from this never exceeded a few degrees crank angle.

Since this is a throttled engine, one has to allow for backflow from the cylinder to the inlet manifold at low manifold pressures. It was assumed that all residual gases flowing from the cylinder to the intake manifold end up in the cylinder again after the intake process. As the intake runners are fairly long, a plug flow was assumed, i.e. the residual gases flowing back from the cylinder into the intake runners do not mix with the fresh mixture and are first to enter the cylinder again when the flow direction reverses. During testing, the pressure difference between intake and exhaust was kept minimal for working points with an intake manifold pressure above 1 bar. A strong negative pressure difference however exists for low load. Therefore no situations exist where the total pressure difference between intake and exhaust becomes positive. It was thus assumed that all fresh mass flowing into the cylinder is trapped, i.e. there is no loss of fresh mass from intake to exhaust during valve overlap. A flow of exhaust gases from exhaust to intake at low manifold pressure though is possible. This was accounted for in the pegging procedure; it’s effects though are not large since the valve lift during overlap is small.

5.5.4 Errors in the compression ratio

When the compression volume is taken too small or too large, this will cause an error in the calculation of the polytropic coefficient which has an increasing effect close to TDC [22]. A compression ratio that is assumed too high causes a polytropic curve that is too low and vice versa. The piston bowl was designed for a compression ratio of 12.5. In this design, the clearance distance was set at 1 mm. However, due to various safety modifications that have been implemented during the repair of the engine damage (see section 4.1.2), the clearance distance measured 2.3 mm. A further difficulty is that the clearance distance can, without the use of sophisticated equipment, only be determined when the engine is not turning. Due to free play and elasticity of the components in the construction, the actual clearance distance may be slightly different from the one measured when the engine
is not turning. Also, the temperature of the components can cause significant errors in the clearance distance. This is caused by the unusual construction of this engine, where a large temperature difference can exist between the assembly of conrod and piston and the side plates that connect the crank case with the cylinder head. An estimation assuming the moving components are at oil temperature showed that this can account for 0.4 mm in clearance distance, compared to a cold engine. This results in an increase of 0.28 in the compression ratio. When assuming the largest clearance distance possible, at low engine speed and a cold engine, the compression ratio should be at least 11.25. This value for the compression ratio was used for all further calculations with this piston bowl configuration.

5.5.5 Blow-By losses

One other cause of a too low pressure during compression is the loss of mass from the engine cylinder. This phenomenon is also known as blow-by. Gas under high pressure can escape the cylinder through an opening in the piston ring-pack. In a normal engine this blow-by is too small to be of importance. With an extended piston construction as applied here however, conditions for the sealing of the engine cylinder by the piston rings are slightly different. There is no lubricating oil film which helps the sealing function of the rings and also the force with which the rings are pressed against the cylinder wall is much lower to limit wear of the rings. When there is a significant amount of blow-by, the pressure during compression is lower and the polytropic exponent is lower than the adiabatic. During expansion, there still is a loss of mass due to the high pressure. This causes the polytropic exponent to be higher than the adiabatic one. In figure 5.5(b), something strange seems to be the case. There seems to be an unusual decrease in the polytropic index during the compression phase, around 330° CA. Since it was suspected that the remaining strange behavior of the polytropic index was caused by blow-by effects, an estimation of the mass loss was made by applying a simple first-law heat release calculation to several (hot) motored cycles. In this calculation, the heat release due to combustion was set to zero, and the mass loss was used as closing term. This showed a mass loss characteristic that was not symmetrical to TDC, as is normally assumed for blow-by. It was decided that, since the kinematics of this engine with an elongated piston, together with an unusual piston ring construction might cause a short moment of increased blow-by during the period the piston switches from the anti-thrust to the thrust side of the liner. To model the mass loss, a single gap was assumed, through which the gases can escape. This gap was then made dependent on the piston position and has a maximum just before TDC to resemble the (assumed) kinematics of the piston. This way of blow-by modeling was then also used in the heat release model. This method however, remained a questionable point as the blow-by can be calibrated using motored cycles only. Since the contact pressure between piston rings and liner wall is also dependent on the gas pressures, the blow-by model might not capture the blow-by effects fully.

5.5.6 Crevice effects

In the emissions results, it already became clear that this engine has a high base hydrocarbon output level. A major cause for this is the presence of relatively large crevices in the combustion chamber, especially between piston and liner. Since these crevices have a large surface to volume ratio, the mass that is compressed into these crevices is at a
lower temperature than the mass in the combustion chamber leading to an apparent increased heat loss during compression. During expansion, some of the mass in the crevices flows back into the combustion chamber again, increasing the mass in there. The effect this has on the polytropic index is not clear as the mean temperature of the charge may change a bit also. The crevice gases have a temperature that is lower than the average gas temperature early in the expansion, and a higher temperature later in the expansion process. These crevice effects were also taken into account for the heat release modeling.

5.6 Heat release modeling

From the measured in-cylinder pressures, the burning rate can be extracted. This was done using a two-zones heat release model. In this model, the effects described in the previous section were accounted for. The model was therefore extended with crevice- and blow-by corrections. The details of the model can be found in appendix C. For each engine working point and air excess ratio, 50-100 cycles were recorded. For air excess ratios close to the limit, complete misfire cycles sometimes occurred. These were removed from the data set by checking the location of the maximum pressure. For each complete misfire, also the successive cycle was removed since the charge composition for this cycle contains only a fraction of the normal amount of residual gases. The remaining cycles were averaged and this average pressure curve served as input for the model.

5.6.1 Signal conditioning

The pressure signals were pegged as described in section 5.5.3. After this step, the pressure curves were filtered to reduce noise. A Savitsky-Golay type filter was chosen as this introduces no phase shift. The degree of filtering was kept to a minimum, only for high speed working points parts of the signal were filtered more to reduce the effects of valve bounce.

5.6.2 Estimation of the residual gas fraction

The estimation of the fraction of residual gases is one of the most difficult subjects in heat release modeling. It is almost impossible to describe this using simple correlations. To describe this accurately, one has to consider the full gas dynamics during the gas exchange process. The following description assumes that the exhaust pressure is equal to or higher than the intake pressure as is the case for this engine. When the exhaust valve opens, a blowdown occurs and the cylinder pressure is quickly reduced to the pressure in the exhaust system. The following upward piston stroke drives out the most of the remaining combustion products. Just before TDC, the intake valves open and the cylinder pressure is reduced to the inlet manifold pressure. Depending on the engine load this manifold pressure can be much lower than the cylinder pressure. During this pressure drop, a certain amount of residual gases flows into the intake manifold to return to the cylinder at a later time, but also exhaust gas can flow back from the exhaust manifold to the cylinder for the period the exhaust valve is still open. For the determination of the residual gas mass fraction these phenomena were taken into account. At intake valve opening, the mass of residual gas residing in the cylinder was determined using the ideal gas law assuming the temperature of the gases remaining in the cylinder equals the exhaust temperature. After
that, the flow through the exhaust valves during valve overlap was calculated to determine
the change in residual gas mass until the exhaust valve closes fully. The residual gas mass
can thus be written as
\[
\frac{p V}{R_0 T_{ex}} + \int_{IVO}^{EVC} \dot{m}_{exvalve} d\theta
\] (5.3)

As driving pressure difference for this flow, the difference between the cylinder pressure \(p(\theta)\) and the mean exhaust pressure \(\overline{p}_{ex}\) was used. Ideally, a crank angle resolved exhaust pressure is used for this. As this was not available, the exhaust pressure during valve overlap was assumed constant at the level just before intake valve opening. The effect of pressure waves is not captured this way and may cause some degree of error.

The fresh charge mass is determined from the air and fuel flow rates as \(m_{fresh} = (\dot{m}_{air} + \dot{m}_{fuel}) \cdot 120/n\). The resulting residual mass fraction then follows from
\[
X_{res} = \frac{m_{res}}{m_{fresh} + m_{res}}
\] (5.4)

The temperature at IVC, which is also the initial temperature for the heat release calculation is determined using the ideal gas law as the total mass, volume, pressure and composition are known at intake valve closure.

### 5.6.3 Results

The model was run for every input pressure curve, for methane and DNG. In figure 5.6(a) and 5.6(b), some typical results are shown. In figure 5.6(a), the spark angle is indicated by a circle (o), the start of heat release by an asterisk (*). For the low-load engine working points, a reference measurement with a stoichiometric mixture was also performed. It can be seen that the ignition delay (distance between the spark angle and the start of heat release) for stoichiometric mixtures is much shorter, and also the pressure rises faster. The curves for the mass fraction burned \(x_b\) for stoichiometric combustion (figure 5.6(b)) show a steep increase up to \(x_b \approx 0.75\), after which the curve flattens for some 10-20 degrees CA. After this, the burning rate seems to increase a bit again throughout the burn-out phase. This behavior is partly caused by the shape of the combustion chamber and partly by the chemistry assumed. The mixture in the piston bowl can burn very fast since the level of turbulence is high and there is relatively little contact surface. Once the flame reaches the edge of the bowl, propagation into the narrow slit between piston and cylinder head slows the flame down. For leaner mixtures, the combustion is phased later and the piston has traveled further down already, leaving a much wider slit for the flame to propagate into.

At stoichiometric conditions, the maximum temperature calculated for the burned zone reaches more than 2500 K. At this high temperature dissociation of \(CO_2\) occurs which leads to a change in average molar mass and specific heat ratio. If complete combustion is assumed (i.e. reaction products carbon dioxide and water only), this flattening of the curve is less pronounced. The maximum mass fraction burned reaches only 90-95% in most cases. This seems to be deficient, however, if the specific hydrocarbon emissions are considered it turns out that the mass fraction burned cannot be higher and is calculated a bit too high already. The combustion efficiency in this case is determined from
\[
\eta_c = \frac{ISFC - ISHC}{ISFC}
\] (5.5)
(a) Cylinder pressure

(b) Mass fraction burned

Figure 5.6: Cylinder pressure and mass fraction burned, 1000 rpm, 8 bar Imep

(a) Temperature unburned zone

(b) Temperature burned zone

Figure 5.7: Temperatures unburned and burned zones, 1000 rpm, 8 bar Imep
in which ISFC is the specific fuel consumption in \( [g/kWh] \) and ISHC the specific hydrocarbon emission, also in \( [g/kWh] \). When comparing stoichiometric and lean combustion, it shows that the combustion efficiency for stoichiometric combustion is a bit higher. This is probably caused by the higher temperatures during expansion, therefore the unburned fuel from the crevices can oxidize easier. The lower hydrocarbon emissions at stoichiometric combustion also indicate this. Another cause could be that the Woschni relation used for the heat transfer coefficient is less suited for stoichiometric combustion in a SI engine and predicts a transfer coefficient that is too high. The difference between the maximum mass fraction burned and the combustion efficiency as determined from the hydrocarbon emission is larger for some working points than others. There are several factors that contribute to the difference. Firstly, for low load operating points, the fresh mass is measured less accurate due to the accuracy of the instruments used. This has an effect both on the fresh mass as well as the pegging of the pressure signal as the fresh mass is used as a reference. Further, for low intake manifold pressures as is the case for the stoichiometric mixtures, the assumptions used in calculating the flows into and out of the cylinder (see section 5.5.3) may not be entirely accurate and the residual mass flows further back into the intake manifold and mixes with the fresh gases. Another point of caution is the calculation of the back flow of exhaust gases from the exhaust to the cylinder during valve overlap, the driving pressure difference may not be correct as a crank angle resolved exhaust pressure was not available. Errors in the amount of residual gas, as well as errors in the fresh mass propagate into the residual gas fraction as well as the initial temperature.

In figure 5.7(a) and 5.7(b), the temperatures for the unburned and burned zone are shown for the same engine working point as figure 5.6. The temperatures for the unburned and burned zones reach the highest values for stoichiometric combustion, due to the higher flame temperatures and also the higher rate of pressure rise. When considering the lean mixtures only, the temperature of the burned zone shows a clear trend towards lower temperatures as \( \lambda \) increases. At \( \lambda = 1.47 \), the maximum temperature reaches some 2270 K, whereas for the leanest mixture (\( \lambda = 1.76 \)) this has dropped to 2020 K. The decrease in maximum temperature is reflected in the emission of NO, this decreases from 7.6 \( g/kWh \) to 0.38 \( g/kWh \). This decrease is to be expected, since the formation of nitrogen oxides is dominated by the thermal mechanism in this application. In figures 5.3(a) and 5.3(b), the emission of hydrocarbons and nitrogen oxides was already plotted against the air excess ratio. The hydrocarbons show a steady increase as \( \lambda \) increases, and a sudden rise for the highest \( \lambda \) used. This steady rise in hydrocarbon emission is caused both by the decreasing completeness of combustion and also by the decreasing temperatures during expansion which prevent oxidation of the fuel emerging from the crevices. The emission of NO shows a decrease with increasing \( \lambda \). The NO concentration seems to decrease very little when \( \lambda \) is increased from stoichiometric to about 1.5. The temperature of the burned zone has decreased from >2600 K to 2270 K over this range. As the formation of NO is determined both by high temperatures and the availability of oxygen, the maximum NO emission is normally found for \( \lambda \pm 1.15 \). This air excess ratio was however not included in the test series. When \( \lambda \) is increased further the NO concentration drops quickly to reach a value of 0.38 \( g/kWh \) at \( \lambda = 1.76 \). This behavior can be explained as the formation of NO through the thermal mechanism is exponentially temperature dependent. Also, the temperature as calculated for the burned zone \( T_b \) is the mass-averaged temperature. As the formation of nitrogen oxides takes place
in or just behind the flame, the actual flame temperature, which is even higher, should
be considered here.

In the model, a turbulent burning velocity was calculated using

$$S_T = \frac{\dot{m}_b}{\rho_u A_f}$$

(5.6)

The flame surface $A_f$ in this relation was calculated assuming a spherical shape for the
flame and a flat piston. The contact surfaces of the flame and the cylinder head, piston
or walls were subtracted; the flame is assumed to propagate using the free surfaces only.
The resulting turbulent burning velocity is shown in figures 5.8(a), 5.8(b) and 5.8(c) for
an engine load of 4 bar IMEP, engine speeds 600, 1000 and 1350 rpm and methane as a
fuel. In figures 5.9(a) to 5.9(c) the same series is shown for DNG for engine speeds of 600,
1000 and 1400 rpm.

The theory for turbulent combustion was earlier expressed using equation 3.36, re-
peated here for clarity.

$$S_T S_L = 1 + C \left( \frac{u'}{S_L} \right)^n$$

(5.7)

This equation states that the magnitude of the turbulent burning velocity with respect
to the laminar one is a function of the ratio of the turbulent velocity fluctuations and
laminar burning velocity. The velocity fluctuations are generally assumed to scale with
engine speed; therefore one would expect a scaling of the turbulent burning velocity with
engine speed (neglecting the factor 1 since $\frac{u'}{S_L} \gg 1$). When the cases for stoichiometric
combustion for methane at engine speeds of 600 and 1000 rpm are compared, it is found
that when the engine speed increases with a factor 1.67, the turbulent burning velocity
increases with a factor 1.5. For DNG, there are 3 stoichiometric cases available. Here the
same observation can be made and the differences are even more pronounced. When the
engine speed increases with a factor 1.67 from 600 to 1000 rpm, the turbulent burning
velocity increases with a factor 1.34 only. When the cases for 600 and 1400 rpm compared,
the engine speed increases with a factor 2.33, while the increase in $S_T$ is only 1.67. The
laminar burning velocity $S_L$ is a function of pressure and temperature and is found to
increase for increasing engine speed (at the same engine load). This increase in laminar
burning velocity does not explain the less than linear increase of $S_T$ with engine speed.
With equation 5.7 in mind, this should result in a more than linear increase. A possible
explanation could be that the magnitude of the turbulent velocity fluctuations is not
linear with engine speed. Also, the flame surface assumed could be incorrect. The flame
propagation was assumed to proceed spherically, starting from the spark plug. The contact
surfaces with the walls of the combustion chamber were subtracted in the calculation of the
total flame surface. This spherical flame shape may be less valid for higher engine speed.
The main conclusion here is that when equation 5.7 is used to describe the development
of the turbulent burning velocity with engine speed (assuming $u'$ to be proportional with
engine speed), then the factor $n$ should be smaller than unity.

In the previous section it was argued that the ratio of turbulent and laminar burning
velocity is a function of the magnitude of the velocity fluctuations and varies thus with
engine speed. When the factor 1 in equation 5.7 is neglected, then the increase of turbulent
burning velocity with engine speed should be the same for stoichiometric and lean mixtures
and the ratio between the two should be the same. For the highest engine speed with
Figure 5.8: $S_T$, $CH_4$, 4 bar IMEP

Figure 5.9: $S_T$, DNG, 4 bar IMEP
methane as a fuel, no measurements with stoichiometric mixtures were available. At 600 rpm, the turbulent burning velocity for stoichiometric mixtures is approximately 35-50% higher than for lean mixtures. For 1000 rpm the maximum burning velocity is approximately 22-42% higher for a stoichiometric mixture. For DNG as a fuel a similar observation can be made. For an engine speed of 600 rpm, the maximum turbulent burning velocity for a stoichiometric mixture is 23-57% higher than for lean mixtures. For 1000 and 1400 rpm, this difference is 17-33% and 16-43% respectively. For both fuels, it is seen that the ratio between stoichiometric and lean mixtures decreases when the engine speed is increased from 600 to 1000 rpm. For DNG at 1400 rpm, there is a further (small) decrease. A likely explanation is that, for lean mixtures, the assumed spherical shape is less valid for high engine speeds. The turbulent burning velocity is thus calculated too high, reducing the ratio between stoichiometric and lean turbulent burning velocity.

When considering only the lean mixtures, the maximum turbulent burning velocity is seen to increase with the engine speed, as $S_T$ increases for methane from 4-4.5 m/s to 8-9 m/s for an increase in engine speed from 600 to 1350 rpm. This increase is less than proportional with engine speed, however the difference is not large. The leaner mixtures have initially a somewhat lower turbulent burning velocity; when combustion progresses however their turbulent burning velocity is higher. This could indicate that the initial development becomes more difficult due to stretch effects etc. However it could also be that the turbulence level is already less due to a later combustion phasing. For DNG the turbulent burning velocity for lean mixtures ranges from about 5 m/s at 600 rpm, through 7 m/s at 1000 rpm to about 8.5 m/s at 1400 rpm. This increase is even less proportional to the increase in engine speed than for methane. This could indicate that the coefficient $n$ in equation 5.7 is not the same for these fuels.

Another observation is that for the leanest mixtures with $\lambda \geq 1.7$, the entire combustion event is phased later. This is most pronounced for DNG. The time needed to initiate combustion (flame development angle $\Delta \theta_d$, see section 3.4) increases much. The remainder of the combustion event is also phased later and has a much lower turbulent burning velocity.

5.7 Discussion

In this chapter, the fired engine experiments have been described. The primary objective of these experiments was the selection of the limiting air excess ratios for every engine working point. These combinations of engine working point and air excess ratio serve as setpoints for the investigations in the transparent engine configuration in chapter 8. It was found that the limiting air excess ratio can be found between $\lambda=1.65-1.75$, and displays little variation for changes in engine load and speed over the range tested. Also, the limiting air excess ratio is virtually the same for methane and natural gas. When the data sets for the limiting air excess ratios are investigated in more detail, it is found that the rise in hydrocarbon emissions and IMEP variation as well as the decrease in engine efficiency is often caused by complete misfires. Sometimes, very slow burning cycles also occurred. This indicates that the limit encountered for this specific engine may be stretched a bit by using a different ignition system with more spark energy or a different concept (flame jet, pre-chamber or laser ignition). When these air excess ratios are used for the engine operation with the optically accessible version, continuous firing
or even a firing period of $O(100)$ cycles will not be possible. If single cycle firing is used, the fraction of residual gases will be less and the mixture will ignite easier.
Chapter 6

Measurement of velocity field and turbulence in the engine cylinder

In this chapter, the measurement of the velocity field and the turbulence characteristics inside the combustion chamber of the engine will be presented. Following this introductory paragraph, the basics of the measurement method applied, Particle Image Velocimetry, will be explained together with a review of the specific problems with the application of this technique in the engine. Further, the set of engine working points for which the flow will be measured will be presented. After that, focus will be on the measured velocity fields and more specific the post-processing techniques that were used to extract useful data from these. Techniques will be presented for the extraction of various quantities, like turbulence intensity and length scales. Also a short analysis of the preferred flow structures, derived using the POD technique will be presented. The chapter will end with a summary of the relevant results that can be used in the analysis of turbulent combustion at the end of this thesis.

6.1 Particle Image Velocimetry technique

6.1.1 Basics

Several techniques are available for the measurement of the velocity of fluid flows. Some of these techniques, like hot-wire-anemometry, insert a measurement device into the flow to be measured. Other techniques are non-intrusive which means the flow itself is not disturbed by the measurement. These techniques have in common that the flow itself is not measured, but instead the velocity of some tracer particle that is introduced into the flow is determined. A further distinction can be made regarding the resolving power of the measurement technique. Here, a distinction can be made between techniques that provide a time resolved measurement of the velocity in a single point (which can be a single velocity component or multiple components simultaneously), and techniques that provide a spatially resolved measurement of the velocity in a certain area at a distinct moment in time. PIV (Particle Image Velocimetry) is an example of the latter category. With PIV, the flow to be measured is seeded with small tracer particles. A laser sheet is introduced into the flow domain. The seeding particles are illuminated by the laser sheet. The laser used to generate this sheet is normally a pulsed laser, which generates two separate pulses within a short interval. A camera is positioned at an angle perpendicular to the laser sheet.
This camera records the light that is scattered from the seeding particles and generates a set of two images (‘double frame’) of the illuminated particles. The PIV technique extracts velocity information by first subdividing the measurement domain into smaller sections called ‘interrogation areas’. In these interrogation areas the cross-correlation between the two frames is calculated, thereby determining the mean displacement of the particle images from one frame to the next. This mean particle displacement, together with the image scale and the time separation between the images results in a velocity vector. Although this vector represents the mean velocity of the particles in the interrogation area (which is actually an interrogation volume due to the finite thickness of the laser sheet), its location is attributed to the center of the interrogation area. Through evaluation of the whole domain, a regular grid of velocity vectors is determined this way. The whole process is illustrated in figure 6.1. The specific PIV system hardware and software that was used in these measurements will be explained in some more detail in the next sections.

### 6.1.2 PIV hardware

The PIV system available in the ICE group is a fairly simple but still effective system from Dantec. It is built around a PIV2100 processor system unit. This unit controls the timing of the lasers, and the image acquisition from the camera. In its core it has a dedicated hardware processor that processes the image frames using a (probably single-pass) cross-correlation algorithm. A controlling PC connected to it is used to store the resulting velocity fields (optionally already validated by some crude validation algorithms) and also the raw dual-frame images. The unit features a trigger input which makes it possible
to synchronize the measurement of velocity fields with the crankshaft of the engine. This synchronization was provided by a custom designed trigger system that could also interact with the engine control unit for PIV measurements in a combusting environment. The trigger system synchronized itself to the engine signals (crankshaft encoder and camshaft signal) and provided trigger pulses at the same crank angle in the engine cycle. It also controlled the engine firing mode for skip- or burst-fire operation. The laser used was a Continuum Surelite dual cavity ND-YAG laser. The two cavities in the laser enclosure both provide a single pulse of laser light. The beams from the cavities are aligned using (dielectric) mirrors in such a way that these overlap when they exit the laser enclosure. This laser provided, after careful alignment of its internal mirrors and frequency doubling crystal, 135 mJ/pulse at 532 nm. The maximum power from both cavities was not exactly equal but differed by about 15%. For PIV the illumination of both frames should be as equal as possible; therefore this intensity difference was minimized by adjusting the Q-switch of the most powerful cavity to a slightly longer activation delay, thereby reducing its pulse energy. Unfortunately, due to the somewhat outdated design of this laser it was not possible to trigger it at random. Due to thermal stability requirements, the nominal pulse frequency of the laser was 10Hz. It could be used in a window triggered mode, which means that external trigger pulses were only accepted when these fell within a small time-window around the 10Hz trigger period. In practice, this meant that only 20% of the trigger pulses sent to the PIV system resulted in a velocity field. To make things worse, engine speeds that were in exact resonance with the laser frequency (i.e. 600rpm) occasionally resulted in very low data rates since all triggers sent to the PIV system fell outside the acceptance window.

6.1.3 PIV software

As explained in the previous section, the PIV hardware processor performs the cross-correlation of the frames in an image. This algorithm however was not accessible and in practice it was only possible to use a rather large, 64 pixel, interrogation size with velocity fields of reasonable quality. It was therefore decided to use the output of the hardware processor only as an on-screen quality indicator when performing the measurements, while the final interrogation of the double frame images was performed afterwards. There are a number of commercial software programs available that will perform this task. For the processing of these images however it was chosen to use the open-source algorithm URAPIV, developed in Matlab® by Shavit, Liberzon and Gurka [83], that is available through the internet. This was a convenient choice, since the validation and interpolation routines were also implemented in Matlab® and all could be packed in a single interface. As a check, the output of this open-source algorithm was compared to the output of the hardware processor for 64 pixel interrogation size and was found to be nearly identical. For 32 pixel interrogation size, the output of the open-source algorithm was found more usable as it delivered velocity fields with less spurious vectors.

6.1.4 Introduction of the light sheet into the engine

The beam emerging from the laser unit was guided to the engine using several mirrors. The laser was positioned outside the engine test cell at some distance (about 7 meters) from
the engine and was mounted on a stable optical support table to make it less susceptible to the vibrations the engine induces. At certain engine speeds a resonance with the floor of the building occurs. The laser beam was transformed into a sheet using a negative cylindrical lens and a spherical positive lens which were positioned close to the engine. The laser sheet thus formed has a usable width of about 120mm. The sheet is focused in the direction perpendicular to the sheet and has a minimal thickness of less than 1mm. The lenses are positioned such that the focus is in the middle of the engine cylinder. The sheet thickness at the edge of the cylinder was less than 1.5mm. This is illustrated in figure 6.2.

6.1.5 Mirror set-up and field of view

As was already shown in section 4.3.1, a mirror is positioned inside the piston at a 45° angle. The PIV camera images the measurement plane at the top of the cylinder via this mirror. The field of view that results from this set-up has a diameter of 80mm minimum. The actual field of view depends on the momentary piston position. When the piston is at TDC, the edge of the bowl limits the view to 80mm, when the piston is further down the cylinder, the visible part of the measurement area increases (figure 6.3). For the measurements described in this thesis, the piston was always within TDC ± 70°. As this results in only a small increase in the visible part, the field of view was taken 80mm diameter for all images for ease of computation.

6.1.6 Generation of seeding particles and introduction into the engine

Various substances can be used as seeding for flow measurements. The preferred seeding material depends on the application. The most important property of the seeding particles is their ability to follow the flow that is to be measured. Other considerations include the scattering cross-section and the mechanical properties of the material. The flow-tracing ability of a particle is determined by a balance between the drag force $F_d$ arising from the relative velocity between particle and fluid $\vec{u} = u_p - u_f$ and the inertia forces that work...
on the particle. When the Reynolds number based on the particle dimension \( Re_p = \frac{\rho_f u_p}{\mu} \)

is \( O(1) \) or less, the drag force on the particle \( F_d \) can be approximated using the Stokes drag coefficient \( C_{D,stokes} = 24/Re_p \). Melling [56] expresses the particle acceleration as

\[
\frac{du_p}{dt} = -C(u_p - u_f),
\]

in which \( C \) is a characteristic frequency which he defines in terms of a drag coefficient as

\[
C = \frac{3}{4}C_D Re_p \frac{\mu}{\rho_p d_p^2}.
\]

For a stokes drag coefficient \( C_{D,stokes}, C \) becomes \( \frac{18\mu}{\rho_p d_p^2} \). For the limit of \( \frac{\rho_p}{\rho_f} \gg 1 \) which is often the case in gaseous flows, he expresses the flow-tracing ability of a particle as the ratio

\[
\frac{u_p^2}{u_f^2} = \left(1 + \frac{\omega_c}{C}\right)^{-1},
\]

in which \( \omega_c \) is the maximum turbulence frequency of interest. For this case, \( \omega_c \) can be estimated a priori using the expected velocity and the grid spacing. The expected velocity is of the order of the mean piston speed. The mean piston speed is about 7.3m/s (at an engine speed of 1400rpm), the grid spacing is 1.6mm (using 32 pixel interrogation areas and 50% overlap). The maximum turbulence frequency of interest then becomes \( \omega_c = 4562 \) rad/s, or 726 Hz.

As seeding material in this application, one could use either solid particles (e.g. \( TiO_2 \)), or oil droplets. More exclusive seeding materials (i.e. micro balloons) are also possible, but these were considered too expensive since the particles cannot be re-used as the engine is an open system. Solid particles have the advantage that these can also be used at a later stage when combustion also takes place. Also, their scattering properties are slightly better than those of droplets. Previous experience with a steady flow set-up using these solid particles has learned however, that pollution of the measurement area is a big problem, and also the abrasive nature of these particles is undesired in an engine. For these reasons, it was decided to use oil droplets for this application. The use of oil droplets has been compared to titanium dioxide in the same steady-flow set-up and was found to
perform satisfactorily, and to produce comparable velocity fields [17]. A short study on various devices for the generation of the droplets [46] has shown that the most suitable device for the generation of oil droplets in sufficient quantity would be a Laskin nozzle. A set of 2 Laskin nozzles was therefore used as seeding device for the PIV measurements in the engine. The size distribution of the generated olive oil droplets was determined using a Malvern Mastersizer ([46], [45]). Almost all droplets had a diameter \( d_p < 1\mu m \). Equation 6.1 then gives, for \( \frac{u_p}{u_f} = 0.99 \) and \( d_p = 0.5\mu m \), a maximum frequency of 1.8 kHz, which is more than sufficient for this application.

The Laskin nozzles were built into an aluminum container which can be pressurized to facilitate the seeding of the engine inlet, even at above-atmospheric manifold pressures. The output was connected to the inlet manifold using a simple plastic hose which functions as a filter for the very few larger oil droplets as these will probably stick to the walls and will not reach the engine.

### 6.1.7 Selection of the time delay between successive frames

With the PIV technique, the time delay between successive frames is determined by several factors for which a common optimum has to be selected. On one hand, the time delay should be as large as possible as this gives the widest dynamic range. On the other hand, an excessive time delay results in a loss of correlation as particles move out of the interrogation area between the laser pulses. In an optimal situation, the in-plane particle displacement should be no more than about 1/4 of the interrogation size to limit the number of particles that move out of the interrogation area in the in-plane direction ([68] and others). In flows where the third, out-of-plane, velocity component is large the displacement of particles in the perpendicular direction is limiting. For the measurements described in this thesis, this is also the case. The average velocity is \( \mathcal{O}(10) \) m/s, the thickness of the laser sheet is 1-1.5mm. If the out-of-plane displacement is limited by particle loss to 1/4 of the sheet thickness, the maximum allowable time delay is 25 \( \mu s \). It was found that loss of correlation in large parts of the field occurred with time delays >50 \( \mu s \). For smaller parts of the field this occurred for time delays of 30-40 \( \mu s \). Therefore, a time delay of 20 \( \mu s \) was used for all measurements. An inevitable drawback of this choice is that the in-plane particle displacement is also limited to approximately two pixels.

### 6.2 Selection of engine working points for flow field measurement

The engine working points (combination of engine speed and manifold pressure) were derived from the engine experiments using the full-metal version of the engine, without optical access as described in chapter 5. This means that the number of engine speeds is limited to 3. For the fired engine operation, the maximum engine speed used was 1400 rpm. This engine speed however produced a resonance in the engine under motored operation. To reduce mechanical loading and also to have a stable optical set-up with minimal vibrations the maximum engine speed was reduced to 1350 rpm. Further engine speeds were kept the same at 600 and 1000 rpm. The maximum engine load used in fired operation was 12 bars IMEP. This translates into a manifold pressure at lean conditions of 200 kpa. Since with the optical engine configuration there is no means of piston
cooling at all, it was decided to limit the thermal loading on the piston and therefore to reduce the number of manifold pressures to only 2, being 80 kpa and 140 kpa. These correspond to respectively 4 and 8 bars IMEP for fired operation. The conditions for the PIV measurements are summarized in table 6.1.

<table>
<thead>
<tr>
<th>Engine speed</th>
<th>IMEP=4 bar</th>
<th>IMEP=8 bar</th>
</tr>
</thead>
<tbody>
<tr>
<td>n=600rpm</td>
<td>IMAP=82kpa</td>
<td>EMAP=100kpa</td>
</tr>
<tr>
<td>n=1000rpm</td>
<td>IMAP=80kpa</td>
<td>EMAP=100kpa</td>
</tr>
<tr>
<td>n=1350rpm</td>
<td>IMAP=76kpa</td>
<td>EMAP=100kpa</td>
</tr>
<tr>
<td></td>
<td>IMAP=140kpa</td>
<td>EMAP=125kpa</td>
</tr>
<tr>
<td></td>
<td>IMAP=135kpa</td>
<td>EMAP=135kpa</td>
</tr>
</tbody>
</table>

Table 6.1: Engine speed and manifold pressure for PIV measurements

It was found that the positioning of the laser sheet close to the cylinder head resulted in direct reflections of laser light into the camera. These reflections are caused by the geometry of the valve pockets in the cylinder head that have been machined at an angle of 45°. The laser sheet partially reflects from the opposing surface of the transparent liner and illuminates the surface of the valve pockets. Light reflects again from these edges and leads to overexposed parts of the image. These images were largely unusable for PIV analysis as the overexposed parts resulted in large areas with only spurious vectors. This problem could only be alleviated by positioning the laser sheet somewhat further from the cylinder head surface. As a compromise, a distance of 6 millimeters was chosen. This resulted in less reflections, but it also meant that a part of the cycle around TDC could not be used as the piston was blocking the laser sheet. In practice, no measurements could be conducted between 340° and 380° CA.

Despite having good flow-tracing properties, the use of olive-oil has a disadvantage. It was found by experience that, after 380° CA, hardly any seeding particles could be detected in the images. It is likely that the oil droplets evaporate during compression as the gas temperature around TDC exceeds the boiling temperature of olive oil. As the aim of these PIV measurements is to determine the flow field under combustion conditions, the velocity fields should be measured as close to TDC as possible. The aforementioned matrix of conditions (table 6.1) was therefore measured at 340° CA.

To extend the measurement range, the olive oil seeding was later replaced with silicon oil. This has less tendency to evaporate than olive oil, which makes velocity measurements after TDC possible. With silicon oil however, the generation of droplets with moderate air pressure feeding the Laskin nozzles resulted in a small fraction of larger droplets. The number of larger droplets was low enough not to influence the flow-tracing ability, but these larger droplets stucked to the liner and piston window leading to much faster fouling. The number of velocity fields that could be recorded was therefore reduced. With olive oil, sometimes several hundred velocity fields could be obtained before cleaning was needed, whereas this number was limited to about 50 with silicon oil. With silicon oil, one sweep was made across a crank angle range from 310°-430° at 1000rpm and 80 kpa MAP to determine the development of the turbulence intensity and length scales during compression. At a later stage, when PIV was applied in a combustion environment, silicon oil was the only option, since olive oil gave no clear images under these higher pressures and temperatures. This subject however will be treated extensively in chapter 8. Due to window fouling, measurements at crank positions further from TDC could hardly be...
made as the necessary seeding density must be achieved within a larger volume, which leads to very large quantities of seeding and window fouling after only very few velocity fields.

6.3 Post processing of measured velocity fields

With the PIV technique, velocity fields are rarely without error. These errors arise partly from the method itself and are partly caused by the experimental conditions under which the measurements take place. Since the PIV method is based on the calculation of an average particle displacement through correlation of two image frames, it is essential that the particle images are present in both frames. If the time between the two frames is excessively long, particles can move out of the interrogation volume in either the in-plane or the out-of-plane direction which results in a loss of correlation. An additional problem in this case is the overexposure of certain areas of the images, together with small variations in absolute intensity and intensity distribution within the laser sheet. These variations result in many cases in a correlation peak which is falsely interpreted as a valid particle displacement. The removal of these and other false or spurious vectors is the task of the PIV post-processing algorithms which will be discussed in section 6.3.1. When these spurious vectors are removed, voids in the velocity fields remain. For the calculation of derived quantities like turbulence spectra, or vorticity, these voids have to be filled by interpolation. This will be explained in section 6.3.2. After that, several derived quantities will be presented.

6.3.1 Validation techniques

Process related validation

Spurious vectors in PIV data stand out because these have a different magnitude and/or direction than what would be expected based on the topology of the velocity field. The occurrence of these errors can partly be detected during the interrogation process. Since the interrogation process results in a correlation plane with one or more peaks in it, one can test these to detect abnormalities. A measure that is often used is the height of the highest peak relative to the second highest or to the average value of the correlation plane. This method is referred to as 'peak height validation' and can also be regarded as applying some kind of lower limit on the signal-to-noise ratio of the process. A correlation plane which does not have a distinctive peak is indicative of an uncertain displacement. In practice, this limit was set to 1.05-1.10.

A second measure that can be used is the width of the correlation peak. This is in practice harder to use, as very clear images of separate particles are seldom found in actual images and therefore the peaks found are always somewhat broader. This method was not used for the measurements described. A peak width that is too small will result in the so-called peak locking phenomenon, as the displacement cannot be determined to a sub-pixel level. In that case, when a histogram of the displacements is plotted, distinct peaks can be observed corresponding to integer pixel displacements. This effect was not detected in the velocity data.
Flow field related validation

For the correlation peaks that met the aforementioned criteria, a displacement is calculated. Through evaluation of the shape of the correlation peak, the displacement can be determined to sub-pixel level. These displacements, scaled with the image-to-world scaling factor result in a raw velocity field. A good and a bad example of a raw velocity field are plotted in figure 6.4. It can be seen clearly that in the good example (figure 6.4(a)), a small number of vectors can be detected that clearly do not fit in. In the bad example (figure 6.4(b)), a larger portion at the bottom of the velocity field is filled with apparently random vectors. These false vectors can be detected either by their magnitude or by their direction or both. The deviation of a vector can be defined either temporal or spatially, and in the latter case as a deviation from the mean of all vectors in the field or from a local neighborhood. Which method functions best depends on the situation. If spurious vectors occur in groups, as is often the case when overexposure of the image is the cause, the deviation from the spatial mean is the preferred method since it is likely that spurious vectors exist at that location in all velocity fields, and also that they occur in clusters so that they do not stand out in a local (spatial) neighborhood. If spurious vectors occur in small groups, the detection from the local mean can be difficult since the group mean is influenced too much by a small number of much larger vectors. In that case, the local median is preferred over the local mean since the median does not weigh the magnitudes but sorts them so that a few very different vectors do not have a large influence on the median. In literature, several studies comparing the performance of these algorithms can be found ([84, 60]). In figure 6.4 another problem can be detected already. There are a number of curved bands which contain vectors that do not differ in magnitude, but have alternating opposite directions. These are caused by the overexposure of the valve pocket areas. This is displayed in more detail in figure 6.5, together with a typical image in which the overexposed areas can be seen. The areas with opposing vectors are hard to detect as the magnitude does not differ from the spatial mean or from neighboring vectors, and also, since the flow is highly turbulent the direction does not differ to a degree that can be detected automatically without rejecting also good vectors. In this case, it was chosen to
remove these areas in all velocity fields by masking them out. The size and shape of the mask needed can be found using the temporal mean as these bands become clearly visible then, as displayed in figure 6.6. The vectors to be masked out can be selected by their relatively small length compared to the valid part of the field. This procedure introduces some subjectivity of the user into the process which is clearly undesired, yet inevitable.

The strategy that was eventually used was to calculate raw velocity fields for all images of a series. After that the series was visually inspected and a start and end were defined by visually evaluating the global number of spurious vectors per velocity field. The start of the usable part of the series was usually the first image with a sufficient seeding density. The end of the usable part of the series was determined by an increase of the number of spurious vectors due to window fouling. The program used also allowed to view the accompanying image to each velocity field.

The validation strategy that was then applied to the selected range of velocity fields was to reject all vectors with a length of more than five times the spatial mean per field and for some series a median validation with the same limit. The overexposed areas were masked out as a whole. The mask for this was constructed using the temporal mean over the selected velocity fields. After this procedure, velocity fields resulted in which sometimes few vectors were missing, and sometimes (for experimental conditions with lots of reflections or extreme window fouling) velocity fields with rather large groups of missing vectors.

### 6.3.2 Interpolation of rejected vectors

As explained in the previous sections, the validation process on the measured velocity fields results in a number of velocity fields with isolated vectors or clusters of vectors missing. While the temporal mean can still be approximated by averaging over all validated vectors (provided there are sufficient valid vectors available at each location), the determination
of several derived quantities however requires that the velocity fields are complete. This means that the locations where vectors have been rejected must be filled with some interpolation of the neighboring vectors. For this purpose, several published methods were tried. For isolated vectors one can simply use linear or polynomial interpolation, in both directions if desired. This procedure becomes less accurate when larger groups of vectors are missing. A common problem of all interpolation methods is that certain features in the velocity field are hard to reconstruct. All interpolation methods employ some degree of low-pass filtering, which smoothes out the smallest structures in the flow. This, in turn, affects the energy distribution among the various flow scales. Another factor for consideration is the computational effort for an interpolation algorithm. Since these data sets consist of tens to hundreds of velocity fields, computationally demanding algorithms like Kriging interpolation ([36]) are not very practical.

For these reasons, two interpolation algorithms were further explored, this being adaptive gaussian convolution and POD interpolation.

**Adaptive Gaussian convolution**

Interpolation by convolution with a Gaussian window is an established technique. This method works well for isolated missing vectors, but it was found that for larger gaps, a substantial filtering effect occurs. Agüí [3] adapted this technique in such a way that only valid data is used. The convolution is then formulated as follows:

\[
    u(x) = \frac{\sum \alpha_i u_i}{\sum \alpha_i}
\]

in which \(\alpha_i\) are the coefficients of the Gaussian convolution matrix and \(u_i\) are the known (validated) velocity components covered by the convolution window only. Thus, the convolution is evaluated using only valid data. The coefficients are determined by \(\alpha_i = e^{-(n_1^2 + n_2^2)/2\sigma^2}\), in which \(n_1\) and \(n_2\) are the grid-distances from the vector to be interpolated. The actual shape of the Gaussian distribution is still a parameter and can be chosen by
the standard deviation $\sigma$. This choice determines how the various velocity samples are weighted. The smaller $\sigma$, the more the closely neighboring vectors are weighted and the smaller the filtering effect is. Here, $\sigma = 2$ was used for all cases. This interpolation method was implemented in such a way that, in case of a larger void, the missing vectors are interpolated starting with the one that has the most valid neighbors and working down to vectors that have only one valid neighbor. If a void still remains, the same procedure is used, but instead of valid neighbors, previously interpolated neighbors are used. This way, the voids in the velocity field are filled from the edges to the center of the void. Strictly speaking, if the void is larger than the interpolation kernel, the procedure should be referred to as extrapolation.

**POD interpolation**

The POD (Proper Orthogonal Decomposition) method is a way to extract 'preferred' structures from an ensemble of velocity fields. While a velocity field can be decomposed in many ways, the POD is special in that the modes found in this decomposition provide the most energy-efficient way of reconstructing the velocity fields. This means that if one would try to reconstruct a single velocity field to a degree that a certain fraction of the energy in the flow is captured, a minimal number of modes would have to be used in the reconstruction. The details of the decomposition itself will be treated further in section 6.3.6. This decomposition has been used to interpolate voids in velocity fields by Gunes et. al. [36] amongst others. The method used is to calculate the eigen modes of the decomposition in such a way that the rejected vectors are not used. From a reconstruction of the velocity field using the POD modes, the missing vectors are calculated. There is, however, a problem with this POD interpolation method. One of the necessary conditions is that data may be missing at various locations and timestamps, but if there are any locations where data is missing at all times, the method breaks down. For the interpolation described in this thesis, a 'hybrid' method was therefore tried. With this method, first a POD decomposition was determined over all vectors excluding those that are missing from all velocity fields (these are the vectors that were masked out due to overexposure of parts of the images). This results in a number of POD-basisfields or eigen modes with the same vectors missing. The idea behind the proposed method is that, as the most energetic POD modes have rather large, regular structures, it might be possible to interpolate these basisfields using conventional methods to an acceptable degree. When these interpolated basisfields are then used to reconstruct the velocity fields using the coefficient matrix determined by projecting the velocity fields with voids onto the POD basis with voids, it may be possible to recover a large part of the energy in the masked sections. The higher order (lower energy) POD modes mostly have only small scale flow structures that represent the smaller scale turbulent fluctuations. It is not expected that there will be any advantage on this level as the turbulent velocity fluctuations can neither be recovered by interpolation in the actual velocity fields nor in the POD-basisfields.

**Comparison of interpolation methods**

Which interpolation method is preferred is not straightforward. One should use an objective test method which is able to determine which interpolation method is best capable of reconstructing the missing parts of the velocity fields. Ideally, one should use an ensemble of velocity fields that have a low number of spurious vectors. From these velocity fields
one can then reject groups of (valid) vectors which resemble the groups in the case of an
overexposed image. The resulting voids could then be interpolated and compared with the
parts that were removed. Unfortunately, no such ensemble is available in the measured
data sets. Also, the generation of complete artificial velocity fields is not straightforward
as the real flow is highly three-dimensional. As an alternative criterium, the turbulent
energy spectrum calculated from the interpolated velocity fields was used. From earlier
measurements on a steady flow set-up using a similar cylinder head and a comparable
velocity range [18], the energy spectrum calculated showed the typical ‘$-5/3$’ slope that
is characteristic for the inertial range of isotropic turbulence decay. While this situation
is slightly different in that it is a confined volume which is not even constant and the
dissipation rates may also be different, it is reasonable to expect a more or less similar be-
havior of the energy spectrum. When the energy spectrum is calculated from un-validated
velocity fields, there is no resulting slope detectable at all since the missing parts distort
the spectrum at almost all length scales (wave numbers). It was assumed that if the
energy spectrum as calculated from the interpolated velocity fields shows a slope similar
to isotropic turbulence decay, then at least the larger scales have been reconstructed to a
reasonable degree and the needed quantities for this research, being the large scale turbu-
 lent velocity fluctuations and the integral length scale can be determined. The procedures
used for the calculation of the energy spectrum and the integral length scale are explained
in section 6.3.5.
Based on visual interpretation of the interpolated velocity fields, is seems that the POD
method is quite capable of interpolating even large sections. There are however some
issues with this method. When a substantial part of the velocity field is missing, the
interpolation of the POD-basisfields becomes questionable as well as the degree of inter-
polation/extrapolation is larger. But also, the calculation of the coefficient matrix
results in a reconstruction vector that is not representative for the velocity field to be
interpolated as a whole. For smaller voids the method seems to work reasonable. On the
other hand, for smaller voids the degree of extrapolation is less and the adaptive Gaus-
sian window technique also performs better. Taking all into account, it was decided to
perform the interpolation of the velocity fields using the adaptive Gaussian convolution
as this method is the most universal to use and is also computationally less demanding.
For both methods remains to be said that recovery of the smaller scales in the flow is
not possible. As an example, the velocity field from figure 6.4(a) is shown with rejected
vectors interpolated in figure 6.7(a). For data sets which have larger groups of rejected
vectors, the interpolation cannot be justified. Figure 6.7(b) is an example of a velocity
field for which the degree of interpolation cannot be justified. The reason for this decision
here is that clear discontinuities in the flow structures exist which are considered unlikely.
The decision to reject or accept the interpolation however, remains a weak point in this
process as it requires user input.

6.3.3 Mean velocity field and turbulence intensity
The mean or ensemble averaged velocity field is calculated from the (interpolated) velocity
fields using equation 3.24, which for a finite number of samples $n$, recorded at the same
crank angle $\theta$ becomes

$$\bar{u}_{EA}(x, \theta) = \frac{1}{n} \sum_{i=1}^{n} u_i(x, \theta).$$

(6.3)
Measurement of velocity field and turbulence in the engine cylinder

Figure 6.7: Various degrees of interpolation

(a) 'light' interpolation
(b) 'heavy' interpolation

Figure 6.8: r.m.s. velocity fluctuation and turbulence intensity, n=1000rpm, MAP=80kpa, θ = 320°CA, mean velocity field as vectors
The turbulence intensity is then calculated by dividing the r.m.s. velocity fluctuation

$$u'_{\text{rms}}(x, \theta) = \sqrt{\frac{1}{n} \sum_{i=1}^{n} (u_i(x, \theta) - \bar{u}_{EA}(x, \theta))^2}$$

by the mean velocity $\bar{u}_{EA}(x, \theta)$. The r.m.s. velocity fluctuation is plotted in figure 6.8(a). Here the 'total' r.m.s. fluctuation $\sqrt{u'^2_{\text{rms}} + v'^2_{\text{rms}}}$ is plotted as a scalar field. In the same figure, the mean velocity field is also plotted as vectors. Already in this figure, an effect of the interpolation becomes visible in the form of areas with lower r.m.s. fluctuation in the upper left and lower right areas. In these areas, the fluctuation is attenuated towards the edge of the domain as the degree of extrapolation is too large. Furthermore, it shows that the center of the global rotation has relatively low absolute velocity and also low fluctuation levels. In the section below the cylinder center, the absolute velocity is higher and also is the velocity fluctuation. In figure 6.8(b), the relative fluctuation or turbulence intensity $u'_{\text{rms}}/|u_{EA}|$ is plotted. Here it shows, in contrast to figure 6.8(a) that in the center of rotation, the relative fluctuation is high and below the cylinder center it is low indicating a stable region. The mean turbulence intensity is less than unity in the major part of the domain, indicating that the mean velocity which convects any flames is higher than the turbulent velocity fluctuation at the largest scales.

One issue with this definition of the turbulent velocity fluctuation however is that it assumes that the mean velocity is the same for all engine cycles. This is not necessarily the case as there can be a large scale cycle-to-cycle variability in the mean flow. The turbulence intensity would be overestimated in that case since part of the large scale fluctuation would be accounted to turbulence (see also [40, 48]). To separate the cycle-to-cycle fluctuation from the in-cycle velocity fluctuations, one would typically need a velocity sampling method that provides a high temporal resolution like high speed PIV or LDA. In that case, one can determine the cycle-to-cycle variation of the mean flow by comparing the mean velocity in a crank angle interval within each cycle. With PIV measurements conducted at a low frequency such that only one velocity snapshot is available every so many cycles, it is however not possible to perform an analysis of the temporal evolution of the velocity field and use this to separate the cycle-to-cycle fluctuations from the in-cycle velocity fluctuation. As an alternative, Li [48] uses a technique that employs a spatial filtering of the single velocity fields instead of a temporal filtering of the time series at one location. The instantaneous velocity at crank angle $\theta$ in cycle $i$, $u(x, \theta, i)$ is split into an average component and a separate low- and high frequency fluctuating component according to

$$u(x, \theta, i) = \bar{u}_{EA}(x, \theta) + u'_{LF}(x, \theta, i) + u'_{HF}(x, \theta, i)$$

Here, the mean velocity $\bar{u}_{EA}$ is averaged over many cycles, the low frequency fluctuation $u'_{LF}$ is associated with the cycle-to-cycle fluctuation and the high frequency fluctuation $u'_{HF}$ is associated with the in-cycle turbulent fluctuations. The procedure used is as follows. First the ensemble mean $\bar{u}_{EA}$ is calculated using equation 6.3. After this, the separate velocity fields are low pass filtered using two dimensional Fourier filtering with a chosen spatial cutoff frequency. This step yields an 'apparent' in-cycle mean velocity $u_B$. The low frequency fluctuating component is then calculated from

$$u'_{LF}(x, \theta, i) = u_B(x, \theta, i) - \bar{u}_{EA}(x, \theta).$$
The high frequency fluctuating component is found using equation 6.5:

\[ u'_{HF}(x, \theta, i) = u(x, \theta, i) - \bar{u} \overline{EA}(x, \theta) - u'_{LF}(x, \theta, i) \]  

(6.7)

The sum of low and high frequency fluctuating components is of course equal to the total fluctuation about the mean \( u'(x, \theta, i) \). The selection of the spatial cutoff frequency is arbitrary but has a large influence on the low and high frequency fluctuating velocity components. Li [48] chooses this cutoff frequency by evaluation of the resulting high frequency fluctuating velocity field \( u'_{HF} \), together with the power spectral density plot of the total velocity. The criterium used for the selection of the cutoff scale is that the appearance of organized structures in the high frequency fluctuating velocity field should be very limited. In figure 6.9, the high frequency fluctuating velocity field is plotted for a spatial cutoff frequency of 0.05 \( mm^{-1} \) and 0.025 \( mm^{-1} \), corresponding to a spatial dimension of 20 and 40 \( mm \) respectively. With 0.05 \( mm^{-1} \), no organized structures like vortices can be detected. With 0.025 \( mm^{-1} \), some small vortices and organized structures are visible. In the corresponding PSD in figure 6.10 (averaged over all velocity fields of the ensemble), it can be seen that the energy also starts to fall off at about 0.025 \( mm^{-1} \). The visual appearance of flow structures in the high frequency fluctuating velocity field does not happen at exactly the same cutoff scale for all conditions, also there are differences from one field to the next. As a compromise, it therefore seems reasonable to use 0.033 \( mm^{-1} \) as cutoff for the separation of cycle-to-cycle fluctuation and in-cycle turbulence. The measured sets of velocity fields have been separated using this threshold, the resulting in-cycle turbulent velocity fluctuations \( u'_{rms, HF} \) are summarized in table 6.2.

6.3.4 Swirl center and vorticity

The mean velocity fields show a global rotation. This might be expected since the cylinder head is designed as a swirl system. The center of rotation is not influenced very much by the engine speed and manifold pressure. This center of rotation was determined using...
Figure 6.10: Power spectral density plot, u-component

<table>
<thead>
<tr>
<th></th>
<th>n=600rpm</th>
<th>n=1000rpm</th>
<th>n=1350rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>IMEP=4 bar</td>
<td>$u'_{rms,HF}=1.41\text{m/s}$</td>
<td>$u'_{rms,HF}=2.69\text{m/s}$</td>
<td>$u'_{rms,HF}=3.39\text{m/s}$</td>
</tr>
<tr>
<td></td>
<td>$\sigma(u'_{rms,HF})=0.52\text{m/s}$</td>
<td>$\sigma(u'_{rms,HF})=0.56\text{m/s}$</td>
<td>$\sigma(u'_{rms,HF})=1.04\text{m/s}$</td>
</tr>
<tr>
<td>IMEP=8 bar</td>
<td>$u'_{rms,HF}=2.44\text{m/s}$</td>
<td>$u'_{rms,HF}=3.38\text{m/s}$</td>
<td>$u'_{rms,HF}=3.87\text{m/s}$</td>
</tr>
<tr>
<td></td>
<td>$\sigma(u'_{rms,HF})=0.52\text{m/s}$</td>
<td>$\sigma(u'_{rms,HF})=0.97\text{m/s}$</td>
<td>$\sigma(u'_{rms,HF})=0.97\text{m/s}$</td>
</tr>
</tbody>
</table>

Table 6.2: High frequency turbulent velocity fluctuations, $\theta = 340^\circ$ CA
a vortex identification algorithm as used by Grosjean [32] and Graftieaux [30]. This algorithm searches the vortex center using the non-dimensional angular momentum $\Gamma_1$. This factor quantifies the average angle between a velocity vector $\mathbf{u}_{i,j}$ and the local position vector $\mathbf{r}_{i,j}$.

$$\Gamma_1 = \frac{1}{N_i N_j - 1} \sum_{N_i} \sum_{N_j} \frac{\mathbf{r}_{i,j} \times \mathbf{u}_{i,j}}{|\mathbf{r}_{i,j}| \cdot |\mathbf{u}_{i,j}|} = \frac{1}{N_i N_j - 1} \sum_{N_i} \sum_{N_j} \sin(\theta)$$  \hspace{1cm} (6.8)

in which $N_i \cdot N_j$ is the size of the area used for evaluation. Here, $\Gamma_1$ was evaluated over a domain of 5x5 vectors. $\Gamma_1$ equals one for a solid body rotation rotating in positive direction and -1 for a rotation in negative direction. In figure 6.11(a) the position of the swirl center for the engine speeds and manifold pressures from table 6.1 at 340°CA is displayed. The location of the swirl center is not much influenced by the engine speed and manifold pressure, the angular speed increases with engine speed and decreases with manifold pressure. In figure 6.11(b), the precession of the swirl center for crank positions ranging from 310-340° and 380-430° CA is displayed along with the rotational speed of the mean vortex. During compression the vortex center moves towards the cylinder center, and also the speed of rotation increases. This behavior is expected. When the piston approaches TDC the rotating charge mass is compressed into the piston bowl, decreasing its moment of inertia. Since (neglecting dissipation) momentum is preserved, the speed of rotation should increase. During expansion (380-430°CA), the center of rotation remains where it is but the angular speed decreases again.

The vorticity $\omega_z$ was calculated using

$$\omega_z = \frac{\partial v}{\partial x} - \frac{\partial u}{\partial y}$$  \hspace{1cm} (6.9)

As expected, the vorticity calculated from the mean velocity field as displayed in figure 6.12(a) gives a similar impression. The vorticity is fairly uniform in the region of the main vortex, indicating an approximate solid-body rotation. In figure 6.12(b), the vorticity calculated from an individual velocity field is plotted in the same figure as the high frequency fluctuating field derived from the same velocity field. The extreme values in the vorticity field coincide with small vortices in the high frequency fluctuating velocity field. The vorticity in the complete field is determined by velocity gradients and therefore related to the velocity fluctuations on a smaller scale.

In a similar fashion, using the gradients in the velocity field, the shearing strain rate $\varepsilon_{xy}$ and extensional strain rates $\varepsilon_{xx}$ and $\varepsilon_{yy}$ can be determined using

$$\varepsilon_{xy} = \frac{\partial u}{\partial y} + \frac{\partial v}{\partial x}$$  \hspace{1cm} (6.10)

$$\varepsilon_{xx} = \frac{\partial u}{\partial x}$$  \hspace{1cm} (6.11)

$$\varepsilon_{yy} = \frac{\partial v}{\partial y}$$  \hspace{1cm} (6.12)

The spatial resolution of the vector grid (1.6 x 1.6mm) is too coarse to determine the vorticity and strain rates at the level the turbulent combustion takes place. However, it may still give some information on the type of turbulent flow the propagating flame experiences. The maximum strain rates found this way are in the range 500-1000s$^{-1}$, in absolute sense.
Post processing of measured velocity fields

Figure 6.11: Swirl center position and angular speed

Figure 6.12: Vorticity
6.3.5 Turbulent energy spectrum and integral length scale

Energy spectrum

The turbulent energy spectrum was already mentioned in section 6.3.2. This turbulent energy spectrum was calculated from the velocity fields using a standard FFT procedure. The calculation of a Fourier transformation using FFT assumes that the domain is periodic (continuous). In this case the (velocity) data is on a rectangular grid, while the geometry is a cylinder. To provide a data set that is continuous, the procedure used was first to define a circle inside the domain. On this circle a number of evenly spaced points is chosen. The number of points depends on the grid spacing and the radius of the circle and is chosen such that the spacing between successive points is smaller than the grid spacing so that the velocity can be resolved down to the smallest scales possible (Nyquist limit). The velocity is then interpolated from the cartesian grid onto the points on the circle. In each point, the actual velocity vector \( \mathbf{u} \) is split into a component that is tangential to the circle \( \mathbf{u}_t \) and a component that is perpendicular to the circle \( \mathbf{u}_r \) and therefore in radial direction. Thus, 2 data sets \( \mathbf{u}_t \) and \( \mathbf{u}_r \) have been constructed to which the FFT is applied. The interpolation process from the cartesian grid onto the points on the circle may not be fully accurate and introduce certain errors. Therefore, the procedure was first calibrated using several artificial velocity fields which consisted of a periodic fluctuation in circumferential direction with fixed frequency of which the energy \( u^2 \) is known. The calculated energy spectrum, in that case a single peak, then shows how much of the input energy \( E(k) \) is found. It turns out that a certain fraction of the energy is lost in the interpolation procedure. This fraction is wavelength dependent and is higher at smaller wavelengths \( \lambda \), or higher wavenumbers \( k = 2\pi/\lambda \). The ratio \( E(k)/u^2 \) was then used to correct the energy spectrum calculated from the real velocity fields. More on this correction procedure can be found in [18]. Two examples of the calculated energy spectrum are shown in figure 6.13. These spectra show a certain slope, which is not exactly \(-5/3\). Also, at the high wavenumber end of the curve, the energy seems not to decrease anymore but instead remains fluctuating around the same level. It is likely that this is caused by the interpolation of the velocity fields as the smallest scales cannot be reconstructed fully. The path along which the spectrum is calculated crosses the interpolated sections of the velocity field several times. For each separate velocity field, a spectrum is determined and these spectra are averaged over the velocity fields in a data set to give the result as displayed in figure 6.13. The different scale on the \( E(k) \)-axis is of course due to the higher velocities at a higher engine speed. The calculated average slope as plotted in the legend is also somewhat influenced by the scattering of the data-points. On average, the slope for the spectrum in radial direction is a bit shallower.

Integral length scale

The integral length scale of the flow can be estimated using the spatial correlation function ([61, 10, 48])

\[
R(r) = \frac{1}{n} \sum_{i=1}^{n} \frac{u'_i(x) \cdot u'_i(x + r)}{u'_{rms}(x) \cdot u'_{rms}(x + r)}
\]  
(6.13)
The integral length scale is then found using

\[ L_i = \int_0^\infty R(r)dr \]  

(6.14)

There is some debate in literature ([62, 61]) on which integration domain should be used for the evaluation of equation 6.14. It was chosen to integrate up to the first zero-crossing of \( R(r) \), as also used by O’Neill [62]. It is thereby assumed that the fluctuation around zero after the first zero-crossing has no further contribution to the integral. For the velocity fields the correlation function was calculated separately for the longitudinal and transversal directions and also for a path in x- and y-direction. The four correlation functions \( R_{x,t}, R_{x,l}, R_{y,t} \) and \( R_{y,l} \) were determined for a number of points randomly distributed in the domain after which the functions were averaged. The integration of these four functions resulted in four estimates for the integral length scale per velocity field ensemble. The integral length scales thus obtained for \( \theta = 340^\circ \) at different engine speeds and manifold pressures are summarized in Table 6.3. Similar to the results of Nomura [61], the longitudinal correlation coefficient results in longer length scales than the transversal correlation coefficient. A similar table for the calculated integral length scales for the crank angle range 310-340° and 380-430° CA at 1000rpm and 80kPa manifold
Table 6.4: Integral length scale estimations, 1000rpm, MAP=80kpa

<table>
<thead>
<tr>
<th>°CA</th>
<th>(L_{x,\ell}) [mm]</th>
<th>(L_{x,\ell}) [mm]</th>
<th>(L_{y,\ell}) [mm]</th>
<th>(L_{y,\ell}) [mm]</th>
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</thead>
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<tr>
<td>310</td>
<td>9.5</td>
<td>14.5</td>
<td>10.3</td>
<td>12.0</td>
</tr>
<tr>
<td>320</td>
<td>8.1</td>
<td>14.3</td>
<td>9.5</td>
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</tr>
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<td>13.0</td>
<td>8.9</td>
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</tr>
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<td>12.5</td>
<td>7.7</td>
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</tr>
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<td>13.5</td>
<td>8.8</td>
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<td>12.4</td>
<td>9.0</td>
<td>12.5</td>
</tr>
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</table>

6.3.6 Decomposition of the velocity field - Proper Orthogonal Decomposition

The POD (Proper Orthogonal Decomposition) technique was mentioned already in section 6.3.2. The method itself is described very well in literature ([30, 6, 26, 71, 11, 19] and others).

The POD decomposition in this work is carried out using the so called 'snapshot' method and consists of the following steps. An ensemble of \(N\) velocity fields is rearranged in a matrix \(U\) of size \((n_x \cdot n_y) \times N\), in which \(n_x \cdot n_y\) is the total number of vectors in one field. Then the correlation matrix \(C\) is determined from

\[
C = \frac{1}{N} U^T \cdot U. \tag{6.15}
\]
From this correlation matrix, the eigenvalues $\lambda_1 \ldots \lambda_N$ and eigenvectors $V_1 \ldots V_N$ are determined. If needed, the eigenvalues (and corresponding) eigenvectors are rearranged in descending order. The POD basis fields or eigen modes $\Phi^{(1)} \ldots \Phi^{(N)}$ are determined by projecting the velocity field ensemble onto the eigenvectors giving $N$ POD basis fields of $n_x \cdot n_y$ vectors which are normalized afterwards. Using the POD decomposition, a single velocity field can be represented as a linear combination of basis fields

$$U^{(i)} = \sum_{j=1}^{N} a_{i,j} \Phi^{(j)}$$

(6.16)

The coefficients $a_{i,j}$ are found through projection of the velocity field onto the POD basis. The POD decomposition is optimal in a sense that, to capture the energy in a velocity field, a minimal number of modes $\Phi$ has to be used. Equation 6.16 is exact if all eigen modes are included in the summation. Thus, using a number of modes $\Phi < N$, one can reconstruct the velocity field to a certain degree. The eigenvalues $\lambda_1 \ldots \lambda_N$ indicate how much energy is associated with each eigen mode.

In a real flow, the POD basis fields can be thought of as flow structures and so, the main features of the flow can be captured by using only a limited number of modes.

In the analysis of the velocity fields using the POD decomposition, it is again confirmed that the global rotation dominates the velocity field. The most energetic POD eigen mode shows this rotation. This is true for all velocity fields evaluated. From the corresponding eigenvalues, it is found that the global rotation accounts for approximately 60% of the total flow energy. On top of this, several smaller eigen modes are present. These eigen modes each contribute only a few percent to the total energy. Typically, the contribution of any mode after the first ten contributes less then 1 percent to the energy. A typical energy distribution among the first 10 eigen modes averaged over the complete velocity field ensemble, again for the same data set, is displayed in figure 6.14(a). The first eigen mode has an average contribution of more than 60%. In figure 6.14(b), the variation of these contributions within the same ensemble is shown. The first, most energetic, eigen mode also shows less variation indicating a more stable flow. In figure 6.15(a), the evolution of the eigen mode coefficients is shown. Again it can be seen that the main vortex is relatively stable, although there exist some velocity fields that have a low contribution from the main mode. Inspection of these fields indeed shows no dominating rotation. The coefficients for the remaining eigen modes show a fluctuating behavior, even changing sign randomly. It is striking that there is no eigen mode beyond the first that shows a consistent contribution. A probable reason for this is that all measurements have been made with the intention of providing information close to TDC, and therefore the crank angle range is limited to a small part before and after TDC where the flow is directed into a rotation by the design of the cylinder head and piston bowl. Further measurements at different crank positions may show a more pronounced multi-mode behavior. In figure 6.15(b), a similar development is shown for the energy contribution of the same eigen modes. It shows clearly that the remaining modes, however large the fluctuation of the respective POD coefficients, hardly contribute to the total energy of the flow. The variation of the modes after the first mode can be viewed as velocity fluctuations, but in doing so one has to keep in mind that the POD decomposition is not a scale separation like a Fourier analysis. It is however clear that the smaller coefficients belonging to the higher eigen modes result in small velocity contributions. The presence of a long 'tail' in the POD energy contribution indicates a kind of randomness resembling small scale turbulence.
Measurement of velocity field and turbulence in the engine cylinder

Figure 6.14: POD energy distribution, MAP=80kpa, n=1000rpm, θ=320°CA

Figure 6.15: POD energy development, MAP=80kpa, n=1000rpm, θ=320°CA
6.4 Discussion

In this chapter, measurements of the flow field inside the engine combustion chamber of a motored engine have been described. The use of the PIV technique, in a low-frequency measurement system has provided snapshots of the velocity field in the part of the combustion chamber that is visible through a window in the piston crown. These velocity fields have been validated and the rejected vectors have been interpolated. The mean velocity field was determined and showed an approximate solid body rotation for all combinations of engine speed and manifold pressure measured. For the determination of the turbulent velocity fluctuations, a spatial filtering method was used to separate the cycle-to-cycle fluctuations from the in-cycle turbulent velocity fluctuations. The resulting r.m.s. velocity fluctuations were found to be in the range 1.5-3.5 m/s, depending on the engine speed and manifold pressure. The topology of the velocity field was investigated using the non-dimensional angular momentum. The center of the main rotation was found to move toward the cylinder center during compression. During this process, the rotational speed of the main vortex increased. During expansion, the center of rotation remains at the cylinder center and the rotational speed decreases again. The integral length scale of the flow field was determined using a separate longitudinal and transversal autocorrelation function. The integral length scale in longitudinal direction was found to be larger than in transversal direction, something also found by other researchers. The integral length scale was determined at 7-10 mm for all measurements conducted at 340°CA. This integral length scale was not dependent on the engine speed and manifold pressure. For a set of measurements conducted at various crank positions before and after TDC, it was found that the integral length scales at crank positions further from TDC were on average somewhat larger but only by about 20%. The energy spectrum was determined using a standard Fourier analysis and showed, on average, a behavior resembling isotropic turbulence decay with a slope of -1.4...-1.6. The Proper Orthogonal Decomposition was used to investigate the time-varying topology of the velocity fields. It was found that the velocity fields can be characterized best by a single large vortex as was already seen in the mean velocity fields. The POD analysis showed that this main vortex accounts for approximately 60% of the energy in the velocity field. The higher eigen modes were found to contribute no more than a few percent each at most. Also, none of the higher eigen modes provided a consistent contribution. Instead, all these modes have small fluctuating contributions.
Chapter 7

Burning velocity measurements

7.1 Introduction

As already mentioned in chapter 3, one of the key parameters for identifying the combustion regime in a SI engine is the laminar burning velocity of the mixture that is burned in the engine combustion chamber. This laminar burning velocity is a fundamental property of the mixture, but depends on the mixture composition (equivalence ratio, dilution with inert components) and the conditions (pressure, temperature) of the unburned mixture before it is consumed by the flame. To identify the combustion regime, the laminar burning velocity at the pressure and temperature occurring in the engine must be known. One may think that this is straightforward, however these data are not readily available from literature. Literature data is mostly limited to a narrow range of pressure and temperature, often around atmospheric conditions. The equivalence ratio for which data can be found is around stoichiometric with small deviations to the fuel rich side and a somewhat larger range toward the lean side. Also, the available data is mostly given for relatively simple fuels like methane. Realistic fuels however are a complex mixture of different fuel components and sometimes even inert species. For this research not only lean equivalence ratio’s are considered, but also a comparison will be made with mixtures that are diluted with inert components as found in recycled exhaust gases (EGR). Nowadays, computer codes which calculate the chemical kinetics have evolved to a point at which the burning velocity of virtually any mixture can be calculated from fundamental properties. However, numerical codes are still validated against real experimental data. This means that the calculation of burning velocities for more "standard" conditions, typically room temperature and atmospheric pressure and "simple" fuels, can be calculated to agree with experimental data. For conditions departing further from these cases, in this research high pressure and temperature and lean equivalence ratios, experimental data is rare and therefore the predictions of these codes cannot be trusted fully. It was therefore decided to measure the burning velocity at the conditions needed by experiment. For the engine experiments in this research, both natural gas and methane were used. Ideally, the burning velocity for both fuels should be known. The composition of natural gas from the grid however is rather complex and is not even constant in time. Since the definition of an average, representative composition of natural gas is difficult, it was therefore chosen to use methane as reference fuel for the research presented in this thesis. Thus, the burning velocity of methane was measured instead of some ‘average’ natural gas. Even with this limitation, the amount of measurements using only methane at various
equivalence ratios and dilutions was fairly high.

7.2 Method

There are several methods available for measuring the laminar burning velocity of a fuel/air mixture. These methods vary in complexity and accuracy and also in the range of conditions that they can be used in. For instance one of the more accurate methods, the flat flame burner ([7]), can be used to determine the burning velocity relatively accurate, but is limited by burner construction to rather low pressures (and temperatures). The most common method that is used at high pressure and temperature is the method of constant volume combustion, also known as the "combustion bomb" method.

7.2.1 Constant volume combustion

With this method, a constant volume vessel is filled with a combustible mixture of fuel and oxidizer up to a certain pressure. The initial temperature is controlled by the temperature of the vessel itself. The mixture is then ignited in the center of the vessel, after which a flame starts to grow from the center outward. During this combustion process, hot combustion products are produced and the temperature and pressure in the vessel rise. Due to the expansion of the hot burned products, the unburned mixture surrounding it is also compressed. As the compression of the unburned mixture takes place very rapidly, its temperature increases also. Because of this coupling of pressure and temperature, the state of the unburned mixture follows a path along a pressure-temperature curve. Mostly, this compression is assumed to be adiabatic which gives the relation \( T = T_i \cdot \left( \frac{p}{p_i} \right)^{(1-\frac{1}{\gamma})} \), in which \( \gamma \) is the ratio of specific heats. In reality, there is heat loss to the walls of the vessel so the adiabatic relation does not hold exactly and the resulting pressure is lower.

7.2.2 Burning velocity calculation

The determination of the burning velocity from an explosion in a constant volume vessel can be performed in different ways, either by using the measured pressure during combustion or by using the expansion rate of the burned products as found through high speed filming of the combustion process. With the first method, the mass burning rate \( \dot{m}_b \) is found using a calculation model solving conservation of mass and energy. The actual burning velocity \( S_L \) is then calculated using this mass burning rate, together with the density of the unburned mixture \( \rho_u \) and the assumption of a certain flame-shape. The second method can be applied when the vessel is relatively large. In that case the pressure rise during the first stages of combustion is negligible and the state of the unburned mixture can be assumed to be constant. The burning velocity can then be found from the expansion rate \( dr/dt \) of the sphere of burned products, corrected for the density ratio \( \rho_b/\rho_u \). In this work the vessel was relatively small and the first method was used for calculating the burning velocity. For most combustion events however the development of the flame was filmed to validate the assumption of laminar combustion.
7.2.3 Calculation model

As already mentioned in section 7.2.2, the burning velocity is calculated from the measured pressure rise in the vessel. The model that was used for calculating the burning velocity is referred to as a two-zones model. The model was adapted from a two-zones calculation model that was developed for use in engine combustion ([23]), the basic assumptions however are the same. More information on this model can be found in appendix E. With this type of model the vessel (combustion chamber) is divided in two zones, one zone which consists of hot combustion products and a second zone which consists of unburned mixture heated by the compression due to pressure rise. The two zones are separated by the flame front which is assumed to be of negligible thickness. Heat transfer between the zones is assumed negligible. The pressure is assumed uniform throughout the vessel. This assumption is valid since the combustion takes place in the form of a deflagration. The composition of the unburned gases is assumed frozen. For the composition of the combustion products chemical equilibrium is calculated.

Heat transfer from the hot gases to the walls of the vessel is modeled using simple convective heat transfer relations of the form \( \dot{Q} = hA(T_{\text{gas}} - T_{\text{wall}}) \). This heat transfer was found to be of little influence for as long as the flame does not touch the walls of the vessel. As combustion progresses, eventually the flame will touch the walls and heat transfer increases considerably due to the higher temperature difference between the hot products and the walls. From this moment on, the assumptions made on the value of the heat transfer coefficient \( h \) have a large influence on the calculation of the energy released in the later stages of combustion. Due to the cubical shape of the vessel however, a large part of the initial mixture is still to be burned when the expanding sphere of combustion products already touches the walls. This deficiency will be explained further in section 7.5.1. Due to buoyancy effects with slow burning mixtures, the sphere of combustion products touches the top surface of the vessel first. This effect, together with the less well defined shape due to buoyancy, makes estimation of the contact surface of the hot products with the wall difficult. It is therefore nearly impossible to extract burning velocity information from the latter part of the combustion process.

In reality, the gases in the zone with burned products are also compressed by the rising pressure in the vessel. Due to this effect, there exists a temperature gradient in the burned products. The gases that are burned in an early stage, are first burned and compressed after that. Gases that are burned in a later stage of combustion are first compressed and then burned. Due to this effect, there exists a temperature difference between the core and the outer layer of the zone with burned products that can be as large as several hundred degrees. Recently, there have been initiatives to model the zone of burned products as several layers instead of a single zone ([73],[50]) to investigate the effect of this temperature gradient. It was found that the effect of neglecting this phenomenon on the predicted burning velocity is only small.

Flame stretch

In chapter 3, section 3.2.4, it was explained that a premixed flame can be influenced by flame stretch. Depending on the values of the Lewis number \( Le \) and the associated Markstein number \( M \), the flame temperature \( T_b \) is altered and, as a consequence, the burning velocity \( S_L \) changes. In almost all recent attempts to measure burning velocities using a constant volume vessel, using either the pressure rise ([21],[69]) or constant pres-
sure technique ([33],[49],[38]), this flame stretch is corrected for. Some researchers who use the constant pressure technique go as far as the derivation of the Markstein numbers from their results, others assign a value for this quantity. The burning velocity is then corrected from the stretched burning velocity $S_L$, as obtained from either the pressure rise or from filming the flame expansion, to the unstretched value $S_{L,0}$ using equation 3.22. The value for the flame stretch $K_A$ is derived from the flame radius and expansion rate using $K_A = \frac{1}{A} \frac{dA}{dt} = \frac{2}{r} \frac{dr}{dt}$, where $r$ is the flame radius. This correction is almost universally applied. It can be shown however ([1]) that when the pressure signal is used to calculate the burning velocity, and the Lewis number $Le$ does not deviate from unity too much, this correction can be omitted. In this work, where only methane was used as a fuel, the Lewis number is 0.97 which is close to unity. Stretch correction was therefore not applied for the measurement of burning velocities.

7.3 Experimental set-up

Mostly, for measuring burning velocities a spherical vessel is used ([75, 21, 43, 57, 8, 33, 69] and many others). In this case, since a constant volume vessel was already available, this was used for this study. The vessel however was not specifically designed for measuring burning velocities. This meant that the set-up had to be adapted at certain points. The vessel itself as well as the gas admission system will be described in detail in section 7.3.1. A Schlieren set-up was used for visualizing the combustion. In section 7.3.2, this system will be described.

7.3.1 Constant-volume vessel

The constant volume vessel that was used in these experiments, the EHPC (Eindhoven High Pressure Cell) is a development of the internal combustion engines group. This vessel was originally designed for the visualization of fuel sprays but was adapted for this work to facilitate the measurement of burning velocities. The vessel (figure 7.1) has a cubical internal shape with dimensions $108^3$ millimeters and a resulting volume of 1.26 liters. Into each side of the vessel a large insert can be placed (figure 7.2). These inserts can hold instruments, valves or a window. Into the corners of the cube, smaller inserts can be fitted which can hold smaller items like pressure sensors or spark plugs. Also a small fan is placed in one of the upper corners. The vessel can be heated electrically by heating elements built into each side. The temperature of the vessel is limited to 475K which is the maximum temperature the seals can withstand.

The vessel can be filled with a mixture of a fuel and oxidizer through a set of massflow-controllers. The amount of each component is pre-determined such that the composition as well as the pressure after filling the vessel match the desired values prior to ignition of the mixture. For the experiments described in this thesis, the following set of gases was used: nitrogen, oxygen, methane and carbon dioxide. During the filling of the vessel, the gases are added sequentially and are mixed by the fan. After filling the vessel, the fan is stopped and the mixture is given time to settle to a quiescent condition before ignition. Ignition is by an electric spark that discharges between two electrodes which protrude from 2 opposite corners of the vessel. A standard inductive type ignition system is used. The static pressure during filling of the vessel is measured using an accurate piezo-resistive
pressure transducer (Druck PMP4070) mounted upstream of the inlet valve. This sensor is disconnected from the vessel by the inlet valve after filling is complete. The pressure rise during combustion is measured with a piezo-electric transducer (Kistler 7061B or 6041A with an accompanying charge amplifier) which is mounted in a corner of the vessel.

### 7.3.2 Combustion visualization

The set-up was originally designed for visualizing the injection of fuel sprays; a high speed camera was therefore present already. This camera was also used for these burning velocity measurements to confirm the assumption of a laminar flame. For imaging the flame, the Schlieren technique was used. With this technique, density gradients due to temperature differences are made visible by using the change in refractive index in the measurement section. The optical set-up used is displayed in figure 7.3. A large Xenon lamp is used as a light source. The beam from this lamp is focused onto a pinhole which functions as a point source. Using an achromatic lens, a parallel beam is formed which is directed through the measurement volume. A second achromatic lens which is positioned after the measurement volume focuses the light onto a second pinhole. The camera is placed behind the second pinhole and is focused at a plane in the measurement volume.
In a situation with uniform density in the measurement volume, a certain amount of light is able to reach the camera lens, depending on the intensity of the light source and the alignment of the pinholes. When an event occurs in the measurement volume (i.e., a flame) which creates a density gradient, the parallel beam is refracted at this gradient. The light that is refracted is not focused at the second pinhole and therefore does not reach the camera anymore. Density gradients thus appear as dark areas in the camera image. Examples of this can be found throughout this chapter.

Figure 7.2: EHPC exploded view showing windows and adapters

Figure 7.3: Layout of schlieren set-up
Definition of measurement range and fuels used

In section 7.2, it was mentioned that the pressure-temperature history of the unburned mixture follows a line of approximately adiabatic compression. This implies that the pressure-temperature combinations for which burning velocities can be measured are determined by the choice of initial conditions. For given initial conditions, the pressure-temperature combinations during combustion are more or less fixed. As it makes little difference whether the mixture is compressed by expansion of burned gases or by an approaching piston, the same holds for compression of the unburned mixture in a SI engine. To identify the range of pressure-temperature combinations of interest, pressures occurring in a typical SI engine measured for low- and high-load conditions were taken as the limits for the range of interest. The associated temperatures were estimated assuming adiabatic compression. This resulted in the boundaries as displayed in figure 7.4. Due to the coupling of pressure and temperature in the vessel, initial conditions have to be chosen such that these are in the area between the boundaries for high and low engine load as displayed in figure 7.4. Given the fact that the initial temperature is limited by construction to 475K, this results in an initial pressure that is at 475K restricted to the range 0.4-1.0 MPa. Combinations of initial pressure and temperature that lie outside this range will result in data that is of less interest. The pressure data used for figure 7.4 was however measured in a light-duty engine. For heavy-duty engines, the pressure range is preferably extended somewhat as charge pressures are higher. Therefore, the maximum initial pressure was chosen a bit higher. Based on this, 3 values for the initial pressure have been chosen together with 2 values for the initial temperature.

Methane was chosen as reference fuel for this research and this was also used for the burning velocity measurements. The equivalence ratio of the mixtures was chosen alike that applied in SI engines. As many engines use an equivalence ratio of 1.0 for at least part of their working conditions, this is the highest equivalence ratio of interest. Lean-burn engines however use equivalence ratios down to ∼ 0.6 (λ ∼ 1.7). The limitation in such an engine is often the ignitability of the mixture. Although the ignition conditions in the vessel are different from an engine, it was decided to measure burning velocities

![Isentropic compression curves](image)

Figure 7.4: Pressure-Temperature curves indicating boundaries of range of interest
106 Burning velocity measurements

<table>
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<td>0.5</td>
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<tr>
<td></td>
<td>0.6</td>
<td>1.2</td>
<td>0.6</td>
<td>1.2</td>
</tr>
</tbody>
</table>

Table 7.1: pressure-temperature combinations

at equivalence ratios down to the ignition limit in the vessel. As will be discussed in section 7.5.1, other problems limit the measurement of burning velocities at these lean conditions.

A further objective of this research is to make a comparison between the combustion of lean mixtures and mixtures diluted with inert gases. Therefore also the burning velocity of mixtures diluted with 'simulated' combustion products was measured. These measurements will be discussed in section 7.8.

7.5 Initial experiments

To test the functionality of the set-up and the validity of the two-zones calculation model, an initial series of measurements was carried out. This series consisted of two equivalence ratios, 2 initial temperatures and 3 different initial pressures. The initial conditions have been summarized in table 7.1. In these initial experiments, the focus was on accuracy and repeatability of the mixture composition and of the measured pressure traces, but also a number of combustion events was filmed to observe the behavior of the flame and the type of combustion occurring.

7.5.1 Model check

In figure 7.5, a sequence of images of a typical combustion event ($\phi = 1.0$, $p_i = 0.4 MPa$, $T_i = 400 K$) is displayed. As the figure shows, a smooth sphere of combustion products expands from the center indicating laminar combustion. A few slight cracks are visible on the surface, these grow with the flame but do not distort the laminar character. The size of the sphere of combustion products as calculated by the two-zones model is shown by a dashed line in the figure. This calculated size corresponds well with the observed size and also the moment the sphere touches the walls is predicted relatively well. In figure 7.6, a similar sequence for a lean mixture ($\phi = 0.67$ is shown. In this sequence, a fundamental problem of measuring the burning velocity of lean mixtures occurs. Due to the slow combustion, the sphere of combustion products has time to rise to the top of the vessel. This effectively shortens the part of the event that can be used, as heat loss becomes too severe when the flame touches the wall. Also, the calculation model assumes a spherical flame-shape (indicated again by the dashed circles), whereas this flame is somewhat flattened and has a different volume to surface ratio.

When the pressure curve and the calculated volume of the sphere of hot products are compared, immediately a problem becomes visible. At the moment the flame touches the walls and starts to loose heat, only about 50 percent of the vessel volume is filled with combustion products. With lean combustion this is even worse as the flame hits the top.
wall first. Because of the high density ratio $\rho_u/\rho_b$, this burned zone contains as little as 20% of the total mass. The pressure at this instant is also far lower than the maximum pressure. This phenomenon is caused by the cubical shape of the vessel. At the moment the flame touches the walls, there is a large amount of unburned, cold mixture compressed into the corners of the cube. This mixture is all burned at a later stage, but as explained in section 7.2.3 no burning velocity can be calculated due to the undefined shape and heat losses. With leaner mixtures ($\phi = 0.67$), this problem is even more severe. Due to buoyancy and slow combustion, the sphere of burned products moves upward and touches the top surface even earlier and also, due to the slower combustion of leaner mixtures, there is more time available for heat loss. Often, a spherical vessel is used for these experiments. This solves the problem with unburned mixture in the corners; it however does not alleviate the problem that occurs due to slow burning mixtures. With a spherical vessel, the usable part of the event is still limited due to buoyancy and subsequent cooling at the wall. There is however an advantage which will be shown in section 7.6.1.

7.5.2 Accuracy of the initial charge composition

The accuracy by which the initial charge in the vessel can be composed is determined by the accuracy of metering the gases composing the initial mixture in the vessel. The mixture composition was checked by recording the partial pressure in the vessel after each component was added. The mass of each component can be determined with the use of the ideal gas law. Due to the finite accuracy of the static pressure transducer and the data acquisition system, there is an uncertainty in the measured partial pressures and thus in the determination of the equivalence ratio. Assuming the maximum error (worst case) for all partial pressure measurements, the overall maximum error in the determination of the mixture composition was determined ([79]). Initially, this turned out to be approximately 34% for the conditions in which the amount of fuel was the smallest. These conditions occur with lean mixtures, low initial pressures and high initial temperature. At the other side of the experimental range, $\phi = 1$ and high pressure, the error was still 8.5%. It was also found that the accuracy of the mass-flow controllers was far from optimal. This type of instrument has a high accuracy when working with a continuous gas flow. In this application however, the total mass for some components was so small that the mass-flow controllers could hardly achieve a continuous flow condition. To avoid this problem, the flow setpoint had to be adjusted to a small value. Still, during the opening and closing of the mass-flow controller valve, the flow rate is not constant. To account for this, a correction procedure was developed ([79]) which took the behavior of the mass flow controllers into account and thus reduced the error in the metering of the gas mass per component.

To reduce the uncertainty in the determination of the equivalence ratio from partial pressures, a number of modifications was carried out. Firstly, the pressure transducer that was used for measuring the partial pressures was re-calibrated and found to be about four times more accurate than the manufacturers specifications. This greatly reduced the maximum error. Also, the input range of the data acquisition hardware was adapted. Furthermore, since the small fuel quantity presented the biggest problem in achieving sufficient accuracy in the desired mixture composition, a change was made from using pure methane to using a mixture of 25% methane and 75% nitrogen for the fuel metering.
Because the added mass is roughly six times larger, the accuracy of fuel metering increases also. The remaining quantity of nitrogen, besides the nitrogen that is added with the fuel, is still large enough to ensure accurate metering. Using the modifications just described, the error in determining the equivalence ratio of the mixture was reduced to maximum 4% at the most difficult conditions and on average about 1%.

### 7.6 Extending the pressure-temperature range

In section 7.5.1, it was found that due to the cubical shape of the vessel, only a small part of the total pressure rise could be used for calculating the burning velocity. To overcome this limitation, quite rigorous changes have to be made to the set-up. Apparently the cubical shape which has the advantage of easily incorporating large windows in the set-up is not that suitable for this application. Two principally different options to overcome this problem were tried and tested. Firstly, it was decided to make the shape of the vessel more spherical. A second option that was identified is to change the thermodynamics of the combustion process. Both options will be explained in the following subsections.

#### 7.6.1 Changing the shape of the vessel

When changing the shape of the vessel to a more spherical one, it is assumed that the physics of the combustion process do not change. A larger fraction of the total volume will be filled with combustion products at the instant the flame touches the walls. As the mixture around the flame is compressed into a smaller volume, the pressure for a given flame size will be higher. This will give an advantage over the cubical shape, which is most prominent for fast burning mixtures. With slow burning (lean/diluted) mixtures there is still the problem of buoyancy which even a perfect spherical shape cannot solve.
The only solution for that is to switch off gravity by performing these experiments in outer space or by igniting the mixture while dropping the vessel into a free fall. This problem is seldom addressed by other researchers as experiments are mostly limited to equivalence ratios around unity.

Due to the design of the EHPC, making the inner shape spherical was not a trivial solution. As the vessel was originally manufactured from a solid piece of stainless steel from which the inner cube was removed through spark erosion, it was decided to design a number of inserts which can be positioned in each corner of the cube, thereby filling the larger part of the ‘harmful’ space. Additional constraints were that it should be possible to insert these pieces through the side openings when one or more windows are removed and that there was a possibility to mechanically hold them in place to prevent them from moving to avoid damage to the windows. Optical access should be preserved as much as possible. The resulting inserts were a set of more or less pyramid-shaped pieces which form a polyhedron when mounted. Due to the limited space available for mounting these inserts, the last two pieces that are positioned into place had to be split again, creating a mechanical puzzle (see figure 7.7). The inserts were made from aluminium. When determining the exact size of the inserts differences in thermal expansion coefficient were taken into account to minimize crevices. The internal volume has been reduced from 1.26 to 0.84 liters. As there were also some concerns on the ability of the small fan that was mounted in the corner to mix the gases, a new, more efficient fan was designed as well. The ‘old’ fan was a simple propeller type (figure 7.8(a)), for the new fan an impeller-derived design was chosen (figure 7.8(b)). Although air flow measurements were not performed, the air motion felt with the window removed was greatly enhanced with the new design.

### Expected improvement

Assuming that the combustion process has no knowledge of the shape of the volume in which it is confined and therefore does not change, the expected pressure trace will be steeper in time. When heat losses are negligible, the maximum pressure should be the same as the energy released per mass is the same. The expansion of the sphere of burned products is slower since the pressure in the vessel increases faster. As there is less mass in the resulting unburned volume when the flame hits the walls, the calculation of the
The burning velocity from the pressure trace should be possible up to a higher pressure and temperature. On the modeling side, the reduced surface area of the new shape will lead to somewhat lower heat losses, although this may not be important as long as the flame does not touch the walls. When the flame is modeled as a perfect sphere that grows from the center, 52.3% of the volume is filled with combustion products when the flame touches the walls for the case of the cubical chamber. With the new shape this fraction will increase to 78.2%. For a rough prediction of the pressure and temperature increase, the pressure and calculated temperature at the higher burned volume fraction from an existing measurement in the cubical shape are used. Using this method, the pressure at which the flame touches the wall is predicted about 60% higher. The temperature of the unburned mixture is expected about 15% higher.

**Realization**

In figure 7.9(a), the pressure traces from a stoichiometric combustion in the cubical and new shape with the same initial conditions are compared. Surprisingly, the rate of pressure rise is similar for both vessel shapes. The maximum pressure for the new shape is a bit lower. In figure 7.9(b) the pressure curves for a lean mixture ($\phi = 0.67$) at the same initial pressure and temperature are displayed. With this combustion event, the pressure does rise faster in the new vessel shape. In this case, the maximum pressure is again lower than with the cubical shape. For these initial conditions this means that the usable pressure range is increased from 2.4 MPa to 3.5 MPa for $\phi = 1.0$ and from 2.2 to 3.1 MPa for $\phi = 0.67$. The temperature of the unburned mixture (figures 7.9(c) and 7.9(d)) that is calculated gives a similar view as the compression takes place almost isentropically in both cases. Due to the increase of the pressure range, also the temperature range is extended from 565K to 620K for the stoichiometric mixture and from 553K to 598K for the lean case. Using the two zones calculation model on both pressure traces, it was found that the results for the calculated burning velocity are different for the stoichiometric (figure 7.9(e)) and the lean case (figure 7.9(f)). Since the burning velocity is a fundamental property, this should not change. The calculated burning velocity in the lean case is about the same for both vessel shapes. However, for the stoichiometric case the burning velocity in the new shape is even lower. This will be discussed in the next section. The
Schlieren recordings show that in both cases the flame develops as a thin laminar sphere, only with the new shape some more cracks/wrinkles can be seen. The asterisk marks in the figures indicate the calculated moment in time the flame hits the walls. This point also serves as upper limit for the interval that was used for determining the burning velocity correlations. When the $S_c$-curves however suddenly deviate from a ‘smooth’ curve (for instance the drop that is observed at the end in figure 7.9(f)), the interval was truncated at an earlier stage.

Discussion

The observed phenomena in the new vessel shape do not fully correspond to the predictions explained in section 7.6.1. The lower maximum pressure for the new shape is most likely caused by the presence of crevices in the volume. In these crevices unburned mixture is compressed during the pressure rise due to combustion. Due to the narrow geometry of these crevices, the temperature of the unburned mixture in these crevices does not rise much. Effectively a third zone can be identified in the vessel. This third zone has a similar effect as a thermal boundary layer, only its volume remains constant. In the two-zones model these crevices were adopted as an additional zone in which the temperature remains at its initial value. The effect of this is a mass loss from the main volume as the pressure rises, similar to blow-by in an engine. The changes with respect to the original model are explained in appendix E. The volume of this extra zone was estimated using a number of combustion events in which the mixing fan was not stopped prior to ignition. These combustion events were highly turbulent and showed a much higher heat release rate. It was assumed that due to the much shorter combustion duration with a turbulent flame, convective heat losses to the walls were also much smaller and any deviations from unity for the burned fraction could be attributed to the unburned mixture that was trapped in the crevices. The model could thus be ‘tuned’ to achieve a burned fraction of unity by varying the volume of the crevice-zone. A volume fraction of 1.5% was found for this extra zone. Considering the construction inside the vessel, this value was considered plausible and it was adopted in the model for all combustion events that were performed in the new vessel shape. An explanation for the lower rate of pressure rise for the stoichiometric case (figure 7.9(a)) and the corresponding lower burning velocity (figure 7.9(e)) were not clear. This observation may also be related to the crevices. Due to the order in which the gases were admitted into the vessel relatively more fuel is captured in the crevices and the overall equivalence ratio in the vessel is lower. As this was only discovered after the experiments, further research was not possible and a satisfying explanation was not found. An observation that supports this assumption is the occurrence of visible flames in the vessel after the main heat release for the stoichiometric case. These flames were visible in a movie, shot with a hand-held digital camera at 60 fps., from the combustion of a stoichiometric mixture at an initial pressure of 1.2 MPa. The phenomenon was not observed with the lean mixtures. Since the burning velocity of stoichiometric mixtures was not the main subject of this research, this issue was not investigated further. The results of the stoichiometric cases however have to be treated with care.
Figure 7.9: Comparison of \( p(t) \), \( T_u \) and \( S_L \) for two equivalence ratios in the cubical and polyhedron shape, the asterisk (*) indicates the calculated moment the flame touches the walls.
7.6.2 Changing the thermodynamics of the combustion process

A second option to extend the pressure-temperature range is to change the thermodynamics of the combustion process by changing the composition of the unburned mixture. A similar approach was used by Rahim ([69, 70]). With this technique, the nitrogen in air is replaced by a different gas. Argon is mostly used for this because it is also non-reactive. Replacing the nitrogen in air by argon has two effects. Firstly, due to the lower specific heat capacity of argon the flame temperature is higher. This causes the maximum pressure in the vessel to be higher. Secondly, due to the higher specific heat ratio of argon ($\gamma = 1.67$) compared to nitrogen ($\gamma = 1.4$), the temperature rise for a given pressure rise due to compression of the unburned mixture is higher. This way, burning velocities can be measured for higher pressures and temperatures. For the pressure-temperature range where measurements using nitrogen/oxygen as oxidizer cannot be made, a translation is made from the burning velocities measured using argon/oxygen as oxidizer to the burning velocity of fuel-nitrogen/oxygen mixtures at higher pressures and temperatures. Scaling relations are used to make this translation. For both oxidizers, the burning velocity as a function of pressure and temperature is expressed in a powerlaw form. The ratio of these powerlaws is then used to ‘translate’ from the burning velocities measured with argon/oxygen to those with nitrogen/oxygen as oxidizer at pressures and temperatures beyond the range that can be achieved with nitrogen/oxygen.

Of course there are a number of uncertainties associated with this method. Probably the most questionable of these is the similarity of the fuel-air and fuel-argon-oxygen chemistry and transport. Although thermal conductivity and diffusion are little affected ([14]), simply scaling the burning velocity seems rather crude without thorough investigation of the chemistry. Argon is present as a minor species in most chemical kinetics codes, however the influence of argon in large quantities is not well known. Especially at high pressures, third body reactions may cause unpredicted effects on the chemistry. Rahim also observes cracks at pressures above 2 atm in the methane-argon-oxygen flames and even cellularity at pressures above 5 atm. She mentions that the calculated burning velocity is not the true laminar one, but is unclear on the severity of the deviations. Rahim [69] addresses these issues, but concludes in her work that this ‘extrapolation’ method produced results that are in agreement with measurements from other researchers. Therefore, it was decided to try this approach also.

Realization

A limited number of experiments using argon/oxygen as oxidizer was performed to evaluate the method. These experiments were conducted in the cubical vessel without inserts. The burning velocity was determined for two equivalence ratios, $\phi = 1.0$ and $\phi = 0.67$. Unfortunately the windows of the vessel were not available at the time of the experiments, these were replaced by full metal dummies; the combustion events could thus not be filmed. In figure 7.10(a) the pressure curves from a combustion with nitrogen/oxygen and with argon/oxygen are compared. It is evident that the pressure rises much faster for the argon/oxygen case and also the maximum pressure is higher. When applying the two-zones calculation model to these pressure curves, the results reveal a complication which was not accounted for. In figures 7.10(b) and 7.10(c) the unburned temperature and mass fraction burned from the same combustion event are shown. The asterisk marks in the figures indicate the calculated moment the flame hits the wall (no visual confir-
It can be seen that the maximum pressure and also the unburned temperature in the argon/oxygen case are higher, however due to the larger density ratio $\rho_u/\rho_b$ the flame also expands faster compared to the nitrogen/oxygen case. The sphere of combustion products therefore touches the walls in an earlier stage. At this instant, the temperature of the unburned mixture is about 50K higher, the pressure however is only 0.3 MPa higher compared to the nitrogen/oxygen case. Rahim [69] claims a temperature increase of about 100K. It was therefore concluded that the replacement of nitrogen with argon is less applicable to a non-spherical bomb as it results in a less significant extension of the pressure-temperature range. It was concluded that this small extension of the measurement range did not make up for the uncertainties regarding the similarity of the chemistry and the possible occurrence of cellular flames at higher pressures. This meant that more measurements using argon were of little value at that moment and this method of extending the pressure-temperature range was not further explored. The final measurements were performed with the vessel with inserts and nitrogen/oxygen as oxidizer.

### 7.7 Burning velocity of lean mixtures

The initial measurements were limited to only two equivalence ratios. For the final series of experiments using lean mixtures, the rich limit was kept at $\phi = 1.0$ and measurements were performed up to the ignition limit of the mixtures. This ignition limit is of course temperature and pressure dependent. At very lean mixtures ignition becomes impossible at higher pressures first. It was decided that the unburned temperature range should be as high as possible. Therefore the major part of the measurements was performed at an initial temperature of 475 K. The mixture was leaned out in steps of 0.1 in air excess ratio, up to a $\lambda = 1.5$ ($\phi = 0.67$). Close to the ignition limit, above $\lambda = 1.5$, the air excess ratio was increased in steps of 0.05 with a maximum of $\lambda = 1.8$ ($\phi = 0.55$), which was the leanest for which all initial pressure conditions could be successfully ignited.

As was already mentioned in section 7.5.1, at very lean mixtures the useable part of the pressure curve becomes shorter as the flame rises to the top of the vessel. As an example, in figure 7.11 a number of frames are plotted for four equivalence ratios ($\phi = 0.67$, $0.63$, $0.59$ and $0.55$) for the same mass fraction burned. For $\phi = 0.59$ and $\phi = 0.55$ the assumption of a spherical flame is less and less valid. As the compression of the unburned mixture takes place along an approximate isentrope, plotting the burning velocity as function of pressure or temperature alone is difficult.
Figure 7.11: Buoyancy during first stages of combustion, $T_i = 475K, p_i = 1.2 MPa$
Therefore, in figure 7.12 the measured burning velocity for a number of equivalence ratios is plotted as function of pressure. In these figures the unburned temperature is plotted as isotherms which cross the lines for the burning velocity. The burning velocity has been plotted up to the (calculated) moment the flame hits the wall. At high equivalence ratios, this moment can be calculated pretty well, for leaner mixtures the moment was derived from the complete trace as the calculated burning velocity starts to behave strange since heat loss is not correctly accounted for and also the shape deviates from an approximate sphere. The buoyancy effect is also visible in these figures; for slower burning mixtures at lower equivalence ratios and higher pressures the temperature at the instant the flame touches the walls is lower and the line for the highest pressure extends less far beyond the isotherm for the highest temperature.

### 7.8 Burning velocity of EGR diluted mixtures

Next to the burning velocity of lean mixtures, also the burning velocity of EGR diluted mixtures at various dilution levels was measured. The purpose of these measurements was to make a comparison between both means of mixture dilution (addition of excess...
air or inert gases), based on their laminar burning velocity at the same pressure and temperature. In SI engines, the maximum dilution that is possible with exhaust gases is far lower than the dilution with excess air. In the following sections, the specific experimental details of the EGR measurements will be explained. After that, the results will be discussed briefly.

7.8.1 Diluent composition

In a SI engine, the exhaust gases that are recycled are a product of the combustion in the engine. Depending on composition of the fuel, the equivalence ratio and the combustion efficiency, the exhaust gases are a mixture of nitrogen, water, carbon dioxide, any remaining oxygen and various minor species. For this work only stoichiometric combustion is considered, therefore the recycled gases contain no oxygen. To make a comparison with the lean mixtures, only methane is considered as a fuel. Neglecting the minor species, the exhaust gases consist of 71.5% $N_2$, 9.5% $CO_2$ and 19% $H_2O$ on molar basis. For measuring burning velocities, it is somewhat unpractical to use exactly this composition as the reactants are stored in gasbottles under high pressure. More specifically, water presents a problem, since it is not a gas at storage conditions. Another potential problem is corrosion of various parts and sensors in the gas supply part of the set-up as the exposure to water occurs at lower temperatures than in an engine, with possibly condensation. Two options for achieving a desired mixture were identified. The first of these would be to compose a stoichiometric mixture in the vessel and combust this. After cooling down, the vessel would be evacuated down to a certain partial pressure which is proportional to the desired molar fraction of diluent. Fresh, stoichiometric mixture would then be admitted to the vessel and mixed with the remaining products from the previous combustion. This option was considered less favorable, since each measurement would require two combustion events. One could expect the overall composition to be less accurate as the reactant quantities are also less for the second combustion. The second option is to substitute the exhaust gases with a different diluent composition which has the same effect on the combustion. Rahim ([69]) and Elia ([21]) use a mixture of 86% nitrogen and 14% carbon dioxide for this. They argue that this ratio simulates the specific heat and molar mass of real exhaust gases, as found in 'a typical internal combustion engine'. Simple calculations show that with this ratio, the specific heat at constant pressure for reference conditions of $p=4.0$ MPa and $T=900K$ (medium engine load) for real exhaust gas is 1298 J/KgK, whereas this mixture gives 1163 J/KgK. The molar mass is higher for the mixture of nitrogen and carbon dioxide, but this quantity is not as sensitive to the carbon dioxide fraction as is the specific heat. Ponnusamy et al ([67]) use a mixture of 81.5% nitrogen and 18.5% carbon dioxide for the same purpose but mention clearly that the specific heat capacity is optimized for his unburned temperature range, which is relatively low (as he also uses a cubical vessel). For the present work, the specific heat capacity was optimized for average SI engine conditions of 4.0 MPa and 900K; this results in a $CO_2$ fraction of 17%. In figure 7.13, the $CO_2$ fraction for equal specific heat is shown. It can be seen that this fraction is fairly constant over the range of pressures and temperatures (>650K) occurring in a SI engine. The maximum unburned temperature in the vessel before the flame hits the wall is somewhat lower. It was however decided to take the engine conditions as reference. This mixture of 83% $N_2$ and 17% $CO_2$ was used in all measurements with diluent.
Burning velocity calculations using Chem1D

To support the choice for using a mixture of only nitrogen and carbon dioxide, the burning velocity was calculated for addition of real exhaust gases as well as for the 'synthetic' EGR. These calculations were performed with Chem1D, which is the one dimensional flame calculation package developed at the TU/e. The burning velocity was calculated over the same pressure-temperature range as displayed in figure 7.4. In figure 7.14, the calculated burning velocity is plotted for both real exhaust gases and 'synthetic EGR' at a diluent mass-fraction of 0.2. Hardly any difference in burning velocity can be detected between both diluents. It was therefore concluded that the substitution of real combustion products including water with a mixture of only nitrogen en carbon dioxide does not change the burning velocity significantly and therefore it is a valid way to measure the burning velocity of EGR diluted mixtures. It seems that the specific heat capacity is the main parameter regarding the effects on combustion.

7.8.2 Realization

Measurements were planned for dilution mass-fractions 10, 20 and 30%, as this is the maximum for application in a SI engine. During the measurements it was found that mixtures with more than 20% diluent could hardly be ignited. This is probably due to the lower temperature at ignition compared to a SI engine. It was decided to change the dilution ratios to 10, 15, and 20% (for $T_i = 400K$, also 18% was used). For the initial pressure and temperature, the same conditions as with the lean mixtures were used. Just as with lean mixtures, the pressure and temperature range for EGR dilution should be as high as possible, therefore only the measurements with an initial temperature of 475 K will be discussed here.

Schlieren recordings were also performed for the EGR measurements. The combustion of EGR diluted mixtures appears roughly similar to the combustion of lean mixtures.
With increasing diluent fraction, less cracks appear on the flame surface. This is most pronounced at low pressures. In figure 7.15, 3 absolute pressures are compared, together with 3 diluent fractions. The images are all taken at a pressure ratio $p/p_i$ of 1.2. At higher pressures, more diluent is needed to reduce the formation of cracks on the flame surface. Similarly to the lean experiments, with the EGR measurements the burning velocity also decreases with increasing diluent fraction and pressure. In figure 7.16, the burning velocity for 10, 15 and 20% diluent is plotted. The temperature range, indicated by the isotherms, is smaller for higher diluent fractions as the maximum pressure is lower. In these and further figures, the diluent fraction is indicated with the symbol ‘D’, as the term ‘EGR’ was found less appropriate. In figure 7.16(c), at the left side of the graph there seems to be a crossing of isotherms. This is caused by the fitting procedure used.

### 7.9 Burning velocity correlations

As mentioned in the introduction of this chapter, the most important objective of these burning velocity measurements was the generation of a burning velocity database for use in the characterization of engine combustion. Ideally, this database should be based on actual measurements of the burning velocity at all temperatures and pressures occurring during the combustion in a SI engine. These temperatures and pressures have been identified in figure 7.4. As became clear during the measurements, the area between the boundary for low and high engine load could not be covered completely. The experimental facilities available do not allow the actual measurement of the burning velocity at very high temperatures. The pressure presents less of a problem, but due to the isentropic coupling of pressure and temperature both should increase to stay within the area of
Figure 7.15: Flame appearance at various initial pressure and diluent fractions, pressure ratio $p/p_i=1.2$
Burning velocity correlations

Figure 7.16: Burning velocity as function of pressure and temperature at various initial pressures
interest. To make the construction of a database possible, extrapolation has to be used. In practice, this means that the effect of pressure, temperature and equivalence ratio or diluent fraction on the burning velocity has to be captured in some way. In previous studies ([4],[43],[33],[41],[21],[75],[69],[64],[58],[49]) in which an attempt was made to represent the effects of pressure and temperature by a correlation, mostly a powerlaw form was chosen. Such a powerlaw has the general form

$$S_L(p, T_u, \phi, D) = S_{L,0}(\phi, D) \cdot \left(\frac{T}{T_0}\right)^\alpha \cdot \left(\frac{p}{p_0}\right)^\beta \tag{7.1}$$

in which the $\alpha$ and $\beta$ represent the influence of temperature and pressure respectively. The values for the reference temperature $T_0$ and reference pressure $p_0$ are arbitrary, but mostly atmospheric conditions (293 K and 1.01325 bar) are used. In many studies, the equivalence ratio is limited around unity and the diluent fraction is also small. In that case, often a separate correlation is used for each equivalence ratio and diluent fraction. Iijama ([43]) did use an equivalence ratio dependent $\alpha$ and $\beta$, but a linear correlation was used and also the lowest equivalence ratio used was only $\phi=0.8$. Also Liao ([49]) used an equivalence-ratio dependent $\alpha$ and $\beta$, but as this work was performed using the constant pressure technique, the temperature range was limited to 300-400K. Also the fuel composition was not identical as natural gas with a methane content of 96% was used. In [64] and [58] a different kind of approach was used. In this work, the dependence of the burning velocity on temperature and pressure was not fitted against measurements, but Peters derives these dependencies from fundamental quantities like the activation energy and inner layer temperature $T^0$. Peters does not compare these results to measurements, only a comparison is made with numerical calculations. Unfortunately, no further attempts to use these fundamental dependencies to derive a correlation were found in literature. It was therefore decided to use the powerlaw form for this work, as more data is available for a direct comparison.

### 7.9.1 Burning velocity correlation for lean mixtures

The values for $S_{L,0}$, $\alpha$ and $\beta$ were varied in a procedure which minimized the mean squared error with the separate measured burning velocity curves. In this procedure, a separate powerlaw was determined for each equivalence ratio. For the determination of these powerlaws, also the measurements with an initial temperature of 400K were used. These were however not performed for all equivalence ratios. As reference temperature $T_0$ and reference pressure $p_0$, 293 K and 101.325 kPa were used. The procedure and a table of the resulting temperature and pressure coefficients can be found in appendix F. To generalize the burning velocity correlation, the pressure and temperature coefficients as well as the base burning velocity $S_{L,0}$ were plotted as function of the equivalence ratio (figures 7.17(a), 7.17(c) and 7.17(e)). The value for $S_{L,0}$ decreases with decreasing equivalence ratio to reach a value of zero at $\phi = 0.5$. Both the temperature and pressure coefficients increase in absolute sense with decreasing equivalence ratio. The temperature coefficient $\alpha$ increases from about 2 at $\phi = 1.0$ to 4.5 at $\phi = 0.55$, the pressure coefficient $\beta$ decreases from -0.35 down to -0.7 at $\phi = 0.55$. This means that at lean mixtures, a change in temperature or pressure has more effect than the same change at stoichiometric conditions. The burning velocity correlation determined this way can be summarized in
equation 7.2.

\[ S_L(p, T_u, \phi) = S_{L,0}(\phi) \cdot \left( \frac{T}{T_0} \right)^{\alpha(\phi)} \cdot \left( \frac{p}{p_0} \right)^{\beta(\phi)} \]  

(7.2)

\[ S_{L,0}(\phi) = 9.302\phi^2 + 35.203\phi - 19.982 \]
\[ \alpha(\phi) = 2.435\phi^2 - 9.427\phi + 9.021 \]
\[ \beta(\phi) = -1.445\phi^2 + 3.086\phi - 1.977 \]
\[ T_0 = 298 \text{ K} \]
\[ p_0 = 101.325 \text{ kPa} \]

7.9.2 Burning velocity correlation for diluted mixtures

For the stoichiometric mixtures diluted with 'synthetic EGR', a similar procedure was used. First, a separate powerlaw was determined for each dilution fraction, in which both measurements conducted with an initial temperature of 400K and 475K were used. To generalize the correlation, the temperature and pressure coefficients and the base burning velocity \( S_{L,0} \) were made dependent on the diluent fraction. These dependencies are plotted in figures 7.17(b), 7.17(d) and 7.17(f). The resulting correlation resembles equation 7.2 and is summarized in equation 7.3. Also for this correlation, more information can be found in appendix F.

\[ S_L(p, T_u, D) = S_{L,0}(D) \cdot \left( \frac{T}{T_0} \right)^{\alpha(D)} \cdot \left( \frac{p}{p_0} \right)^{\beta(D)} \]  

(7.3)

\[ S_{L,0}(D) = 0.0169D^2 - 1.305D + 24.407 \]
\[ \alpha(D) = -1.273 \cdot 10^{-3}D^2 + 0.099D + 2.012 \]
\[ \beta(D) = 6.580 \cdot 10^{-4}D^2 - 0.0254D - 0.343 \]
\[ T_0 = 298 \text{ K} \]
\[ p_0 = 101.325 \text{ kPa} \]

7.10 Overall accuracy

In the previous sections, the measurement of the burning velocity was presented for lean and diluted mixtures. These measurements have been fitted with a correlation to make them useable in for instance engine modeling applications. Due to all the steps involved, it is hard to state an absolute accuracy for the final results. Errors are present in all steps of the process. There are errors in the determination of the composition of the initial mixture. These errors were minimized as described in section 7.5.2, and range from 1\% to 4\% depending on the initial mass and fuel fraction. The measurement of the pressure rise in the vessel has an estimated accuracy of no more than one percent. The two zones model also introduces errors. The assumptions made like a spherical shape for the burned products become less and less valid for leaner or more diluted mixtures. Also the heat transfer becomes a disturbing factor then. Finally, the fitting procedure as described in the previous sections and also in appendix F shows large differences in the quality of the fit for various conditions. One can argue whether the power law assumption was the best choice here. Considering all these error sources, the accuracy of the final result is estimated around 10\%, and possibly somewhat worse for very lean or diluted mixtures.
Figure 7.17: $\alpha$, $\beta$ and $S_{L,0}$ as function of equivalence ratio $\phi$ (while $D=0$) and dilution fraction $D$ (while $\phi=1$)
7.11 Comparing air and EGR diluted combustion

For application in a SI engine, it is known that the dilution level that can be tolerated is different for lean burn and $\lambda = 1 + EGR$ strategies. Just as $D_{EGR}$ is defined as a (mass) replacement fraction with exhaust gases

$$D_{EGR} = \frac{m_{EGR}}{m_{total}},$$  \hspace{1cm} (7.4)

the air excess ratio $\lambda$ can be expressed as a replacement fraction with additional air according to

$$D_{air} = \frac{m_{excess\ air}}{m_{total}} = \frac{\lambda - 1}{\lambda}. \hspace{1cm} (7.5)$$

The maximum dilution level defined this way for excess air is approximately 40%. For dilution with exhaust gases, $D_{EGR}$ is often limited to about 20-25%. Besides differences in ignitability, this may be caused by differences in (turbulent) combustion properties. The overall combustion duration is influenced by the laminar burning velocity as well as the response to the turbulent flow field. It is therefore interesting to compare the laminar burning velocity of mixtures with additional air and mixtures that are diluted with exhaust gases. In figure 7.18(a), a comparison is made between the burning velocity of a mixture diluted with synthetic EGR and with excess air, for a temperature range of 400-600 K and a pressure of 2.0 MPa. The quantity plotted in this figure is the dilution fraction with excess air $D_{air}$ that is required to achieve the same burning velocity as results from a dilution with (synthetic) EGR. The EGR dilution fraction $D_{EGR}$ is on the x-axis of the figure. In figure 7.18(b), a similar plot is displayed for a variation in pressure from 1.0-3.0 MPa. Comparing both figures, it shows that the dilution with excess air needed for a similar burning velocity depends on the temperature of the unburned mixture, but not on the pressure. This may be caused by the temperature dependence of the specific heat capacity for carbon dioxide in EGR diluted mixtures as the reduction of burning velocity is related to the reduction in flame temperature. Other (thermo)chemical effects are of course also possible. In both figures, the lines should theoretically pass through the lower left corner as an air dilution fraction $D_{air}=0$ and an EGR dilution fraction $D_{EGR}=0$ represent the same mixture. This is probably caused by the insensitivity of the laminar burning velocity to dilution at very low dilution ratios. Also, the limited amount of data available for the EGR burning velocity correlation could attribute to this.

7.12 Comparison with literature

A direct comparison of the derived burning velocity correlation with literature data is limited to the equivalence ratio, temperature and pressure range at which these are both valid. The derived correlation is valid in a relatively wide equivalence ratio range of $\phi = 0.55 - 1.0$ and a temperature and pressure range of 480-620 K and 0.4 - 3.5 MPa respectively. Literature data however is mostly limited to the range $\phi = 0.8 - 1.2$ ([43],[21],[75],[69]) or is specified for only a few equivalence ratios ([33]). The few literature sources that are valid in a wide range are either a fit to a collection of various literature sources ([41]) or a relation that was based on theory only ([58]). Only Liao ([49]) derived a similar correlation for $\phi=0.6-1.4$, but this correlation is limited to $T_u=300-400$K. Despite these limitations,
Burning velocity measurements

Figure 7.18: Comparison of dilution with excess air and synthetic EGR, plotted is the 'air dilution' needed for the same burning velocity as with EGR dilution

(a) $D_{air}$ vs. $D_{EGR}$ as function of temperature at fixed pressure
(b) $D_{air}$ vs. $D_{EGR}$ as function of pressure at fixed temperature

7.13 Turbulent combustion experiments

For general interest, a number of burning velocity measurements was carried out under turbulent conditions. With these measurements, the mixing fan (figure 7.8(b)) was not stopped prior to the ignition of the mixture. While the turbulence intensity and spatial scales were not known, it was assumed that these were roughly the same in all experiments performed in this manner. This way, a comparison could be made between a number of equivalence ratios. In figure 7.20, the first stages of combustion for three equivalence ratios is displayed. The frames are taken at equal mass fraction burned; of course the timescales are different. When mixtures get leaner, the combustion takes longer and there is more time for the flame to be influenced by the flow pattern as caused by the fan in the vessel. As is most clearly visible with $\phi = 0.59$, the fan creates a vortex flow in the vessel. The flame gets 'sucked' into the fan and is dispersed in radial direction. The dashed circles indicate the flame radius as calculated by the two-zones model. In this situation this is obviously wrong as the combustion does not take place in the form of a smooth spherical laminar flame. It does show that the volume encapsulated by the outer edge of the turbulent flame is larger than the volume of the theoretical, spherical laminar flame, calculated assuming only burned products behind the flame front. This corresponds with the way turbulent combustion is thought to proceed, in (multiply connected) separate flame sheets. In chapter 3, equation 3.36, the ratio of turbulent and laminar burning velocity was defined as $S_T/S_L = 1 + C(\frac{u'_{rms}}{S_L})^n$. In section 7.9.1, a general burning velocity correlation was presented for lean mixtures. With the use of this correlation and the turbulent burning velocity as calculated from the pressure traces measured during the turbulent combustion experiments, this $S_T/S_L$ ratio can be determined. This ratio is plotted in figure 7.21 for...
Figure 7.19: Comparison with literature references as function of pressure and temperature
Burning velocity measurements

Figure 7.20: Development of turbulent flame during first stages of combustion, $T_i = 475K$, $p_i = 1.2MPa$

(a) $\phi = 1.0, T_i = 475K, p_i = 1.2MPa$

(b) $\phi = 0.71, T_i = 475K, p_i = 1.2MPa$

(c) $\phi = 0.59, T_i = 475K, p_i = 1.2MPa$
equivalence ratios from $\phi = 1.0$ down to $\phi = 0.59$. The sudden bend which is visible in figure 7.21(d) for lower equivalence ratios at approximately $T_u = 600K$ is most likely caused by the combustion acceleration as the flame surface is increased by the radial dispersion by the fan.

### 7.14 Discussion

This chapter has described the measurement of the laminar burning velocity of lean methane-air and methane-air-diluent mixtures at elevated pressure and temperature. The objective of these measurements was the generation of a database (or set of correlations) for the laminar burning velocity which supplies the values for $S_L(p, T_u, \phi)$ for use in the characterization of lean combustion in gas engines. The desired scope of this correlation (e.g. pressure, temperature and equivalence ratio range) was identified as the range of pressure and temperature occurring during the combustion in a gas engine. Unfortunately, it was found that due to limitations of the experimental set-up used, this full range could not be covered and extrapolation has to be used. The maximum temperature of the unburned mixture in a gas engine can be as high as $\pm 800K$, whereas the maximum
temperature attained in the vessel at the moment the flame hits the wall was only ±600K. A similar limitation holds for the pressure. The maximum pressure at the moment the flame hits the walls was 3.5 MPa, whereas this value can reach up to 8-10 MPa (at high load) in a gas engine. Given the acceptable agreement of the measured values with literature values, and the fact that burning velocity measurements at even higher temperature and pressure are scarce, the derived correlations for the burning velocity will be used in the characterization of lean combustion in chapter 8. The use of extrapolation in this case is undesired yet inevitable.
Chapter 8

Characterization of lean engine combustion

8.1 Introduction

In this chapter, the results from the previous chapters will be combined to identify the combustion regime in which combustion in this engine takes place. Starting from the theoretical basics as explained in chapter 3, and using the results from the non-optically accessible version of the engine presented in chapter 5 as well as the expressions for the laminar burning velocity as derived in chapter 7, the ratio of (apparent) turbulent and laminar burning velocity will be determined. Using also the results from the flow measurements (i.e. the magnitude of turbulent velocity fluctuations and the typical spatial scales of the flow), the progress of the combustion process from ignition to the moment that a substantial fraction has been burned will be plotted in a combustion diagram. The remainder of the chapter is devoted to the analysis of the Mie scattering images that were recorded in the same engine configuration as in which the flow measurements were performed. Several aspects of the shape and size of the burned zone will be discussed. Also, for selected conditions, combined Mie scattering/PIV images are presented and some relevant aspects are discussed. The edges of the burned zone were traced and an attempt is made to use fractal analysis to derive a flame surface area from the flame topography. The flame surface area thus derived is subject to a discussion in which the concept of flame surface area ratio is reviewed.

8.2 Burning velocity ratio

As already mentioned in chapter 3, the turbulent burning velocity $S_T$ is generally expressed using a similar concept as is used for laminar combustion. Instead of the real flame surface, the turbulent burning velocity thus found is related to an apparent surface on which combustion takes place. This concept was already presented in equation 3.35, repeated here for clarity.

$$S_T \cdot A_T \cdot \rho_u = \dot{m}_b = S_L \cdot A_L \cdot \rho_u$$

This can also be written as

$$\frac{S_T}{S_L} = \frac{A_L}{A_T}$$  \hspace{1cm} (8.1)
This rather general relation assumes that combustion locally proceeds in a laminar manner and that the enhancement of the overall mass burning rate is due to surface increase only. In doing so, it assumes that the laminar burning velocity does not change and therefore neglects any effects of flame stretch and strain. Several attempts have been made to capture these effects using relations like

\[ \frac{S_T}{S_L} = 1 + C \left( \frac{u'}{S_L} \right)^n. \]  

The measured or calculated values for the turbulent burning velocity are then fitted using the 2 model constants, \( C \) and \( n \).

In reality, the phenomena occurring are more complex. Describing these phenomena using mathematical relations is also not straightforward, at least when one wishes to do this using a single formula. To evaluate relation 8.2 in this context, the ratio of turbulent and laminar burning velocity has been calculated for the combustion occurring in the engine. From the pressure traces that were measured during combustion in the non-optical version of the engine, the mass burning rate \( \dot{m}_b \) was determined using a thermodynamic model (see also appendix C). Using also an assumption for the shape of the flame, an apparent turbulent burning velocity can be calculated from

\[ S_T = \frac{\dot{m}_b}{\rho_u A_T} \]  

For the shape of the flame, a spherical surface was assumed which starts at the spark plug location and propagates outward from there. This apparent turbulent burning velocity \( S_T \) was determined for all engine working points using the average pressure trace of \( O(100) \) engine cycles. The corresponding laminar burning velocity is found from the correlation as presented in chapter 7, equation 7.2 where the input pressure and temperature of the unburned mixture are taken from the engine calculation model output. Using the turbulent and laminar burning velocity, the ratio \( S_T/S_L \) was determined from the ignition up to the point where a substantial part of the fuel mass has burned. These results are shown in figure 8.1. In this figure, the engine speed increases from left to right; the engine load increases from top to bottom.

A few remarks should be made here. The correlation for the laminar burning velocity \( S_L \) as found in chapter 7 was derived from measurements in a constant volume chamber. The pressure and temperature levels occurring in the engine often exceed those of the burning velocity measurements. Extrapolation beyond this range is therefore necessary. This of course will not help the overall accuracy of the process, however it cannot be avoided. Another aspect of the laminar burning velocity correlations is that these were derived for mixtures that were either lean \( (\phi < 1) \) or diluted \( (D > 0) \). Depending on the engine operating conditions, the mixture being burned in the engine contains some residual gases that result from imperfect scavenging in continuous firing mode. Therefore, for the stoichiometric mixtures relation 7.3 was used. Depending on the engine load (and thus manifold pressure since this is a throttled engine) the fraction of residual gases can be about 13% maximum due to the relatively low compression ratio and negative pressure difference over the engine. For lean mixtures the fraction of residual gases is much lower since the engine does not need as much throttling for the same engine load. In this case, a different method was used. Since for these lean mixtures the effect of residual gases is not so much to lower the available oxygen concentration but to absorb
heat from the flame and thus lower the flame temperature. To determine a representative laminar burning velocity the adiabatic flame temperature for a lean mixture containing also a small fraction of residual gases was determined. Then, a mixture equivalence ratio was determined for which the adiabatic flame temperature under the same conditions of pressure and temperature was the same as for the mixture containing also residual gases. The ‘new’ equivalence ratio thus found was used together with relation 7.2 to determine the laminar burning velocity. This method can be criticized since it extrapolates relation 7.2 even further, considering the data available however it was decided that this method was a reasonable option. When examining figure 8.1 with equation 8.2 in mind, some clear trends can be seen. The burning velocity ratio $S_T/S_L$ increases when the engine speed increases. This effect is best visible when focusing on $\lambda$ values around 1.5-1.6. For even leaner mixtures, the calculated laminar burning velocity decreases very fast leading to high $S_T/S_L$ values. Comparing two equivalence ratios that do not differ that much can then still display large differences in $S_T/S_L$. As the pressure and temperature levels do not change much when the engine speed is increased and therefore the laminar burning velocity is roughly the same, the increase in burning velocity ratio can be attributed to an increase of the apparent turbulent burning velocity. For the higher load, a similar trend is
visible. Since the turbulent velocity fluctuations are about 10 times as large as the laminar burning velocity, this indicates that the \( C \) and \( n \) coefficient in equation 8.2 are of the order one. For a constant engine speed, either 1000 or 1350 rpm, the burning velocity ratio can be seen to increase with engine load. This observation is due to the decrease in laminar burning velocity when the cylinder pressure increases. The turbulent burning velocity is roughly constant when the load is increased for the same engine speed. In the figure, the burning velocity data has been plotted for a mass fraction burned ranging from \( x_b = \pm 0.01 \) to \( x_b = 0.70 \).

When looking at the shape of the curves when the engine speed or load is increased, it can be seen that the burning velocity ratio decreases only for low engine speeds when combustion progresses, whereas it increases to a certain maximum before decreasing again for higher engine speeds. This effect is even more pronounced at higher engine loads. For the higher engine speeds, the turbulent burning velocity starts lower and increases to a maximum before decreasing, while it increases less at the lower engine speed. The laminar burning velocity shows a maximum some 10 degrees after TDC. For the higher load cases, the relative change in laminar burning velocity is larger due to the higher pressure levels. The accompanying plots of \( S_T \) and \( S_L \) only can be found in appendix D.

For all engine tests, the intention was to phase the combustion for maximum brake torque. In practice however, since the engine was a single cylinder engine running at only moderate speeds, there was quite some variability in the IMEP per cycle. Therefore, the location of the maximum pressure was kept constant at about 8 degrees ATDC. This may be a bit late, but at least it provides a reference condition for all measurements. Some influence of this phasing may also be present in the burning velocity ratio found.

### 8.3 Combustion diagram

In combustion science, turbulent combustion phenomena are often characterized using a combustion diagram. This was already mentioned in chapter 3. Such a diagram graphically shows the relative (local) importance or influence of the flow phenomena on the combustion process. As already discussed in chapter 3, section 3.3.2 the relative importance of mixing and chemistry can be evaluated either by considering the respective time scales (Re-Da diagram), or by considering the local ratio of length and time scales (Borghi diagram). For engine combustion these kind of diagrams can be constructed once the various length and velocity scales are known. For this case, the integral length scale \( L_i \) is known from PIV measurements while the flame thickness \( \delta_F \) can be estimated from the viscosity and the laminar burning velocity using equation 3.14, repeated here for completeness.

\[
\delta_F = \frac{\nu}{S_L}
\]  

The velocity scales that are applicable are the turbulent velocity fluctuation \( u' \) which is also derived from the PIV measurements and the local laminar burning velocity \( S_L \) which is found using the relations presented in chapter 7. The turbulent velocity fluctuation that is used here is actually the high-frequency component \( u'_{HF} \) as it is called in section 6.3.3.

The variables that are used to locate the type of combustion are not constant but change
during the combustion event in the engine. Due to the evolution of pressure and temperature the laminar burning velocity changes with crank angle and also the length scales and velocity fluctuations change due to the volume change and turbulent dissipation. This makes that the combustion event is plotted as a trajectory rather than a single point. In the following figures, this trajectory is plotted from the ignition up to the point where 80% of the charge has been burned. To further illustrate the various stages of combustion, the ignition angle $\Theta_{\text{spark}}$, the start of detectable heat release (start of combustion, here 1% burnt was used) "Start HR" and the angle of maximum pressure $P_{\text{max}}$ are indicated by a marker. The same holds for the crank angles where respectively 10%, 50% and 80% has been burned.

For stoichiometric combustion as is the case for figure 8.2, the combustion occurs in the corrugated flamelets regime. The major reason that combustion occurs in this regime, is that the laminar burning velocity is high for stoichiometric mixtures and therefore the flame thickness is small. The flamelets are deformed by the turbulent flow field, but the flame sheets are still continuous. The smallest scales in the flow are too large to penetrate the preheat zone and merely push the flame around. When the 2 engine speeds are compared, it is seen that there is only a vertical shift from 600 to 1000 rpm. The engine load is the same, and also the pressure and temperature levels differ little. The laminar burning velocity and flame thickness are therefore also similar. The only difference is the magnitude of the velocity fluctuations, which causes the vertical shift in the diagram. Unfortunately no data is available for an even higher engine speed at this load. The corresponding Re-Da diagram can be found in figure 8.6(a). In this diagram, it is seen that the "combustion trajectories" are above the line labeled "$\eta/\delta_F = 1$", meaning that the flame thickness is smaller than the smallest scales in the flow and the flame structure is not influenced by the smallest eddies. The trajectories are also still close to the "$u'/S_L = 1$" line, indicating that the turbulent velocity fluctuations are not much greater in magnitude than the laminar burning velocity.

In figure 8.3, a similar diagram is shown for a leaner mixture with $\lambda=1.72$ (close to the limit for this engine).

For these leaner mixtures, the combustion regime has shifted to the thin reaction zones regime. Due to the lower laminar burning velocity and therefore larger flame thickness, the smallest structures in the flow are now able to penetrate the preheat zone. This shift to the thin reaction zones regime occurs for all three engine speeds. During the main heat release phase the cases are found close together. On average, the cases for the lower engine speeds are found further into the thin reaction zones regime. For the lower engine speeds, the calculated laminar burning velocity is lower, while the flame thickness is larger. The results for low load however are influenced by the higher fraction of residual gases. This, together with the way the laminar burning velocity correlations for lean and diluted mixtures are combined, leads to large variations in laminar burning velocity when the residual fraction is slightly different. The same information can be found in the corresponding Re-Da diagram in figure 8.6(b). The trajectories are completely below the "$q/\delta_F = 1$" line, again indicating that the smallest eddies penetrate the preheat zone.

This observation indicates that, if the combustion is affected by the velocity fluctuations, this is most likely to occur during the initial and final stages of the combustion of the premixed charge. It is however not likely that this will lead to large scale quenching. In some cases, this may lead to failure to initiate the combustion at all (misfire) or to local quenching in colder areas close to surfaces during the final stages. This however is rather
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Figure 8.2: Borghi diagram for 600+1000rpm, $\lambda = 1$, 4bar IMEP

Figure 8.3: Borghi diagram for 600/1000/1350rpm, $\lambda = 1.72$, 4bar IMEP
**Figure 8.4:** Borghi diagram for 1000rpm, $\lambda = 1, 4$ and 8 bar IMEP

**Figure 8.5:** Borghi diagram for 1000rpm, $\lambda = 1.66 - 1.74, 4, 8$ and 12 bar IMEP
speculative and more detailed research is needed for that.

A comparison of 2 different engine loads with stoichiometric combustion is shown in figure 8.4. This figure shows a small shift to the right, further into the corrugated flamelets regime when the engine load is increased from 4 to 8 bar IMEP. Since the flow scales and velocities do not change much as the engine speed is unchanged, this shift is due to a slight decrease of the laminar flame thickness and a slight increase of the integral length scale. The latter however may be influenced by the combustion phasing. The corresponding Re-Da diagram can be found in figure 8.6(c). A comparison of 3 different engine loads, 4, 8 and 12 bars IMEP with a lean mixture, either at 1000 or 1350rpm engine speed shows both a vertical and a right shift as the engine load increases. In figure 8.5 the case for 1000rpm is shown. The vertical shift is again caused by the decrease in laminar burning velocity as the pressure increases; the right shift is due to the decrease in flame thickness. In the corresponding Re-Da diagram (figure 8.6(d)), the combustion trajectories are completely below the \( \eta/\delta F = 1 \) line, indicating a considerable influencing of the flame structure by the smallest eddies in the flow.

Considering the figures shown, it is clear that the combustion in this engine occurs, at least for stoichiometric mixtures, in the corrugated flamelets regime. For the lean mixtures, the combustion trajectories are found in the thin reaction zones regime, however the main part of the combustion event (fast heat release) is still found near the boundary with the corrugated flamelets regime. If combustion instabilities occur it is likely that these will manifest themselves either at the start or end of the combustion process. Based on the combustion regimes, the occurrence of instabilities is more likely for the lean mixtures. This of course corresponds to what is observed in practical situations.

### 8.4 Flame tomography using Mie scattering

Using a similar methodology as was used for the PIV measurements that were described in chapter 6, also measurements were performed in which Mie scattering was employed to visualize the burned and unburned zones. For these measurements the engine was motored while a fuel/air mixture was used instead of just air. The result of the PIV measurements was a velocity field; for these Mie scattering images the result is a cross section of the 3-dimensional flame structure. The fuel/air mixture was seeded with silicone oil droplets in much the same way as was done for the flow measurements. The seeding density was approximately equal. The seeded fuel/air mixture was then ignited as in normal engine operation. The micron sized silicone oil droplets evaporate and/or burn when the flame front passes them. This way, the burned zone becomes visible as a darker area in the field of view (i.e. no light scattering from oil droplets) whereas in the area around the flame scattering from the droplets is still visible. As the engine was equipped with optical access for these measurements, the corresponding design modifications limited engine cooling to a bare minimum. Therefore, no continuous firing was possible and the engine was skip-fired at a rate of 1 to 7 giving the piston and other surfaces time to cool down in between. A further limitation for these measurements was the fouling of the windows in the engine by the deposition and accumulation of oil droplets. This caused a scattering of the laser sheet that passed through the upper liner and also obscured the view of the camera through the piston window. This fouling was most severe at higher manifold pressures. It is likely that the higher cylinder pressures slow down the evaporation of the
Flame tomography using Mie scattering

(a) 600+1000rpm, $\lambda = 1$, 4bar IMEP
(b) 600/1000/1350rpm, $\lambda = 1.72$, 4bar IMEP
(c) 1000rpm, $\lambda = 1$, 4 and 8 bar IMEP
(d) 1000rpm, $\lambda = 1.66 - 1.74$, 4, 8 and 12 bar IMEP

Figure 8.6: $Re-Da$ diagrams
oil droplets in the non-fired cycles which causes more deposition on the surfaces. The hardware of the PIV system did not allow the laser to be triggered at random. To maintain thermal stability in the laser cavity, the laser flashlamps were flashing at a 10Hz rate (i.e. not synchronized with the engine); the PIV system only accepted triggering within a narrow time window before and after the free-running repetition. These limitations effectively resulted in the following operation mode: The engine was motored with a fuel/air mixture of the desired equivalence ratio. The laser heads flash at a 10Hz rate - but not synchronized with the engine - to maintain their operating temperature. The air supply to the Laskin nozzles is opened and once an acceptable seeding density is observed, the PIV system records images until the windows have fouled too much (depending on the working point, typically 50-100 cycles). Of every 7 cycles, only one is ignited resulting in 14 power cycles per 100 engine cycles. Of these 14 power cycles, only 20% (on average 3 cycles, assuming fully asynchronous operation) are timed in such a way that the trigger that is generated at a desired crank position falls within the acceptance time-window of the PIV system. This means that for every 3 images in which a flame may be recorded the engine has to be stopped, the windows cleaned etc. Simultaneously, the pressure signals of these 100 cycles were recorded by the engine indicating system in such a way that the pressure traces could be matched to the corresponding images afterwards. For selected conditions it was also possible to apply the PIV algorithm on the scattering images. This was the case when conditions were such that the seeding density around the flame was suitable for PIV analysis and both frames of the double image were useable. This may seem trivial, however the shutter opening time was fixed at a longer duration for the second frame which caused this frame to be overexposed for flames that exhibited a high luminance. This was the case for stoichiometric mixtures with a shorter combustion duration. In the images for which application of PIV was possible, some influence of the expanding burned mixture on the surrounding flow structures can be seen. This is shown in section 8.5. The operating conditions for these Mie-scattering series were chosen based on the lean limit encountered in the continuously fired operating mode as this should maximize the chance of observing 'interesting' phenomena. The manifold pressure was set equal to that used for the continuously fired engine operation. As the air flow is a bit different when the engine is skip-fired, the fuel flow was adjusted such that the mixture equivalence ratio based on the measured air and fuel flow rates was the same as for the continuously fired engine operation that was selected as reference. This could not be checked against measured emissions or an oxygen sensor as these could not be used during skip-fired operation. Because there are virtually no residual gases present in skip-fire operation, the 'effective dilution' is a bit lower. For normal engine operation a few percent residual gases are present which dilute the mixture even more. This dilution adds to the air dilution already present in lean mixtures. For stoichiometric mixtures, this effect is even more pronounced as the residual gases are the only dilution present in that case. Also the fraction of residual gases is higher for stoichiometric mixtures because - for low load - the intake is throttled more in that case. For the same reason, the resulting IMEP of these skip-fired cycles is about 5-15% higher than for continuously fired operation at the same equivalence ratio and manifold pressure. This difference was noted but it was not adjusted further. A further important difference that prevents a one-to-one comparison with the continuously fired operation is that the compression ratio was at some stage lowered for the
optical engine configuration. The compression ratio was lowered from 11.25 to 9.5 by trimming the edge of the piston crown by 5mm. This enabled the application of a laser sheet throughout the entire engine cycle without the piston blocking the path of the laser sheet. Without this modification, the application of the laser sheet was not possible between 340 and 380 °CA, preventing the observation of the flame structure around TDC for normal combustion timing. Images have been recorded before and after this modification. For the images recorded after the modification, the flow field is somewhat different from the flow field as measured in the unmodified configuration. In this section, a selection of images that were recorded in the modified configuration will be presented as this allows for a more elaborate display of the flame evolution.

In figures 8.7(a) through 8.7(f), a number of Mie scattering images for 600rpm, λ=1 is shown. The burned zone is visible as the darker area at the center of the images. These six images have not been recorded in succession, between each two fired cycles are 6 motored cycles. Assuming a stable (average) operating temperature, this should provide equal starting conditions for each fired cycle. In figure 8.7 it can be seen that for stoichiometric combustion the development of the turbulent flame shows a low variability. The flame has similar size and is always centered around the spark plug. This indicates that the flow field has no time to convect the flame away from the spark plug. The inflamed volume is not spherical as is often assumed in modeling, but still propagates at about the same speed in all directions. What also can be seen is that there are no pockets of unburned mixture behind the flame front, nor are there pockets or large peninsulas of burned mixture found in the fresh charge ahead of the flame front.
Figure 8.8: 600rpm, 4 bar IMEP, $\lambda=1.72$, ignition: 318°CA, images: 346.4°CA, CR=9.5

In figures 8.8(a) through 8.8(f), a similar set is displayed for lean engine operation, $\lambda=1.72$, at the same engine speed and load. For this series, one can notice that the flame kernel is convected away from the spark plug. The direction in which these flame kernels are convected is mostly upward in the image, as this is the direction of the mean flow. When looking at the edge of the inflamed volume, there appear to be finer structures compared to the stoichiometric case. On a scale of $O$(mm), several small peninsulas of unburned mixture can be seen on the inside of the turbulent flame front (flame envelope). Also in image 8.8(d) and 8.8(e) isolated islands of unburned mixture can be seen inside the burned volume. Whether these islands are truly isolated remains unclear as the image is a 2D-slice of a 3D-structure.

The development of the flame kernel at a higher engine speed under lean conditions is illustrated in figures 8.9, 8.10 and 8.11, for an engine speed of 1350 rpm and 4 bar (nominal) engine load.

In figure 8.9, it can be seen that the initial flame kernel can fully detach from the spark plug during the initial development. The time scales involved illustrate this well: in figure 8.7 ($\lambda = 1$), the time needed to develop a flame kernel of approximately 40mm takes only 11°CA. The time interval in figure 8.9 ($\lambda = 1.72$) is already 33°CA, and still hardly any flame kernel is visible. Figure 8.10 shows the development of the flame kernel some 10°CA later. Here, it becomes apparent that there is a high variability in flame kernel development. Whereas in figure 8.10(b), the flame has consumed about 20% of the visible part of the combustion chamber, in figure 8.10(c) which was recorded at the
Figure 8.9: 1350rpm, 4 bar IMEP, λ=1.72, ignition: 310°CA, images: 343.2°CA, CR=9.5

Figure 8.10: 1350rpm, 4 bar IMEP, λ=1.72, ignition: 310°CA, images: 353.2°CA, CR=9.5

Figure 8.11: 1350rpm, 4 bar IMEP, λ=1.72, ignition: 310°CA, images: 363.2°CA, CR=9.5
same crank angle there is only a small kernel that has completely detached from the spark plug. In figure 8.11, some $53^\circ$CA have passed between ignition and the instant the image was taken. The flame has consumed a larger part of the combustion chamber, but still the mass fraction burned is no more than about 20-30%. There are pockets of unburned mixture visible in the burned zone, as well as pockets of burned mixture in the unburned zone in front of the turbulent flame. Also from these images it is not clear whether these pockets are connected or isolated. Another important observation from these images is that, if the flame kernel detaches from the spark plug shortly after ignition (i.e. carried away by the flow), the initial part of the combustion may as well take place close to the walls of the combustion chamber. It is therefore quite possible that the heat transfer to the walls in the initial stages of the combustion process is higher than assumed in some heat release models. For the calculation of the heat transfer it is normally assumed that the hot burned products contact the walls only at later stage when the flame is more developed. These images show clearly that this assumption may not be justified.

At a higher engine load, it proved much more difficult to obtain clear images. One cause for this was the much quicker fouling of the piston window, which after a short time caused images as if the camera was imaging through a layer of droplets. The other problem was the less distinctive edges of the flames. One assumption is that the evaporation of the oil droplets is much harder under the higher pressures associated with the higher engine load. These droplets evaporate, but take more time to do so. The area some distance behind the flame front is dark again, but in the finer flame structures the edges of the flames are not visible very well. Also, there is no information whether the droplets that were generated at a higher pressure using the Laskin nozzle seeding generator have the same size as under atmospheric pressure. This was not tested within the scope of these experiments. However, other research ([45]) has shown that the generation of droplets under even higher pressures using a similar Laskin nozzle with olive oil yielded droplets of a similar size. The images taken at a higher engine load show more or less the same phenomena as those recorded for a load of 4 bars IMEP. Phenomena like convection of the initial flame kernel and the occurrence of pockets of burned and unburned mixture were observed. The variability between cycles was larger at a higher engine load. This is likely caused by the lower initial laminar burning velocity at higher pressures which makes the initial stages of flame kernel formation even more difficult and time consuming. Also the number of complete misfires and partial misfires increased much for the higher engine load. This was initially not expected since the effective dilution is lower than with the same equivalence ratio in continuously fired operation. The temperature at the time of ignition however may be lower since the compression ratio was reduced and the engine as a whole runs at a lower temperature level in skip-fire operation. Also the deposition of oil droplets on the spark plug may have had an influence. In figure 8.12 3 sets of 2 images for an engine load of 8 bar IMEP and a lean mixture are shown. The trigger timing for these series was $10^\circ$CA apart. In figure 8.12(a) and 8.12(d), $44^\circ$CA have passed since ignition and still there is only a small flame visible. In figure 8.12(a), this flame seems to be split in 2 parts that are some 40mm apart. In figure 8.12(d) there are also two small fragments of flame visible, but far from the spark plug. Again here, these flame fragments may be connected outside the imaging plane. In both figure 8.12(b)/8.12(e) and 8.12(c)/8.12(f), taken $10^\circ$CA apart (but not in the same engine cycle) pockets and peninsulas of unburned mixture are visible in the burned zone and islands of burned products are found in the fresh mixture ahead of the flame.
8.5 Flame-Flow interaction

As was already mentioned in the previous section, it was possible to perform PIV analysis on selected Mie scattering images from which the flame topography was analyzed. The seeding density applied was suitable for PIV analysis. For higher manifold pressures however, similar restrictions were encountered as with the PIV-only measurements with high manifold pressures. Fouling of the windows and unsharp images yielded velocity fields with a high fraction of rejected vectors. Also for the case of a stoichiometric mixture with normal combustion phasing, the luminescence of the flame caused the second frame to be overexposed and PIV analysis was not possible for these cases (though for stoichiometric combustion with retarded phasing the overexposure was less). For a lot of cases however velocity fields could be determined. For the flame topography analysis, the focus was on the leaner or more diluted mixtures. With these PIV measurements some more differences between the stoichiometric and lean cases become apparent. In this section, also images will be shown that were recorded when the compression ratio was not lowered yet. These images recorded with the unmodified piston will be shown first. For this situation with the unmodified piston, no laser sheet application was possible between 340 and 380 °CA. For that reason, the ignition timing was sometimes advanced or retarded to phase the combustion such that the flame development stage was either before 340°C or after 380°C. For the stoichiometric cases, the only choice here is to retard the ignition to prevent the maximum pressure from occurring too far before TDC. The ignition timing in that case was shifted from 20°C BTDC to 5-20°C ATDC. The cylinder volume in that case is already increasing during flame development, but the pressure and temperature
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Figure 8.13: $\lambda = 1$, 4 bar IMEP, 1350rpm, CR=11.25

(a) ign: 375°CA, img: 383.2°CA

(b) ign: 370°CA, img: 383.2°CA

Figure 8.14: $\lambda = 1.72$, 4 bar IMEP, 1350rpm, late ignition, CR=11.25

(a) ign: 345°CA, img: 383.2°CA

(b) ign: 345°CA, img: 383.2°CA
levels at the time of ignition are alike those at normal timing. As the lean mixtures already have a much longer flame development duration, the ignition timing for these cases was retarded, but still before TDC. The normal ignition timing of ±50°CA BTDC was then retarded to 30-15°CA BTDC. In this case the temperature at ignition will be higher and ignition should be less difficult. Two examples for a stoichiometric mixture, ignited after TDC are shown in figure 8.13.

In these figures it can be seen that there is a strong motion outward. This motion is partly caused by the flow into the increasing volume of the squish zone as the piston travels downward. The velocity vectors also have a tangential component, indicating that the swirl rotation of the charge has not decayed fully yet. Close to the flame front, the velocity vectors are directed almost completely in radial direction indicating a rapid expansion of the flame kernel. For stoichiometric combustion, convection of the flame kernel is not an issue. Instead, the expansion of the flame kernel takes place on a time scale smaller than the motion of the large vortices. For lean mixtures however, the combustion process is much slower. The expansion of the burned zone (or burned zones) is therefore of less influence on the flow field. In figure 8.14 that was recorded for a retarded ignition, it can be seen that the flame kernel has several small and large peninsulas. Due to this fragmented flame, detection of the flame contour was not possible. On the (right) edge of the figure the effect of the flow into the squish zone can be seen, similar to figure 8.13. In the center area however (most visible in figure 8.14(b)), the flow field is irregular and contains several smaller structures that indicate the local effects of the flame front on the flow field and vice versa. Because of this fragmented flame it was not possible to outline the flame contour as was done in figures 8.13 and 8.15. In figure 8.15, the same mixture is shown for similar engine speed and load, but now for unchanged ignition timing. At the (right) edge of the figure, the large scale flow can again be seen. Since the piston in this case is moving upward, the flow direction is from the squish zone to the center of the piston bowl. The flame kernel is still in a less developed stage and it can very well be seen that this flame kernel as a whole is convected by the flow.

Next, some images that were recorded with the modified piston will be shown. In these experiments, the mixture equivalence ratio was kept the same as with the unmodified piston. As a result of the changed squish height and lower compression ratio the flow field is slightly different and also the pressure and temperature levels are a bit lower. In figure 8.16, a similar condition as in figure 8.15 is shown. The ignition timing was 5 °CA later which shows as a less developed flame kernel. The crank angle the image was recorded however is the same. When examining the flow field in figure 8.16, an inward flow can still be detected. This flow is less prominent than in figure 8.15 which can be explained through the increased squish height with a lower compression ratio. Apart from this lower squish flow, the overall velocity magnitude is similar.

8.6 Fractal analysis of flame fronts

In section 8.2, equation 8.1 shows that the dominating factor in the determination of the turbulent burning velocity \( S_T \) is the surface increase due to flame wrinkling. It would therefore be interesting to see if it is possible to determine this surface ratio \( A_L/A_T \) experimentally. The turbulent flame surface \( A_T \) that was assumed in the heat release
Figure 8.15: $\lambda = 1.72$, 4 bar IMEP, 1350rpm, early ignition, CR=11.25

(a) ign: 305°CA, img: 343.2°CA  
(b) ign: 305°CA, img: 343.2°CA

Figure 8.16: $\lambda = 1.72$, 4 bar IMEP, 1350rpm, early ignition, CR=9.5

(a) ign: 310°CA, img: 343.2°CA  
(b) ign: 310°CA, img: 343.2°CA
calculations was a spherical surface. From the images shown in the previous section it already became clear that this is often not very realistic. The laminar flame surface \( A_L \) may be determined from the flame images as presented in the previous section. This however is not straightforward. The flame front is irregular and it is deformed by flow structures of different scale and magnitude. In principle, this deformation of the flame surface will therefore occur on scales that range from the integral length scale of the flow \( L_i \) down to scales that are of the order of the flame thickness \( \delta_F \). The images of the flames however can only contain information above a minimum length scale that is determined by the pixel size and imaging optics. One other issue here is that the images present a 2D slice of the flame structure; therefore a flame 'length' can be determined only within a two dimensional plane.

A possible approach to this problem is the use of fractal theory. This fractal theory was first presented by Mandelbrot ([52]). The first proposal to use this theory for the description of turbulent flames is generally attributed to Gouldin ([29]). It was rather popular in the late eighties and early nineties and has since been used for the description of burner flames ([15, 12, 76, 77, 78, 34, 59, 35]) and also flames in engines ([27, 54, 55, 85, 53, 37]). More recent applications have also been found.

8.6.1 Fractal theory

When a curve or shape is said to have fractal properties then this implies that there is a self-similarity in its shape. This self-similarity is present throughout a range of scales. When observing the object on ever smaller scales similar structures are found. The size or dimension of an object can then be determined with the help of a so called 'fractal dimension' and a set of bounding scales between which this fractal dimension is valid. The fractal dimension of a wrinkled line (or flame front) is determined, effectively, by measuring the length of the line with rulers of different lengths. If the wrinkled line is self-similar over a range of scales then a logarithmic plot of the length of the line as measured using each ruler versus the ruler length yields a straight line. The fractal dimension of the wrinkled line, \( D_2 \), can be determined from the slope of this line. This fractal dimension quantifies the wrinkling, the greater \( D_2 \) the greater the intensity of the wrinkling. For a curve in two dimensional space like the contour of the flame in the presented images the fractal dimension \( D_2 \) is between one and two. For a three dimensional surface the fractal dimension \( D_3 \) is between two and three. It can be shown ([52]) for an isotropically wrinkled surface that the fractal dimension \( D_3 \) is equal to the fractal dimension of a 2-dimensional slice \( D_2 \) plus one (i.e. \( D_3 = D_2 + 1 \)). This assumption of an isotropically wrinkled flame surface will finally be used to relate the wrinkling of the contour in a 2D slice to the surface increase needed for the turbulent burning velocity. This assumption is supported to a certain degree by the work of Mantzaras ([53]) who used four parallel laser sheets to assess the homogeneity and isotropy of the fractal dimension of the border line between burned and unburned mixture in an engine. Because this remains an assumption, also the fractal dimension \( D_2 \) of the 2-dimensional slices was compared to results from other researchers.

For very small scales (ruler lengths), the plot of the curve length versus the ruler length will cease to be a straight line. This scale is called the inner cutoff \( \epsilon_i \) and for flames this represents the smallest wrinkling scale. Also for longer rulers at some point the relation
will not be a straight line anymore. This is called the outer cutoff \( \epsilon_o \) and represents the largest wrinkling scale. The turbulent and laminar burning velocities \( S_T \) and \( S_L \) are then related by [29]:

\[
\frac{S_T}{S_L} = \left( \frac{\epsilon_o}{\epsilon_i} \right)^{D_3-2}
\]  

(8.5)

8.6.2 Practical implementation

The fractal dimension \( D_2 \) was determined from the flame images as presented in the previous section. The image quality was not optimal since the experiment was designed for PIV analysis and fractal analysis was performed in hindsight. One aspect of the images that complicated the analysis was that the edges of the flames (i.e. the separation between unburned mixture and burned products) were not always very sharply defined. The droplets used for seeding the flow should ideally have been replaced by something smaller. In that case however PIV analysis would probably not have been possible as no distinct particles can be discerned. The presence of the droplets together with a non-uniform laser sheet causes local intensity gradients, which make it difficult to set a single threshold intensity for detecting the boundary. An example of this is shown in figure 8.17(a). This image is already relative to the background, therefore the burned products show up as a lighter instead of a darker zone. For a sharply defined image with a homogeneous light distribution an intensity histogram may be used for selecting the appropriate threshold. Ideally such an intensity histogram would show a bimodal distribution and the threshold can be found between the histogram peaks ([76]). An intensity histogram for these images however did not show distinct peaks for the burned and unburned zone. This is shown in figure 8.17(b) for the same image. This largely prevented automated image processing. Instead, the images were processed by hand by first subtracting the image from an appropriate background image. This background image could either be an average from a set of images or an image recorded without a flame. Both methods are problematic; the first only works if the flame is not found in the same place in every image, the second suffers from unrepresentative background images due to variations in laser intensity distribution and also seeding density and -distribution. The next step was a slight filtering operation such that the local intensity variations were smeared out without washing away also the finest structures on the boundary. A two dimensional median filter was used for this step. The image was then interactively enhanced by increasing the contrast. After that it was converted to a binary image using an interactively set threshold intensity. In this step, the original image was used as comparison. Once the binary image is generated, edge detection is used to generate a two dimensional curve on which the fractal analysis can be applied. The final result for the same image is shown in figure 8.17(c), figure 8.17(d) is a closeup of the region above the middle of the image.

For the determination of the fractal dimension several algorithms have been developed in the past. A short but useful overview of these is given by Hall ([37]). Three of these algorithms have been used on the flame images. These were the box counting method, the area-caliper method and the caliper method. The box counting method is computationally the least demanding. This method simply counts the number of boxes of a given size needed to cover the curve. The area-caliper method takes the most time. With this method a ribbon is drawn by including all points (pixels) that lie within a certain distance of the curve. Dividing the ribbon area by its width then results in a computed length.
Figure 8.17: Flame contour detection process
The caliper method finally measures the curve using a caliper of fixed size by stepping along the curve whereby each next point is determined by the crossing of a circle with radius caliper size and originating in the current point. The curve is thus measured using straight segments with the caliper size as length. A general observation for all results is that the resulting plots of the fractal dimension do not show a lower cut-off scale. The curves show a good linear fit which does not deviate at the lower end of the scale. For turbulent flames in the regime as was found for these engine flames, the inner cut-off scale is of the order of the flame thickness. Strictly speaking, for stoichiometric mixtures the flow scales are limiting as the flame thickness is smaller. This inner cut-off scale however is of the order $O(0.1\text{mm})$ and often even smaller, which is equal to or smaller than the pixel size. Scales smaller than the pixel size cannot be seen, the inner cut-off scale $\epsilon_i$ may therefore well be obscured. The outer cut-off scale $\epsilon_o$ is expected to be of the order of the integral length scale $L_i$. The total size of the flames observed in most images however is only about $20-30\text{ mm}$. This is only twice the integral length scale, so the outer cut-off scale may also be beyond the range considered for determining the fractal dimension. For larger flames the outer cutoff scale can be found. This is shown in figure 8.18(a). Here, it can be seen that the (logarithmic) plot of $L(\epsilon)$ vs. $\epsilon$ becomes a horizontal line for a 'ruler size' of $\exp(4.5)$ pixels which corresponds to about $9\text{ mm}$. The corresponding flame contour is indicated in figure 8.18(b). For most conditions, about 2-8 images were available for analysis (actual number of images is indicated in column 'img' of table 8.1, 8.2 and 8.3). For this low number, a representative average is not expected, however the calculated fractal dimensions do provide useful information. The calculated fractal dimension $D_2$ for stoichiometric combustion for all three methods has been plotted in table 8.1. The corresponding values of $D_2$ for lean combustion and for an IMEP of 4 and 8 bars can be found in table 8.2 and 8.3. The methods have been plotted in the table header; the area-caliper method has been abbreviated to 'A-C'. The experimental conditions have also been included in the table header.

When looking at the values found for the fractal dimension $D_2$ in tables 8.1 to 8.3, one can see that there is a difference in the values found using the boxcounting method, the area-caliper method and the caliper method. The boxcounting method gives the lowest
### Fractal analysis of flame fronts

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Table 8.1: Fractal dimension $D_2$ for stoichiometric combustion, 4 bar IMEP. See text for further explanation.

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Table 8.2: Fractal dimension $D_2$ for lean combustion, 4 bar IMEP. See text for further explanation.

### 1000rpm, $\lambda = 1.74$, ign: 308°CA

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Table 8.3: Fractal dimension $D_2$ for lean combustion, 8 bar IMEP. See text for further explanation.
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values; the caliper method gives both very high and very low values. These methods also display the most variation. The area-caliper method gives the most consistent values. A similar conclusion was also made by Hall ([37]). The boxcounting method suffers from a lack of resolution because only box sizes of $2^n$ pixels were considered. The caliper method suffers from uncertainties that are inherent to the method as there is almost never a perfect fit, i.e. after the last caliper fit a part of the curve smaller than the caliper size remains. This leads to uncertainties in the linear fit that is used to derive the fractal dimension. For this evaluation therefore, only the values found using the area-caliper method will be considered further. The values for stoichiometric combustion (table 8.1) show little variation between engine speeds of 600 and 1000 rpm. When looking at the results for lean combustion at the same engine load and speed (table 8.2), values are found that are on average a bit higher. These values however also show a considerable spread due to the small number of images so care should be taken when drawing conclusions. For an engine speed of 1350 rpm, approximately the same values are found. For a higher engine load of 8 bars IMEP (table 8.3), the values found seem a bit smaller on average, but again there is considerable spread. The comparison of different conditions therefore does not show a strong correlation with engine speed or load, only for stoichiometric conditions smaller values are found. The fractal dimensions found however are very much alike those found by other researchers ([37, 53, 54, 27]). This finding even holds when these values are compared to values found in turbulent burner flames ([15]).

8.6.3 Burning velocity ratio determination

What remains is the question whether these fractal dimensions and scales can be used to derive the increase in flame surface area needed for the turbulent burning velocities as calculated in section 8.2. In equation 8.5, the relation between the burning velocity ratio, the fractal dimension and the inner and outer cutoff scale was shown. Since the fractal dimension $D_2$ was determined with 1.25-1.30, using this value with the appropriate cutoff scales should result in a burning velocity ratio of the magnitude as shown in figure 8.1. For flames the choice of the inner and outer cutoff scales is subject to some discussion ([34, 63]). For the outer cutoff scale mostly the integral length scale of the flow is used. It was already argued that for these images the outer cutoff may not always be visible as the flame size is of the order of the integral length scale. A slightly different option would be to assume that only scales up to the size of the flame are able to wrinkle the flame (developing or effective turbulence concept, see also [81, 82]). For flames with a size significantly less than the integral length scale however, the determination of a consistent fractal dimension is difficult and these small flames were not investigated in this respect. It was therefore chosen to use the integral length scale as outer cut-off scale. For the inner cutoff scale a more elaborated discussion is found in literature. For non-reacting turbulent flows the smallest scale occurring would be determined by viscous dissipation and the Kolmogorov scale $\eta$ would be the appropriate scale to use. For reacting flows as in premixed turbulent combustion the properties of the flame surface are to be considered. Depending on the flame thickness, laminar burning velocity and turbulent velocity fluctuations, the smallest turbulent scales may or may not be able to penetrate the flame front. For these conditions, on average, the Kolmogorov scale and the flame thickness are of comparable magnitude. The Kolmogorov scale is about 0.5-2 times the laminar flame thickness. This would be an
argument to use the Kolmogorov scale as inner cutoff for use with the fractal dimension. Peters ([66, 63]) however states that the Gibson scale is more appropriate here. For an eddy of the Gibson scale, the turnover velocity is equal to the laminar burning velocity. Eddies of a larger size can push the flame front around, causing wrinkling; smaller eddies are not able to wrinkle the flame front. The Gibson scale is found using

\[ L_G = L_i \left( \frac{S_L}{u'} \right)^3 \]  

(8.6)

The definition of the Gibson scale however is only meaningful in the corrugated flamelets regime, since for the transition to the thin reaction zones regime the laminar burning velocity \( S_L \) is equal to the velocity at Kolmogorov scale and the flame thickness \( \delta_F \) equals the Kolmogorov scale. Above this line, in the thin reaction zones regime, Gibson scales calculated using equation 8.6 would be smaller than the Kolmogorov scale which is obviously not meaningful. A substantial part of the images has been recorded during the initial flame development stages. In figures 8.3 and 8.5 it was seen that the appropriate combustion regime for these lean mixtures is the thin reaction zones regime. Even for stoichiometric combustion as was seen in figures 8.2 and 8.4, the very early stages of combustion are found in this regime. For this reason, when applicable (i.e. in the corrugated flamelets regime) the Gibson scale will be used, otherwise the Kolmogorov scale will be used for the inner cutoff scale in the burning velocity ratio calculations.

The comparison of the \( S_T/S_L \) values as calculated from the heat release and the surface ratio \( A_L/A_T \) from the fractal analysis will be performed on a per-image basis. The fractal dimensions presented in tables 8.1 to 8.3 are averages from a number of images (the number of images used is shown in the column ’img’). The different realizations however show some spread; it is therefore not representative to use the average fractal dimension for that combination of crank angle, engine load and equivalence ratio. Instead the fractal dimension as calculated for the individual images was used. The graphs shown in figure 8.1 for the non-optical version of the engine show a maximum for more developed combustion where a substantial fraction of the mixture has been burned. Due to the limited optical access, most of the flame images have been recorded during the initial stages of flame development. Due to deficiencies in the modeling, the heat release in the initial stages of combustion (\( x_b < 1\% \)) and therefore also the calculated turbulent burning velocity \( S_T \) and velocity ratio \( S_T/S_L \) are less accurate and these images will therefore not be considered for this analysis.

Table 8.4 shows the burning velocity ratio as determined from the fractal analysis for stoichiometric combustion for an engine speed of 600 and 1000 rpm. The mass fraction burned is no more than about 15%. For higher mass fraction burned the flame size is too large for the limited optical access. For the different realizations, the fractal dimension \( D_2 \) shows a considerable spread. The burning velocity ratio from the heat release analysis is quite consistent however. For these conditions the Gibson scale \( L_G \) is larger than the Kolmogorov scale \( \eta \) and this Gibson scale is therefore the scale to use as inner cutoff. The burning velocity ratio determined using equation 8.5 is a factor 3-4 smaller than the ratio found from the heat release. Some of this difference can be explained by the assumptions that were made in the analysis presented in this chapter. The surface that is used to calculate the turbulent burning velocity from the pressure profiles is half a sphere. In figure 8.7 to 8.12 it can be seen that this assumption, despite accepted in these kind of analysis, is not justified. The turbulent flame envelope is irregular in shape.
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<td>1.263</td>
<td>7.2</td>
<td>0.028</td>
<td>0.85</td>
</tr>
<tr>
<td>4</td>
<td>11.4</td>
<td>1.305</td>
<td>6.1</td>
<td>0.027</td>
<td>0.92</td>
</tr>
</tbody>
</table>

| 1000rpm     |                                  |                                  |                                   |                                   |                                   |
| img         | \(x_b\) [\%] | \(D_2\) [-] | \(S_T/S_L\) | \(\eta\) | \(L_G\) | \(S_T/S_L\) Fractal | \(\chi\) [%] |
| 1           | 15.5             | 1.247           | 7.4          | 0.016    | 0.16     | 2.8                 | 2.09            |
| 2           | 14.4             | 1.232           | 7.8          | 0.016    | 0.16     | 2.7                 | 1.71            |

**Table 8.4:** Calculated burning velocity ratio \(S_T/S_L\) for stoichiometric combustion, 4 bar IMEP

| 600rpm      |                                  |                                  |                                   |                                   |                                   |
| Img         | \(x_b\) [\%] | \(D_2\) [-] | \(S_T/S_L\) | \(\eta\) | \(L_G\) | \(S_T/S_L\) Fractal | \(\chi\) [%] |
| 1           | 10.3             | 1.218           | 15.8         | 0.026    | 0.055    | 3.2                 | 1.81            |
| 2           | 9.7              | 1.243           | 13.3         | 0.026    | 0.059    | 3.6                 | 2.03            |
| 3           | 10.6             | 1.301           | 12.2         | 0.024    | 0.081    | 4.4                 | 2.68            |
| 4           | 10.6             | 1.213           | 9.6          | 0.024    | 0.082    | 2.8                 | 1.68            |

| 1000rpm     |                                  |                                  |                                   |                                   |                                   |
| Img         | \(x_b\) [\%] | \(D_2\) [-] | \(S_T/S_L\) | \(\eta\) | \(L_G\) | \(S_T/S_L\) Fractal | \(\chi\) [%] |
| 1           | 8.0              | 1.282           | 13.9         | 0.016    | 0.015    | 6.3                 | 3.29            |

| 1350rpm     |                                  |                                  |                                   |                                   |                                   |
| Img         | \(x_b\) [\%] | \(D_2\) [-] | \(S_T/S_L\) | \(\eta\) | \(L_G\) | \(S_T/S_L\) Fractal | \(\chi\) [%] |
| 1           | 6.9              | 1.267           | 11.7         | 0.014    | 0.005    | 5.9                 | 2.10            |

**Table 8.5:** Calculated burning velocity ratio \(S_T/S_L\) for lean combustion, 4 bar IMEP

| 1000rpm     |                                  |                                  |                                   |                                   |                                   |
| Img         | \(x_b\) [\%] | \(D_2\) [-] | \(S_T/S_L\) | \(\eta\) | \(L_G\) | \(S_T/S_L\) Fractal | \(\chi\) [%] |
| 1           | 5.0              | 1.296           | 23.9         | 0.011    | 0.003    | 7.7                 | 3.04            |
| 2           | 5.7              | 1.257           | 26.6         | 0.011    | 0.003    | 5.9                 | 1.94            |
| 3           | 2.7              | 1.317           | 26.4         | 0.011    | 0.003    | 8.9                 | 2.69            |
| 4           | 3.8              | 1.250           | 34.0         | 0.011    | 0.003    | 5.6                 | 1.93            |
| 5           | 5.7              | 1.242           | 27.5         | 0.011    | 0.003    | 5.4                 | 2.66            |

**Table 8.6:** Calculated burning velocity ratio \(S_T/S_L\) for lean combustion, 8 bar IMEP
This is illustrated in figure 8.19. In this figure, the length of the flame circumference as found from the image analysis was 958 mm (red line in figure 8.19). Filtering this curve with the integral length scale - in this case 11 mm - reduces this to 441 mm (yellow line in figure 8.19). This filtered circumference of the burned zone can be seen as the true turbulent flame front envelope. When the surface area of the burned zone cross-section is converted to a circle with the same surface, then the circumference of this circle is only 136 mm. This means that the surface of the turbulent flame front is underestimated and the turbulent burning velocity as calculated from the heat release assuming a half sphere is thus too large. This ratio of the length of the filtered contour and the circumference of an equivalent circle is plotted in the last column of the tables, named $\chi$. One could use this information to derive a correction factor. For this however, the ratio of the 3-dimensional surfaces of the true turbulent flame front and an equivalent half-sphere has to be known. These images only show a cross section of the flame and provide no information on its 3-dimensional shape. This, together with the observation that the flame cross section is sometimes divided into multiple 'islands', does not allow for the determination of a correction factor of any accuracy. The numbers are therefore presented 'as is' and the difference between the burning velocity ratio from the heat release and the fractal analysis should be attributed to differences in true flame surface. The burning velocity ratios correspond in order of magnitude, but the difference is a factor two to five. In tables 8.5 and 8.6 the corresponding numbers are shown for lean combustion, for an engine load of 4 and 8 bars IMEP respectively. These numbers on average are somewhat larger than the corresponding numbers for stoichiometric combustion. Again here care should be taken as the amount of data is limited.

8.7 Discussion

In this chapter, the results of the heat release analysis, flow field measurements and flame visualizations have been brought together in an attempt to characterize the turbulent com-
bustion process in this engine. In chapter 3, diagrams were presented in which the combustion regime was determined from the applicable length and velocity scales. These length and velocity scales have been determined experimentally or, when this was not possible, estimated. Plotting the combustion process in these combustion diagrams showed that the combustion of stoichiometric mixtures occurred in the corrugated flamelets regime, only the start of the combustion process was found in the thin reaction zones regime. For lean mixtures the full combustion process is found in the thin reaction zones regime.

The corresponding Re-Da diagrams show the same information; the lean combustion is found much ‘lower’ in the diagram where the mixing time scales are relatively short and the turbulent mixing process is affecting the process of flame propagation. Especially the start and end of the combustion process for lean mixtures is affected by turbulence. This may be interpreted as likelihood of failure to successfully initiate or complete the combustion process. In other words, there is a greater probability of misfires or partial burns which is also observed in engines running close to the dilution limit.

Mie scattering images of the burned zone showed that the degree of wrinkling or deformation is also much larger for lean mixtures. The eddies in the flow have more time to deform the flame and also their velocity is larger relative to the propagation speed of the flame front. The Mie scattering images showed that stoichiometric flames are more likely to behave as it is assumed in traditional heat release models; the burned zone expands more or less as a sphere propagating from the spark plug. For lean mixtures the time needed to form a self sustaining flame kernel is much longer and it was often seen that the initial flame kernel is convected away from the spark plug. When comparing a number of images recorded at the same crank position and for the same conditions, substantial differences in flame development are found. Flames are sometimes continuous, but often separate islands of products are seen in the unburned mixture ahead of the flame front and also pockets of unburned mixture are seen in the burned zone. Whether these pockets and islands are truly isolated is unknown since the Mie scattering images show a two-dimensional slice of a three-dimensional flame structure. This observation is however illustrative of the degree of wrinkling where different ‘flame sheets’ interact as is typical for the thin reaction zones regime. The fractal analysis has shown that the fractal dimensions found are comparable to those found by others in engines and even in burner flames. When these fractal dimensions are used to determine the ratio of laminar and turbulent flame surface, it turns out that the surface ratio thus determined is a factor 2-5 lower than the burning velocity ratio as calculated from the heat release analysis. Analysis of the images and flame contours shows that the major reason for this is the invalid assumption of a spherical flame shape. The true turbulent flame envelope is much larger than the half-sphere normally assumed. In this whole analysis, effects of flame strain and stretch have been neglected. These will play a role, however since this concerns lean methane/air mixtures these effects are thought to be less severe. Quantification of these effects would require more details on the local flame propagation. In chapter 6, some numbers were given for strain rates in the cold flow, but it is doubtful whether this is sufficient to determine the strain rates needed to quantify the effects of flame strain on the local laminar burning velocity.
Chapter 9

Conclusions and recommendations

In this thesis, a study on the phenomena occurring during turbulent combustion in a gas engine was presented. The aim of this study was twofold: firstly it was aimed to illustrate the occurrence of a practical limit to the amount of mixture dilution in natural gas engines. The second goal was to increase the understanding of the combustion process and its influencing factors, something that is needed if one wishes to model these lean combustion processes in predictive engine simulation codes. The investigation was conducted in separate parts which zoomed in on various aspects of the flow and combustion processes. The flow field -a property of the engine configuration- was investigated using Particle Image Velocimetry while the laminar burning velocity -a property of the mixture- was determined using a combustion vessel. Finally, the results of these investigations were used together to evaluate the combustion regime and draw further conclusions.

9.1 Conclusions

Literature survey

In the literature survey that was conducted it was found that the dilution limit is virtually the same for all lean burn natural gas engines. This limit is found around $\phi = 0.62 \ldots 0.59$, in terms of air excess ratio this corresponds to $\lambda = 1.6 \ldots 1.7$. The maximum efficiency however is mostly found for a slightly higher equivalence ratio.

Experimental set-up

For this research, an experimental engine was designed and constructed that features optical access to the combustion chamber through a window in the piston crown and through a transparent liner section. This engine was used for both continuously fired experiments -with the transparent sections replaced by full metal versions- and also for flow measurements and combustion visualizations. This engine design has a number of advantages but did also shown several shortcomings. The cooling facilities proved insufficient for an engine load exceeding 12 bars IMEP for a prolonged time. Furthermore, it proved a challenge to keep oil from the sump from entering the mirror section. This limited operation while recording images through the piston crown to about 30 minutes. The dry lubricated contact between the compression rings and the top liner does not
provide sufficient sealing against the high compression and combustion pressures. This resulted in increased blow-by which complicated analysis of the pressure signal.

**Continuously fired operation**

Measurements in the single cylinder engine with the transparent sections replaced by full-metal ones showed an operational dilution limit of $\lambda = 1.6 \ldots 1.7$. This is in line with the limit found in the literature survey that was conducted. At the operational limit both misfires and partial burns were observed. The maximum thermal efficiency was somewhat lower than expected; this was attributed to the increased blow-by in this engine. From the turbulent burning velocity that was calculated using a classical spherical flame assumption it was concluded that the scaling of the turbulent burning velocity with engine speed also holds for these lean mixtures over the engine speed range tested. The turbulent burning velocity found was about the same for methane and natural gas.

**Measurement of engine turbulence**

The turbulent flow field was measured using Particle Image Velocimetry. The engine was equipped with a transparent liner section for introduction of a laser sheet; the images were recorded through a window in the piston crown. Reflections from the valve recesses caused parts of the measurement section to be overexposed. Interpolation techniques had to be used to complete the velocity fields. It was found that this process is difficult to automate since the disturbances are not always the same, some user input is required which makes the process less robust. Reconstruction of larger structures is less accurate due to the inevitable filtering of interpolation algorithms. Due to the low acquisition rate of the PIV system, only one velocity field every few engine cycles could be recorded. The cycle-to-cycle fluctuations were separated from the in-cycle turbulent fluctuations using a spatial filtering technique. The spatial cut-off between high and low frequent fluctuations was chosen based on power spectral density diagrams combined with the visual appearance of ordered structures in the velocity field. Also this process has some user subjectivity in it. This process resulted in a single separation cut-off scale that was found appropriate for all measurements. The integral length scale which was determined using spatial autocorrelation functions did not show a significant variation for changing engine speed or manifold pressure. POD analysis was used to evaluate the structures in the flow. It was found that the dominating structure is a large vortex, this was expected since the (diesel) cylinder head has been designed for swirling flow. Besides this main vortex, several other structures were found though none of these had a consistent contribution to the energy of the flow.

**Burning velocity measurements**

The laminar burning velocity of the air-fuel mixtures was measured using the constant volume technique. These experiments were conducted in the EHPC. The burning velocity was determined along a pressure-temperature isentrope for three initial pressures and two initial temperature levels. It was found that the pressure-temperature range that could be covered was limited due to three reasons. Firstly by the choice of initial temperature which was limited to 475 K by material constraints. Secondly, due to the cubical shape of the EHPC the mass fraction burned was still low when the expanding sphere of combustion
products touched the walls and heat loss prevented the calculation of a burning velocity. Finally, the buoyant rise for slow burning mixtures limited the useable part of the pressure curve even further. The limitation imposed by the cubical shape was relieved by modifying the internal shape of the chamber. This resulted in an extension of the unburned temperature range of 50-100K. Despite this modification, the temperature-pressure range does not cover the range that is needed for engine combustion and extrapolation has to be used. The various measurements have been captured in a power-law type relation that describes the effects of pressure, temperature and equivalence ratio. A power law relation was constructed for lean mixtures as well as mixtures diluted with inert gases.

**Characterization of combustion**

The results from the work described were used to characterize the turbulent combustion process in the engine. The combustion ‘trajectories’, from ignition up to 80% burned were plotted in a Borghi diagram. From this it was found that the combustion process for stoichiometric mixtures occurs in the corrugated flamelets regime. For lean mixtures however the thin reaction zones regime is more appropriate. The start and end of the combustion process are found further into the thin reaction zones regime indicating a more ‘difficult’ combustion process with a higher probability of combustion instabilities. Mie scattering images were recorded during the initial stages of combustion for both stoichiometric and lean mixtures. This showed that the assumption of a spherically expanding flame is more or less justified for stoichiometric combustion. For lean mixtures however the shape of the expanding flame kernel is far from spherical, nor is it found in the cylinder center. The spherically expanding flame that is commonly used in engine modeling is thus less appropriate for lean or diluted mixtures. Fractal theory was applied to the Mie scattering images in an attempt to quantify the flame surface and compare it with the surface that is calculated from the overall mass burning rate and the laminar burning velocity. It was found that this fractal approach results in the right order of magnitude for the flame surface, however a factor 2-5 still remains. It was concluded that these flames exhibit fractal properties, however these are only valid for the range of scales between the applied inner and outer cut-off scale. The flame deformation on a scale larger than the applied outer cut-off is thus not accounted for. It is proposed that these large scale deformations (caused by large scale flow structures) should be regarded as a ‘property’ of the engine. For predictive engine modeling using lean mixtures therefore, both knowledge on the fundamental properties of the mixture as well as some basic characterization of the engine flow structures is needed.

### 9.2 Recommendations

The work presented in this thesis was largely experimental. Experimental work is often characterized by constructive issues and also limited resources. For the various experiments as described in this thesis, this was no different.
Experimental set-up

The most important recommendation in case one would start all over is that if one aims to perform both fired and non-fired (optically accessible) measurements in a single engine then the engine should have been designed as such from scratch. In this case the basic design of the engine, which was not part of this work, was made for flow measurements only. For this work the engine was adapted to also allow fired operation. This implies that the design of systems in and around the engine -intake, gas admission, exhaust but foremost the cooling system- is a compromise. This has caused a number of piston seizures. It is therefore recommended to use a different design for such an engine, or, use two separate engines for the fired and optically accessible measurements.

Flow field characterization

It would be interesting to verify the turbulence intensity as presented in chapter 6 with some other technique. The PIV system used was only capable of a low acquisition frequency. Typically one velocity field was recorded every few engine cycles. The separation of cycle-to-cycle fluctuations and in-cycle turbulent velocity fluctuations was therefore done using spatial filtering. With high speed PIV the in-cycle development of the velocity field can be measured directly.

When the reflections from the laser inside the combustion chamber could be avoided, the quality of the velocity fields would improve. For large inter/extrapolated areas the flow structures cannot be reconstructed fully; this may have an effect on the quantities derived from the velocity fields.

Burning velocity measurements

The measurement of the laminar burning velocity as described in chapter 7 has been limited by the experimental set-up. The use of a cubic chamber limits the range of pressures and temperatures for which the burning velocity can be determined. It is therefore recommended to use a spherical chamber. Ideally, for lean or diluted mixtures that burn slower, gravity should also be switched off. This however is more difficult to realize.

For very lean mixtures, deformation of the sphere of combustion products is observed along with the effects of buoyancy. This deformation, which affects the flame surface was not accounted for in the calculation of the burning velocity as it could not be quantified from a shadowgraphy image. A different technique which is able to visualize the shape of the flame (i.e. Mie scattering with a laser sheet) may be used for an improved estimation of the flame surface. This would then improve the burning velocity calculation for lean mixtures.

Characterization of combustion

The Mie scattering images as presented in chapter 8 could be improved by using a different seeding material. It is expected that smaller particles that evaporate or burn more easily will result in more clearly defined edges of the flame envelope. Also, since the Mie scattering images are a two-dimensional slice of a three-dimensional flame structure, images
recorded in a plane parallel to the cylinder axis could confirm the isotropy assumed in
the fractal approach. Furthermore, just as with the flow measurements the use of a high
speed laser could reveal information on the temporal evolution of the flame structure.
Appendix A

Fast response analyzer

The operation of a fast-response FID is in principle no different from a normal FID analyzer. The detection mechanism is the same. A small amount of exhaust gas is burned along with a hydrogen-helium-air flame. This process generates ions during the oxidation of the carbon atoms in the sample. These ions are collected at an electrode near the flame and generate a small current. This current is proportional to the amount of carbon atoms that react. Effectively the FID analyzer is a carbon-atom counter. This is also a weakness since the type of hydrocarbon cannot be distinguished. The reading of a FID is a total hydrocarbon reading and is expressed in C1 equivalent. The specific characteristic of a fast response FID is its speed. In this type of analyzer there is a much higher sample flow which is controlled by a separate vacuum chamber (CP-chamber in figure A.1). The vacuum port, together with the CP-bleed keep the CP chamber at a fixed low pressure which guarantees a continuous high flow from the sample probe. The pressure in the FID chamber is slightly lower than the pressure in the CP chamber. This pressure difference assures a continuous, constant flow of sample gas to the FID flame. Since the CP chamber has a relatively large volume and its pressure is actively controlled by the vacuum and CP bleed, the fast FID is insensitive to pressure variations at the sample location. This makes sampling possible at locations where the pressure pulsates (e.g. an exhaust port) or even at high pressure (with some precautions.
even in-cylinder samples are possible). Due to the thin sample line and high sample flow the response time is short.

In figure A.2, the location of the FFID probe is shown. The tip of the probe is located in the exhaust port in the cylinder head, approximately 100mm from the valves. The probe has been constructed double-walled, so as to enable in-situ calibration through the sample line. Of course the engine has to be stopped or a very low load should be applied. The calibration gases are admitted through the outer concentric pipe, the tip of the sample pipe is recessed a small distance such that it is made sure that only calibration gases are sampled. Calibration can be performed in little time. In figure A.3, the actual set-up is shown. The sample line to the analyzer is still about 1 meter. This causes an additional delay which must be accounted for. See also figure 4.5.
Figure A.3: Actual set-up with FR analyzers and heated sample line, 1) heated sample line (length: 1 meter), 2) CLA analyzer (NO), 3) FFID analyzer (HC), 3) NDIR analyzer (CO/CO₂)
Appendix B

Emissions calculations

In this appendix, the different methods for calculating the air excess ratio $\lambda$ from emissions measurements are presented. In the formulas presented, the emissions concentrations are indicated with square brackets. All species concentrations are expressed in mole fractions. In the relations, the following symbols are used to avoid repetition and save space:

$$\alpha = \text{fuel CO}_2 \text{ fraction}$$
$$n = \text{fuel H/C ratio}$$
$$m = \text{fuel O/C ratio}$$
$$k = \text{constant for water-gas reaction} = 3.5$$
$$X = [CO] + [CO_2] + [HC]$$

Some emissions are measured 'wet' (i.e. without condensation of the water fraction in the exhaust gases prior to measurement). An example of this is the hydrocarbon emission which is measured using the FID. As these relations for the calculation of $\lambda$ should be used with either only dry or only wet concentrations, this has to be converted to 'dry' measurement using the water fraction in the exhaust gases. In the formulas, all species concentrations are on a dry basis.

**B.1 Carbon balance method**

The carbon balance method requires the concentration of $CO_2$, $CO$ and unburned hydrocarbons $CH_x$. Additionally, the $H/C$ and $O/C$ ratio of the fuel are required.

$$\lambda_C = \frac{1 - (1 - \alpha) \cdot X - 0.5[CO] + ([HC] - (1 - \alpha) \cdot X) \cdot \left( \frac{0.75m}{1 + \frac{[CO]}{[CO_2]}} \right) - (1 + 0.25n - 0.5m)}{(1 - \alpha) \cdot X \cdot 4.76 \cdot (1 + 0.25n - 0.5m)}$$  \hspace{1cm} (B.1)

**B.2 Oxygen balance method**

The oxygen balance method requires, besides $CO_2$, $CO$ and $HC$, also the concentration of nitrogen oxides $NO_x$ and the oxygen concentration $O_2$ in the exhaust gases. The
corresponding relation is

\[
\lambda_O = \frac{[CO_2] + 0.5[CO] + [O_2] + 0.5[NO_x] + ((1 - \alpha) \cdot X - [HC]) \cdot \left( \frac{0.25n}{1 + \frac{k}{k+[CO]/[CO_2]}} - 0.5m \right)}{(1 - \alpha) \cdot X \cdot (1 + 0.25n - 0.5m)}
\]

(B.2)

B.3 Brettschneider method

Finally, the Brettschneider relation ([9]) for determining the air-excess ratio \( \lambda \) also accounts for humidity and water in the fuel. While the latter is not often found, the humidity of the intake air is something that should be accounted for. Next to the symbols already defined, the following quantities are used.

\[
\begin{align*}
\mu &= \frac{M_{\text{air,dry}}}{M_{\text{water}}} (= 1.608) \\
w &= \text{fuel } W/C \text{ (water/carbon) ratio} \\
H_a &= \text{absolute humidity intake air } [g/kg]
\end{align*}
\]

The Brettschneider relation then becomes

\[
\lambda_{\text{Brettschneider}} = \frac{21}{21 + 50\mu \cdot \frac{H_a}{1000 \frac{[CO]/[CO_2]}{k+[CO]/[CO_2]}}} \times \ldots \\
\frac{[CO_2] + 0.5[CO] + [O_2] + 0.25n \cdot \frac{k}{k+[CO]/[CO_2]} - 0.5m}{{(1 + 0.25n - 0.5m) \cdot (\frac{[CO_2] + [CO]}{[CO_2] + [CO] + [HC]})}} \times \ldots \\
0.5w \cdot \frac{\frac{[CO]}{k+[CO]/[CO_2]}}{(1 + 0.25n - 0.5m) \cdot X}
\]

(B.3)
Appendix C

Heat release model for engine combustion

The two-zones combustion model used for processing the measured pressure traces is derived from the model as described by Ferguson [24]. The treatment of blow-by flow however was changed to better suit this specific engine.

C.1 Geometry

For the model, the combustion chamber is divided into a main volume and a crevice volume as displayed in figure C.1. During compression, when the pressure rises gases flow into the crevice volume. Due to the piston ring slots and also because these rings do not have the contact pressure and surface smoothness of a normal piston ring, gases can slip past the piston rings and out of the engine. This phenomenon is known as blow-by and occurs in all engines to a certain degree. Because of the special construction which employs piston rings without lubrication, the amount of blow-by gases in this engine is larger than average. This is why these flows have to be taken into account in the model. Also, due to the construction with a removable liner top, the piston rings are located relatively far from the piston top. This leads to crevice volumes that are larger than in a standard engine.

C.2 Calculation model

The calculation is performed for the closed-valve period only. The cylinder contents at IVC is determined by the sum of the fresh mass as determined from the air and fuel flow rates and a residual mass that is determined by estimating the mass at IVO and calculating the back flow from the exhaust. The equivalence ratio $\lambda$ is known from the measured air and fuel flows and also from emissions measurements. The pressure at IVC is taken from the pressure curve, assuming the pressure curve has been pegged correctly. The model starts as a single zone calculation (with allowance for crevice effects and blow-by) which is used up to the point where combustion starts. For each step, starting at IVC, the following calculations are performed. First, the heat loss $\dot{Q}_u$ in the interval is determined from the current gas temperature, the wall temperature and heat transfer
Figure C.1: Combustion chamber boundary

\[ \dot{Q}_u = hA(T_{u,i} - T_w) \]  
\[ (C.1) \]

The change in specific entropy is then determined from the heat transfer rate and the current mass in the main volume.

\[ \frac{ds_u}{d\theta} = -\frac{\dot{Q}_u}{\omega m T_u} \]  
\[ (C.2) \]

The temperature of the unburned mixture \( T_{u,i+1} \) is found using Newton-Raphson iteration on

\[ T_{u,i+1} = T_{u,i} \left[ 1 + \frac{s_{u,i+1} - s_u(T_{u,i}, p_i)}{c_{p,i}} \right], \]  
\[ (C.3) \]

thereby calculating the unburned temperature that belongs to the specific entropy \( s_{u,i+1} \) at the next CA step. Since the rate of change in pressure, temperature, volume and heat loss are known, the apparent heat release can be determined if the flow \( \dot{m}_{\text{loss}} \) into or out of the main volume is also known. To determine \( \dot{m}_{\text{loss}} \), the crevice volume needs to be modeled also. It is assumed that, since the crevices have a large surface-to-volume ratio, the temperature of the crevice gases \( T_{cr} \) is always the same and is equal to the average of the wall and piston temperature, \( T_{cr} = (T_w + T_p)/2 \). The change in crevice mass can then be expressed as

\[ \frac{dm_{cr}}{d\theta} = \frac{V_{cr}}{R_{cr} T_{cr}} \frac{dp}{d\theta} + \frac{p}{R_{cr} T_{cr}} \frac{dV_{cr}}{d\theta} \]  
\[ (C.4) \]

The contribution from the change in crevice volume \( dV_{cr} \) is small, but since the top liner has a slightly larger diameter the crevice volume changes with piston position. The blow-by flow across the piston rings is modeled using a choked flow relation, since the ratio
of in-cylinder pressure to atmospheric pressure exceeds the limit for choked flow almost always. The area $A_{bb}$ through which the blow-by flow takes place is determined from a calibration procedure using a number of (hot) motored pressure traces for which no heat input occurs. The blow-by flow is then used as a closing term in the apparent heat release. A consistent value could only be determined for the part of the cycle in which the pressure is high (around TDC). When the piston is further from TDC, the pressure is low and the resulting low mass loss is obscured by variations in heat transfer. From the part of the cycle around TDC, an average area of 5-8 mm$^2$ was determined. This area is not constant but has a maximum some 20 degrees before TDC. This observation is believed to be related to the piston motion as the piston rings may have trouble to follow the piston movement as the piston crosses the bore to the thrust side. This blow-by model however is a compromise, as limited data was available for the calibration and the effects of heat and even higher gas pressure during combustion cannot be accounted for.

The blow-by contribution follows from

$$\frac{d m_{bb}}{d \theta} = C_D A_{bb} \sqrt{\frac{\rho_c P_{cr}}{\gamma_{cr}}} \frac{2}{\gamma_{cr} + 1} \gamma_{cr} \frac{\gamma_{cr} - 1}{\gamma_{cr}} (C.5)$$

The total mass loss from the main volume is then the sum of both contributions

$$\frac{d m_{loss}}{d \theta} = \frac{d m_{cr}}{d \theta} + \frac{d m_{bb}}{d \theta} (C.6)$$

The change in internal energy follows from

$$\frac{d U}{d \theta} = -\frac{d W}{d \theta} - \frac{d Q_u}{d \theta} - \frac{d m_{loss}}{d \theta} h_{loss} (C.7)$$

The enthalpy $h_{loss}$ in equation C.7 depends on the direction of the flow. During compression, gases flow from the main volume into the crevice and the enthalpy is evaluated at the temperature $T_u$ in the main volume. Late in the expansion, gases re-emerge from the crevice (provided that the blow-by flow is sufficiently low) and the properties are evaluated at crevice temperature. After each step, the fluid properties are updated using the new temperature $T_{u,i+1}$ and the pressure $p_{i+1}$.

In the two-zones formulation, the cylinder contents is split in two separate zones. During the two-zones part of the model, the mass average energy and volume can be distributed over the burned and unburned zone.

$$U/m = (1 - x_b)u_u + x_b u_b (C.8)$$

$$V/m = (1 - x_b)v_u + x_b v_b (C.9)$$

After each step in the single zone part of the model, starting at the spark angle, equations C.8 and C.9 are solved for the mass fraction $x_b$, assuming that the temperature of the burned zone equals the adiabatic flame temperature. A switch is made to a two zones calculation if $x_b$ exceeds some small value.

For the unburned gas, the composition is assumed frozen and the specific energy and volume are functions of $T_u$ and $p$. For the burned gas, chemical equilibrium is assumed among 10 major species. The method of Olikara and Borman is used. Given the composition, the specific energy and volume are again functions of $T_b$ and $p$. The steps involved to
get from each pressure data point to the next are again to first determine the temperature of the unburned mixture using equation C.3. Since a fraction of the cylinder volume is occupied by burned products, the contact surface for heat loss from the unburned mixture is less than the total surface of piston top, cylinder head and exposed liner wall. For fast burning mixtures ($\lambda = 1$), the assumption of a spherically propagating flame as displayed in figure C.1 is more or less justified. For lean mixtures however, the convection velocity of the flame is large so that the flame touches the walls already in an early stage of combustion. It was therefore decided to not assume any flame shape a priori, but to take the contact surface as proportional to $(1 - x_b^{1/2})$ and $x_b^{1/2}$ for the unburned and burned zones respectively.

The total internal energy at the next point is determined from

$$\frac{dU}{d\theta} = -\frac{dW}{d\theta} - \frac{dQ_u}{d\theta} + \frac{dQ_b}{d\theta} - \frac{dm_{loss}}{d\theta} h_{loss}$$

(C.10)

Next, equations C.8 and C.9 are solved for the temperature of the burned zone $T_{b,i+1}$ and the mass fraction burned $x_{b,i+1}$ at the next point. The mass in the main volume $m_{i+1}$ is found using equation C.6 again. A difference is that the properties of the gases flowing into the crevice volume are evaluated from both burned and unburned zones using the mass fraction burned as weighting factor (equations C.11 and C.12).

$$h_{loss} = (1 - x_b^{1/2})h_u + x_b^{1/2}h_b$$

(C.11)

$$u_{loss} = (1 - x_b^{1/2})u_u + x_b^{1/2}u_b$$

(C.12)

The composition of the gases in the crevice is also treated differently. From early model results, it was found that the total blow-by mass is of the same order as the gas mass residing in the crevice at maximum pressure. It is therefore not justified to assume that the gases in the crevice consist of only unburned mixture during the entire period. The composition of the gases in the crevice was therefore also taken as weighted sum of burned and unburned composition using again the mass fraction burned as weighting factor, similar to equation C.11 and C.12.

Working through the pressure curve up to EVO, the model results in the mass fraction burned, the volumes of the burned and unburned zones and the (mass averaged) temperature of both zones. Since the aim was to investigate the effects of turbulence, an (apparent) turbulent burning velocity can be obtained from the mass fraction burned if one assumes some shape for the flame. As no real preference for one shape or the other exists, the flame was assumed to propagate from the spark plug outward. Using the momentary burned volume $V_b$, a flame radius can be determined and from that a flame surface. The resulting turbulent burning velocity $S_T$ is determined from the mass burning rate $\dot{m}_b$, the flame surface $A_f$ and the unburned density $\rho_u$ as in equation C.13.

$$S_T = \frac{\dot{m}_b}{\rho_u A_f}$$

(C.13)

Of course, this representation of the turbulent flame has the minimum area possible. In reality, due to large scale gas motion, the flame is not regular. The enveloping surface is larger and as a result the turbulent burning velocity is lower.

The mass fraction burned $x_b$ is always relative to the momentary mass in the main volume.
This way, an engine efficiency based on the amount of fuel entering into the engine may be somewhat low since some of the fuel escapes through blow-by or fails to burn due to crevice effects.
Appendix D

Graphs of laminar and turbulent burning velocity

Figure D.1: Turbulent burning velocity for different engine working points, $S_T$ in [m/s]
Figure D.2: Laminar burning velocity for different engine working points, $S_L$ in [m/s]
Appendix E

Heat release model for constant volume combustion

The calculation model for constant volume combustion was derived from a two-zones model for engine combustion that was developed by Ferguson ([23]). Basically, the model was modified by removing terms associated with volume change. The model calculates \( \frac{dp}{dt} \), \( \frac{dT_u}{dt} \), \( \frac{dT_b}{dt} \) and heat losses to the walls. At a later stage, also crevice effects were incorporated.

As is shown in figure E.1, the burned zone is assumed to expand as a sphere from the center of the vessel (either cubical or polyhedron). During the initial stages, heat losses to the walls are modeled using convective heat transfer from the unburned zone to the walls. The heat transfer surface is simply the sum of the walls that make up the combustion chamber. The temperature difference is \( T_u - T_w \). Once the burned zone has grown such that it touches the walls, several contact patches will form through which the heat transfer will increase due to the higher temperature difference \( T_b - T_w \). The heat transfer surfaces \( A_u \) and \( A_b \) for the unburned and burned zone are a function of the size of the burned zone, and are determined from the flame radius \( r_f \) (assuming spherical symmetry).
\begin{align*} 
A_u &= f(r_f) 
\text{(E.1)} \\
A_b &= f(r_f) 
\text{(E.2)} \\
& \text{(E.3)} \\
\end{align*}

The heat transfer coefficient was determined using the assumption of a constant Nusselt number. An empirical ’calibration factor’ was used which was multiplied with the thermal conductivity of the gases.

\begin{align*} 
h_{c,u} &= Nu \frac{k}{L} = \text{calibration factor} \cdot k_u \quad \text{(E.4)} \\
h_{c,b} &= Nu \frac{k}{L} = \text{calibration factor} \cdot k_b \quad \text{(E.5)} \\
\end{align*}

This calibration factor was ’tuned’ such that, after the ’tuning’ of the crevice volume, the burned fractions reached unity on average. For the part of the pressure rise that was used for extracting the burning velocity however, this is not critical as the major part of the total heat transfer takes place when the flame touches the walls. The resulting heat transfer terms become

\begin{align*} 
Q_{\text{conv},u} &= h_{c,u} A_u (T_u - T_w) 
\text{(E.7)} \\
Q_{\text{conv},b} &= h_{c,b} A_b (T_b - T_w) 
\text{(E.8)} \\
\end{align*}

The mass loss from the main volume $dm_{\text{out}}$ and into the crevice zone is determined by

\begin{equation} 
\frac{dm_{\text{out}}}{dt} = - \frac{dm_{\text{cr}}}{dt} = \frac{V_{cr}}{R_u T_w} \frac{dp}{dt} \quad \text{(E.10)}
\end{equation}

The mass loss from the main volume was incorporated in the same fashion as the blow-by in the original model from Ferguson. The resulting equations become:

\begin{align*} 
C_{\text{loss}} &= \frac{1}{m} \frac{dm_{\text{out}}}{dt} 
\text{(E.11)} \\
A &= \frac{V}{m} C_{\text{loss}} 
\text{(E.12)} \\
B &= \frac{1}{m} \left( \frac{v_b}{c_{p,b} T_b} \frac{\partial lnv_b}{\partial lnT_b} Q_{\text{conv},b} + \frac{v_u}{c_{p,u} T_u} \frac{\partial lnv_u}{\partial lnT_u} Q_{\text{conv},u} \right) 
\text{(E.13)} \\
C &= - (v_b - v_u) \frac{dx_b}{dt} - v_b \frac{\partial lnv_b}{\partial lnT_b} h_u - h_b \frac{dx_b}{dt} - (x_b - x_b^2) C_{\text{loss}} 
\text{(E.14)} \\
D &= x_b \left( \frac{v_b^2}{c_{p,b} T_b} \frac{\partial lnv_b}{\partial lnT_b} \right)^2 + v_b \frac{\partial lnv_b}{\partial np} 
\text{(E.15)} \\
E &= (1 - x_b) \left( \frac{v_u^2}{c_{p,u} T_u} \frac{\partial lnv_u}{\partial lnT_u} \right)^2 + v_u \frac{\partial lnv_u}{\partial np} 
\text{(E.16)}
\end{align*}
The resulting relations for the change in pressure and temperature for the burned and unburned zone as well as the total heat loss then become

\[
\frac{dp}{dt} = \frac{A + B + C}{D + E} \quad (E.17)
\]

\[
\frac{dT_b}{dt} = \frac{Q_{conv,b}}{mx_b c_{p,b}} + \frac{v_b}{c_{p,b}} \frac{\partial ln v_b \, dp}{\partial ln T_b} dt + \frac{h_u - h_b}{c_{p,b} x_b} \left( \frac{dx_b}{dt} - (x_b - x_b^2) C_{loss} \right) \quad (E.18)
\]

\[
\frac{dT_u}{dt} = \frac{Q_{conv,u}}{m(1 - x_b) c_{p,u}} + \frac{v_u}{c_{p,u}} \frac{\partial ln v_u \, dp}{\partial ln T_u} dt \quad (E.19)
\]

\[
\frac{dQ_{loss}}{dt} = Q_{conv,u} + Q_{conv,b} \quad (E.20)
\]
Appendix F

Burning velocity data fit

In this appendix, the procedure used for deriving the burning velocity correlation is explained in more detail. More information can be found in [79]. Burning velocity measurements have been performed in a constant volume chamber for various equivalence ratios, and combinations of initial pressure and temperature. The two zones model was used to calculate the laminar burning velocity $S_L$ from the measured pressure traces. This results in the burning velocity along a pressure-temperature polytrope.

F.1 Lean mixtures

For every mixture equivalence ratio used, the burning velocity has been determined for three or six initial conditions. For all equivalence ratios, three initial pressures have been used with an initial temperature of 475K. For selected equivalence ratios, also an initial temperature of 400K has been used. The burning velocity measured this way has been fitted with a correlation in power law form as explained in section 7.9.1. This power law form is repeated here for clarity.

$$S_L(p, T_u, \phi, D) = S_{L,0}(\phi, D) \cdot \left(\frac{T}{T_0}\right)^\alpha \cdot \left(\frac{p}{p_0}\right)^\beta$$

In the procedure used the coefficients for temperature and pressure dependence $\alpha$ and $\beta$ as well as the base burning velocity $S_{L,0}$ were varied such that the mean error $\overline{err}$

$$\overline{err} = \frac{\sqrt{\sum_{i=1}^z ((S_{L,i})_{meas}-(S_{L,i})_{powerlaw})^2}}{z}$$

is minimized. First, this procedure was used on the burning velocity data for each equivalence ratio separately. This results in a set of $\alpha$, $\beta$ and $S_{L,0}$ for each equivalence ratio or air excess ratio. These parameters have been plotted along with the resulting mean error $\overline{err}$, maximum error $err_{max}$ and regression coefficient $R^2$ in table F.1. On average the mean error $\overline{err}$ is only a few percent except for the air excess ratios $\lambda=1.5$, 1.6 and 1.7. For these air excess ratios data for $T_i=400K$ as well as $T_i=475K$ was used. When using also the data for 400K initial temperature, the quality of the fit becomes less. This is also visible in figure F.1, where both the measured data and the resulting burning velocity
Burning velocity data fit

\[
\begin{array}{cccccc}
\lambda [-] & S_{L,0} [cm^2] & \alpha [-] & \beta [-] & \text{err} [\%] & \text{err}_{\text{max}} [\%] & R^2 [-] \\
1.0 & 24.47 & 2.01 & -0.345 & 3.0 & 13.1 & 0.951 \\
1.1 & 19.45 & 2.58 & -0.369 & 1.6 & 6.7 & 0.989 \\
1.2 & 15.97 & 2.83 & -0.390 & 1.5 & 5.3 & 0.993 \\
1.3 & 13.47 & 3.05 & -0.443 & 1.4 & 4.8 & 0.994 \\
1.4 & 10.20 & 3.36 & -0.493 & 1.2 & 4.0 & 0.996 \\
1.5 & 6.77 & 4.00 & -0.577 & 11.6 & 26.8 & 0.840 \\
1.55 & 6.26 & 4.04 & -0.615 & 1.6 & 9.2 & 0.992 \\
1.6 & 4.83 & 4.34 & -0.640 & 14.4 & 27.8 & 0.816 \\
1.65 & 6.36 & 3.92 & -0.708 & 1.5 & 5.7 & 0.997 \\
1.7 & 4.24 & 4.10 & -0.654 & 9.8 & 22.0 & 0.916 \\
1.75 & 3.81 & 4.37 & -0.721 & 2.0 & 8.5 & 0.997 \\
1.8 & 2.50 & 4.54 & -0.649 & 2.0 & 6.6 & 0.995 \\
\end{array}
\]

Table F.1: Base burning velocity \(S_{L,0}\), temperature and pressure coefficients \(\alpha\) and \(\beta\) for a fit per air excess ratio, together with the mean error \(\text{err}\), maximum error \(\text{err}_{\text{max}}\) and regression coefficient \(R^2\) for a fit using this local correlation from the power law fit have been plotted for an air excess ratio of 1.5. In this figure, it can be seen that an individual fit for each curve would be a bit steeper. Apparently, the physics are not captured fully with this type of power law equation. It was however decided to leave the data for \(T_i=400K\) in since it extends the temperature range where the fit can be used to the low side. A further reason for the different slope of the measured data and the fit may be the error that is made by assuming a spherical shape. For lean or diluted mixtures, buoyancy effects lead to a non-spherical shape with a larger surface. The burning velocity is thus overestimated by assuming a sphere. The upward 'bend' in the measured burning velocity curves could indicate this.

In section 7.9.1, it was already explained that the various power law equations per equivalence ratio have been combined to one general power law relation by making the base burning velocity \(S_{L,0}\) and the temperature and pressure coefficients \(\alpha\) and \(\beta\) a function of the equivalence ratio. This operation reduces the quality of the fit per separate equivalence ratio. In the last three columns of table F.2, the quality of the fit in terms of both the mean error \(\text{err}\) and maximum error \(\text{err}_{\text{max}}\) as well as the regression coefficient \(R^2\) have been plotted for a similar comparison of the measurement data with a power law fit, only now the \(\phi\)-dependent power law equation (equation 7.2) has been used. On average, the regression coefficient \(R^2\) has decreased.

F.2 'EGR' diluted mixtures

For the mixtures diluted with synthetic EGR the number of measurements was less. This reflects in the quality of the fit that was derived from the data. In tables F.3 and F.4, the fit parameters \(S_{L,0}\), \(\alpha\) and \(\beta\) as well as the mean error \(\text{err}\), maximum error \(\text{err}_{\text{max}}\) and the regression coefficient \(R^2\) are shown in a similar fashion for a local fit using a correlation
Table F.2: Base burning velocity $S_{L,0}$, temperature and pressure coefficients $\alpha$ and $\beta$ for a global fit (equation 7.2), together with the mean error $\overline{err}$, maximum error $err_{max}$ and regression coefficient $R^2$ for a fit using this overall correlation.

<table>
<thead>
<tr>
<th>$\lambda$ [-]</th>
<th>$S_{L,0}$ [cm s$^{-1}$]</th>
<th>$\alpha$ [-]</th>
<th>$\beta$ [-]</th>
<th>$\overline{err}$ [%]</th>
<th>$err_{max}$ [%]</th>
<th>$R^2$ [-]</th>
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<tr>
<td>1.0</td>
<td>24.52</td>
<td>2.03</td>
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<td>17.1</td>
<td>0.912</td>
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<td>1.1</td>
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<tr>
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<td>8.9</td>
<td>0.978</td>
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<td>3.82</td>
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<td>9.6</td>
<td>20.5</td>
<td>0.897</td>
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<td>1.55</td>
<td>6.60</td>
<td>3.95</td>
<td>-0.587</td>
<td>7.5</td>
<td>15.4</td>
<td>0.950</td>
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<td>5.65</td>
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<td>1.65</td>
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<td>-0.709</td>
<td>14.5</td>
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Figure F.1: Measured laminar burning velocity for $\lambda = 1.5$ ($\phi=0.67$) and various initial conditions (continuous lines). Power law fit results have been shown in dotted lines.
per dilution fraction and a global fit where the coefficients are a function of the dilution fraction (equation 7.3). The regression coefficients here are clearly lower than those found for the lean mixtures.

<table>
<thead>
<tr>
<th>$D$ [%]</th>
<th>$S_{L,0}$ cm s$^{-1}$</th>
<th>$\alpha$ [-]</th>
<th>$\beta$ [-]</th>
<th>$\text{err}$ [%]</th>
<th>$\text{err}_{\text{max}}$ [%]</th>
<th>$R^2$ [-]</th>
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<td>0</td>
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<tr>
<td>15</td>
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<td>20</td>
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**Table F.3:** Base burning velocity $S_{L,0}$, temperature and pressure coefficients $\alpha$ and $\beta$ for a fit per dilution fraction, together with the mean error $\text{err}$, maximum error $\text{err}_{\text{max}}$ and regression coefficient $R^2$ for a fit using this local correlation.

<table>
<thead>
<tr>
<th>$D$ [%]</th>
<th>$S_{L,0}$ cm s$^{-1}$</th>
<th>$\alpha$ [-]</th>
<th>$\beta$ [-]</th>
<th>$\text{err}$ [%]</th>
<th>$\text{err}_{\text{max}}$ [%]</th>
<th>$R^2$ [-]</th>
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<td>15</td>
<td>8.63</td>
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<td>-0.589</td>
<td>11.1</td>
<td>21.8</td>
<td>0.835</td>
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**Table F.4:** Base burning velocity $S_{L,0}$, temperature and pressure coefficients $\alpha$ and $\beta$ for a global fit (equation 7.3), together with the mean error $\text{err}$, maximum error $\text{err}_{\text{max}}$ and regression coefficient $R^2$ for a fit using this overall correlation.
Appendix G

Air cooling system

In this appendix, some more info on the piston cooling system that was used for the fired engine experiments is given. The system that was finally used consisted of a pipe bundle trough which fluid was circulated using the piston acceleration and several non-return valves. This was explained in chapter 4. Here, some calculations of the cooling performance are presented. The complete cooling system is shown in figure G.1.

![Air cooling system for fired engine operation without optical access](image)

Figure G.1: Air cooling system for fired engine operation without optical access

The airflow is induced by a centrifugal blower that was located in the basement. The pipe leading to this blower is shown on the left side of the engine. On the entry side of the cooling section, cold air with a temperature down to zero degrees C is supplied using an air conditioning unit. The amount of cold air this unit could deliver however
was less than what the airflow induced by the blower. Ambient air is therefore added. The amount of air that was flowing through the cooling section was determined using a velocity probe. The flow velocity was measured in the straight, wide pipe section at different radial positions using a hand-held flow probe. This is indicated by the dashed line in figure G.1. The flow profile thus determined was an approximate plug flow profile. The volumetric flow rate could therefore easily be determined from the velocity and pipe cross section area. This flow rate was approximately 900 m$^3$/hr. The warm air flow leaving the blower exited outside the building as also blow-by gases from the engine are mixed with the cooling air stream.

For making an estimation of the cooling performance, the piston cooler can be regarded as a cylinder-array. Such a cylinder array is a geometry for which engineering correlations can be found in literature. A major difference of this piston cooler compared to the cylinder array found in ordinary heat exchangers is that these 'cylinders' are moving up and down with the piston speed. The maximum piston speed in this case is about 10 m/s at an engine speed of 1400 rpm. The flow velocity found just upstream or downstream the cooler, assuming incompressible flow, is 35 m/s. This however is not the speed with which the air flows through the pipe bundle. Due to the space around the cooler, which is shown on the left side of figure G.2, part of the cooling air travels around the actual cooler. In the vertical direction, it is likely that the air flow 'stream tube' is widened also. This is shown on the right side of figure G.2. For the calculation of the cooling capacity,

![Cross section A-A (2:1)](image)

**Figure G.2:** Air flow detail

it is assumed that the convective heat transfer on the outside of the tubes is limiting. The maximum allowable surface temperature was taken 100°C to prevent excessive pressure buildup in the cooler. In literature, correlations for the Nusselt number for flow through
a cylinder array can be found. Bejan [5] gives for a staggered cylinder array:

\[ Nu_D = 0.27 C_n R_e_D^{0.63} P_r^{0.36} \left( \frac{P_r}{P_r_w} \right)^{0.25} \]  

(G.1)

The effective flow velocity through the cooler is estimated at 20 m/s. This seems reasonable since the air flow 'jet' is directed by the entry pipe entering the cooling section and the available cross section through the cooler is approximately the same as the flow cross section around the cooler. This flow velocity however was not verified by measurement. Together with the diameter of the cooler pipes this leads to a Reynolds number \( R_e_D \) of

\[ R_e_D = \frac{v D}{\nu} = \frac{20 \cdot 0.008}{16.5 \cdot 10^{-6}} = 9700. \]  

(G.2)

Taking \( P_r = 0.71 \) for the average air temperature in the cooler section and \( P_r_w = 0.703 \) for the Prandtl number at a pipe surface temperature of 100°C, the Nusselt number becomes \( N u_D = 73.6 \). The coefficient \( C_n \), taken 0.95 here is a function of the number of pipes in the array ([5]). The resulting heat transfer coefficient \( \overline{h} \) then becomes

\[ \overline{h} = \frac{N u_D k}{D} = \frac{73.6 \cdot 0.027}{0.0072} = 276 \text{ W/m}^2\text{K}. \]  

(G.3)

The effective pipe diameter used here is somewhat smaller since these pipes were threaded to increase their surface area.

The effective pipe length (i.e. the part of the pipes that is cooled by the air stream) is difficult to determine since the flow pattern is unknown. For a minimum effective pipe length of 96mm (the diameter of the entry hole to the cooling section) the total pipe surface area is 0.204 m². This results in a cooling capacity of

\[ \dot{Q} = \overline{h} A \Delta T = 276 \cdot 0.204 \cdot 70 = 3.9kW. \]  

(G.4)

The average temperature difference of 70°C was found from measurement of the temperature at entry and exit of the cooling section. Considering that, due to the piston motion, the flow through the cooler also gets a vertical component and also that effectively a substantially larger part of the total pipe length is cooled the cooling power to be realized may be \( \mathcal{O}(10kW) \).
List of Symbols

Roman symbols

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Description</th>
<th>Unit</th>
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<tr>
<td>$\dot{m}_b$</td>
<td>mass burning rate</td>
<td>$kg\ m^{-2}\ s^{-1}$</td>
</tr>
<tr>
<td>$A$</td>
<td>reaction frequency factor</td>
<td>–</td>
</tr>
<tr>
<td>$C_D$</td>
<td>Stokes drag coefficient</td>
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\( Y \)  \( \text{species mol fraction} \)

**Greek symbols**

- \( \delta_f \)  \( \text{laminar flame thickness} \)  \( m \)
- \( \delta_{il} \)  \( \text{inner layer thickness} \)  \( m \)
- \( \epsilon_i \)  \( \text{fractal inner cutoff scale} \)  \( m \)
- \( \epsilon_o \)  \( \text{fractal outer cutoff scale} \)  \( m \)
- \( \eta \)  \( \text{Kolmogorov length scale} \)  \( m \)
- \( \gamma \)  \( \text{ratio of specific heats} \)  
- \( \lambda \)  \( \text{air excess ratio} \)  
- \( \mu \)  \( \text{dynamic viscosity} \)  \( kg \, m^{-1} \, s^{-1} \)
- \( \nu \)  \( \text{kinematic viscosity} \)  \( m^2 \, s^{-1} \)
- \( \omega \)  \( \text{angular frequency} \)  \( rad \, s^{-1} \)
- \( \omega \)  \( \text{vorticity} \)  \( s^{-1} \)
- \( \phi \)  \( \text{equivalence ratio} \)  
- \( \rho \)  \( \text{density} \)  \( kg \, m^{-3} \)
- \( \varepsilon \)  \( \text{energy dissipation rate} \)  
- \( \varepsilon \)  \( \text{strain rate} \)  

**Subscripts**

- \( b \)  \( \text{burnt} \)
- \( EA \)  \( \text{ensemble averaged} \)
- \( f \)  \( \text{fluid} \)
- \( HF \)  \( \text{high frequency part} \)
- \( il \)  \( \text{inner layer} \)
- \( L \)  \( \text{laminar} \)
- \( LF \)  \( \text{low frequency part} \)
- \( p \)  \( \text{at constant pressure} \)
- \( p \)  \( \text{particle} \)
- \( rms \)  \( \text{root mean square} \)
- \( t, T \)  \( \text{turbulent} \)
- \( u \)  \( \text{unburnt} \)
- \( v \)  \( \text{at constant volume} \)

**Superscripts**

- \( 0 \)  \( \text{unstretched} \)

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<td>Bottom Dead Center</td>
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<tr>
<td>CI</td>
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<td>DNG</td>
<td>Dutch Natural Gas</td>
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<td>EGR</td>
<td>Exhaust Gas Recirculation</td>
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<td>EMAP</td>
<td>Exhaust Manifold Pressure</td>
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<td>EVC</td>
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<td>Proper Orthogonal Decomposition</td>
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**Dimensionless numbers**

- \( Da \) Damkohler number
- \( Le \) Lewis number
- \( Nu \) Nusselt number
- \( Pr \) Prandtl number
- \( Re \) Reynolds number
- \( Re_t \) turbulent Reynolds number

**Miscellaneous**

- \( O \) Order of magnitude
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Samenvatting

Aardgas motoren worden zowel toegepast voor transportdoeleinden als voor het opwekken van energie (elektriciteit). Aardgasmotoren hebben voor veel toepassingen een wat lager rendement dan dieselmotoren. Gasmotoren hebben daarentegen ook specifieke voordelen. Door de hogere H/C verhouding van de brandstof zijn gasmotoren die zijn geoptimaliseerd tot het niveau van een vergelijkbare dieselmotor in staat lagere CO₂ emissies te behalen. Dit is van belang gezien de bijdrage van CO₂ aan het broeikaseffect. Een andere motivatie voor het gebruik van aardgas als motorbrandstof is de diversificatie van de brandstofvoorziening en de daarmee verminderde afhankelijkheid van olie. Heavy-duty gasmotoren worden veelal gebruikt met een vorm van mengselverdunning. Deze mengselverdunning kan men gerealiseerd door verdunning met extra lucht of door het bijmengen van uitlaatgassen. Bij de eerste vorm wordt gesproken van "lean-burn" motoren, de tweede vorm wordt vaak aangeduid met "lambda 1 + EGR". Door het gebruik van mengselverdunning worden smoorverliezen verminderd wat het rendement bij lage last ten goede komt. Door mengelverdunning daalt de maximaal optredende temperatuur tijdens de verbranding; dit leidt tot een vermindering van de vorming van stikstofoxide (NOₓ). Mengselverdunning kan men in de praktijk slechts beperkt toepassen. Bij te sterke mengselverdunning treden instabiliteiten op in de verbranding hetgeen leidt tot emissie van onverbrande brandstof en een onregelmatige motorloop. Door het gebruik van mengselverdunning daalt de verbrandingssnelheid van de brandstof-lucht mengsels. Dit wordt veelal gecompenseerd door de turbulentie in de verbrandingskamer te verhogen om zo een voldoende snelle omzet te bewerkstelligen. Veelal wordt hierbij aangenomen dat de verbranding lokaal plaatsvindt in een laminaire vorm (een "klassieke" vlam met een duidelijk scheidingsvlak tussen verbrand en onverbrand), en dat de globale omzettingsnauwkeurigheid van het brandstof-lucht mengsel kan worden gevonden door de lokale snelheid te vermenigvuldigen met het totale "vlam" oppervlak.

Bij het ontwerpen van motoren wordt tegenwoordig veel gebruik gemaakt van simulatiecodes. Voorbeelden hiervan zijn o.a. GT-power, AVL Boost en Ricardo Wave. Bij de toepassing van simulatiecodes is kennis nodig over het gedrag van het verbrandingsproces teneinde de warmtevrijstelling te kunnen berekenen. Veelal wordt hier de klassieke aanname gebruikt dat de verbranding wordt geïnitieerd door de bougie en dat de "vlam" zich vanuit deze bougie als een bolvorm uitbreidt. De snelheid van deze vlam, de turbulente verbrandingssnelheid S₇ relatif ten opzichte van het onverbrande mengsel wordt daarbij geschraald met de snelheidsfluctuaties welke geacht worden te schalen met het toerental van de motor. Deze aanname is relatief goed voor stoichiometrische mengsels; voor verdunnde mengsels daarentegen is deze aanname veel minder goed gevalideerd. Het voorspellend karakter van simulatiecodes is daarom minder sterk voor verdunnde mengsels.

In dit proefschrift wordt een onderzoek gepresenteerd waarbij het verbrandingsproces van deze verdunnde mengsels wordt onderzocht. De resultaten hiervan kunnen o.a. worden
gebruikt ter verbetering van de voorspellende capaciteiten van motorsimulatiecodes. De vragen die centraal staan in dit proefschrift zijn:

- Gaat de schaling van de turbulente verbrandingssnelheid op voor (zeer) verdunne mengsels?
- Mag de verbranding van zeer verdunne mengsels worden beschouwd als "klassieke" vlamvoortplanting?
- Is de aanname van een zich als een bolvorm uitbreidende vlam gerechtvaardigd voor zeer arme mengsels?

Dit onderzoek kan worden opgedeeld in een aantal delen die elk een aspect van het verbrandingsproces en de interactie met het stromingsveld in de motor beschouwen. Voor dit onderzoek is een speciale een-cilinder proefmotor gebouwd. Deze motor is voorzien van optische toegang tot de verbrandingskamer middels vensters in de zuiger en een deel van de cilinder. Met deze motor, maar met de vensters vervangen door metalen exemplaren is de grens van mengselverdunning opgezocht voor deze specifieke motor. De warmte vrijstelling werd daarbij bepaald uit de gemeten druk in de cilinder. Hieruit werd de turbulente verbrandingssnelheid bepaald middels de klassieke aanname van een sferische vlam. De laminaire verbrandingssnelheid van dezelfde brandstof-luchtmengsels (een eigenschap van het mengsel) werd bepaald middels de techniek van constant-volume verbranding in een separate opstelling. Hieruit is een correlatie afgeleid voor de laminaire verbrandingssnelheid als functie van samenstelling, druk en temperatuur. Deze data zijn samen met de gegevens uit de motor gebruikt om de verhouding tussen turbulent en laminaire verbrandingssnelheid te bepalen. Hieruit is gebleken dat deze verhouding ongeveer schaalt met het toerental van de motor, hetgeen de algemene aanname bevestigt.

Om een verdere, meer gedetailleerde uitspraak te kunnen doen over de verbranding in deze motor zijn het vlamvoortplantingsproces en ook het stromingsveld in de motor in kaart gebracht. Het verbrandingsregime (de manier waarop de verbranding lokaal plaatsvindt) wordt bepaald door lokale snelheden en lengteschalen. Het stromingsveld en de turbulent snelheidsfluctuaties in de verbrandingskamer zijn gemeten met PIV (Particle Image Velocimetry). Hiertoe was de motor uitgerust met vensters in zuiger en cilinder. Met deze opstelling zijn de turbulent snelheidsfluctuaties bepaald en ook de typische lengteschalen van het stromingsveld in de cilinder voor verschillende toerentallen en in-cilinder drukniveaus. Het verbrandingsregime is vervolgens gesitueerd in een zgn. "Borghi" diagram, daarbij gebruik makend van de gemeten snelheden en schalen van de stromingsveld en tevens van de laminaire verbrandingssnelheid zoals die is gemeten voor de gebruikte brandstof-luchtmengsels. Voor stoichiometrische (onverdunde) mengsels is gevonden dat het toepasselijke regime dat van "corrugated flamelets" is; voor sterk verdunne mengsels daarentegen is het "thin reaction zones" regime meer van toepassing. Voor beide gevallen kan men nog steeds spreken van klassieke vlamvoortplanting.

In dezelfde motoropstelling is vervolgens de vorm van de verbrandende zone gevisualiseerd middels Mie scattering aan kleine oiledruppeltjes. Hiermee kon een 2-dimensionale doorsnede van de 3-dimensionale vlam zichtbaar worden gemaakt. Hieruit kon worden afgeleid dat de aanname van een bolvormige vlam vrij goed geldig is voor stoichiometrische mengsels. Voor sterk verdunne mengsels daarentegen werd gevonden dat de vorm verre van bolvormig is. De "vlam" wordt daarbij uit elkaar getrokken en de verbranding vindt
plaats in een -op grote schaal- sterk vervormde 3-dimensionale structuur. In een poging om het omhullend oppervlak van de waargenomen verbrande zone (het vlamfront) te kwantificeren is gebruik gemaakt van fractal theorie. Hierbij werd gevonden dat het oppervlak bepaald middels fractal analyse, vermenigvuldigd met de laminaire verbrandingssnelheid niet volstaat om de omzettingssnelheid van het brandstof-lucht mengsel te beschrijven. Het verschil hiertussen, een factor 2-5, wordt daarbij toegeschreven aan de te beperkte informatie over de 3-dimensionale vorm van de vlam en het feit dat niet alle lengteschalen konden worden meegenomen in de analyse.
Dankwoord

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Bedankt allemaal!

Erik Doosje, juni 2010.
Curriculum Vitae

Erik Doosje was born in Blokzijl (the Netherlands) on September 9th 1973. He first attended secondary school (Athenaeum) but at some point switched to a technical education (MTS). In 1994 he started a Bachelor in Mechanical Engineering at Hogeschool Windesheim in Zwolle. After receiving his Bachelor of Mechanical Engineering in 1998 he started a study of Mechanical Engineering at Eindhoven University of Technology. During this study, he specialized in Thermofluids Engineering/Combustion Technology and for his final year opted for the group of Internal Combustion Engines. For an internship he worked on LIF diagnostics of the combustion of propane in a constant volume vessel. In 2000 he started his Master’s project on the characterization of flows induced by swirl ports on a heavy-duty diesel cylinder head on a stationary flow bench using PIV. In February 2002 he obtained his Master’s degree - cum laude. He then started his Ph.D. project in the same group of Internal Combustion Engines on ”Limits of mixture dilution in gas engines.” The project was initiated and partly sponsored by TNO automotive, resulting in this thesis. As of July 2008, Erik Doosje is working at TNO Automotive as a research engineer in the diesel emission control and sensing (DECS) group.