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IN

THERMO- AND FLUID DYNAMICS

Challenges and Advantages of Stratified Combustion in Gasoline Direct-Injected Engines

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Att doktorera är precis som älskande elefanter. Allt försiggår på ett högt plan, mycket damm virvlas upp, men man får vänta i åratal på resultatet.

Modified quote by Dag Hammarskjöld, secretary-general of the United Nations 1953-61

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Abstract

The modern world is based on an extensive transport network in which passenger vehicles play a major role. Although passenger vehicles have improved significantly in recent decades, they still contribute to the pollution of our environment and global warming. Consequently, new ways of reducing their emissions are needed. Moreover, most modern passenger vehicles are propelled by a combustion engine that operates on non-renewable fuels such as gasoline or diesel produced from crude oil. However, such fossil fuels are limited resources, so there is also a need to reduce the fuel consumption of passenger vehicles. As such, improvements in engine technology will play a central role in the development of more efficient and cleaner passenger vehicles. In recent years, alternatives to the internal combustion engine such as electric motors and fuel cells have attracted increasing attention. While these technologies are undergoing rapid development and electric vehicles in particular are gaining market share, the combustion engine remains the dominant power source for passenger vehicles. The most common type of combustion engine used in passenger vehicles is the gasoline engine. Several advanced combustion concepts have been developed to make gasoline engines cleaner and more efficient. One such concept is stratified combustion, which is discussed in this thesis. In gasoline engines, stratified combustion increases fuel efficiency. However, it also tends to produce high particulate emissions and can reduce combustion stability. These problems mainly occur because stratified combustion involves a complex mixing process that generates a heterogeneous air/fuel mixture with both rich and lean regions. This thesis describes work undertaken to minimize or eliminate these drawbacks, particularly the increased particulate emissions, while maintaining low fuel consumption.

Most of the studies presented herein were performed with metal and optical single-cylinder engines, but a four-cylinder production engine was also used in some cases. Tests were performed in steady state mode at various engine operating points. All engines utilized in the studies were fitted with a Spray-Guided Direct-Injected (SGDI) system and multi-hole solenoid-actuated fuel injectors. Depending on scope of the study, the engines were equipped with different measurement devices such as instruments for measuring pressures and temperatures or emissions of HC, NOx, CO, CO_2, O_2 and particulates (both mass and number).

The results obtained during this thesis work have been presented in five publications. The first of these publications describes a study on the relationship between particulate emissions during stratified combustion and generic combustion variables such as the fuel injection pressure, as well as injection and ignition timings. This was done to identify variables that could be manipulated to reduce particulate emissions. The later publications describe how measured particulate emissions are affected by forced induction, increased fuel injection pressure, the use of novel ignition systems, air movements, and the use of different sampling systems.

Key objectives of these studies were to find ways of reducing particulate emissions and increasing combustion stability. It was found that stratified combustion in SGDI gasoline-fueled engines fitted with a solenoid multi-hole injector can increase fuel efficiency but does not alleviate the problem of high particulate emissions. In addition, a positive correlation between the extent of non-premixed flames and particulate mass and number emissions was identified. The use of novel ignition systems was shown to expand the ignition window, while boosting and increasing the fuel injection pressure were found to reduce soot levels. Finally, the internal relationship between the ignition and injection timings was found to strongly affect combustion stability and soot levels, primarily because it influenced the dilution of the air-fuel mixture and the risk of the fuel spray striking the piston top.

The thesis concludes with some suggestions for ways of improving stratified combustion and directions for future research.

Keywords: Particulates, Particulate Mass, Particulate Number, Particulate Sampling, Spray-Guided, Direct-Injected, Gasoline, Stratified Charge, SGDI, GDI, DI

List of Publications

Some of the contents of this thesis are based on the studies reported in the following papers (attached to the thesis):

1. **Johansson, A.,** Hemdal, S., Dahlander, P., ”Experimental Investigation of Soot in a Spray-Guided Single Cylinder GDI-Engine Operated In a Stratified Mode”, SAE Technical Paper 2013-24-0052, 2013, doi: 10.4271/2013-24-0052

2. **Johansson, A.,** Hemdal, S., Dahlander, P., “MEASUREMENTS OF PARTICULATE SIZE DISTRIBUTION FROM A GDI ENGINE USING A NAFION DRYER AND A DMS500 WITHOUT SAMPLE DILUTION", Fisita World Automotive Congress F2014-CET-131, 2014

3. **Johansson, A.,** Dahlander, P., “Experimental Investigation on the Influence of Boost on Emissions and Combustion in an SGDI-Engine Operated in Stratified Mode”, SAE Technical Paper 2015-24-2433, 2015, doi: 10.4271/2015-24-2433

4. **Johansson A.,** Dahlander, P., “Experimental Investigation on the Influence of Boost on Combustion and Particulate Emissions in Optical and Metal SGDI-Engines Operated in Stratified Mode”, SAE International Journal of Engines 2016-01-0714, 2016, doi: 10.4271/2016-01-0714

5. **Johansson A.,** Hemdal, S., Dahlander P., ”Reduction of Soot Formation in an Optical Single-Cylinder Gasoline Direct-Injected Engine Operated in Stratified Mode using 350 bar Fuel Injection Pressure, Dual Coil- and High Frequency-Ignition Systems”, Manuscript submitted to the journal SAE International Journal of Engines in November 2016

Acknowledgements

Sometimes things do not go as planned. As a ninth-grade student, I had a clear plan: I would study electronics in the gymnasium and then mechanics at university so as to understand both subjects well. During my subsequent master's degree studies at Chalmers University of Technology, I decided that I wanted to work for an automotive manufacturer located slightly north of Gothenburg. I did my master's thesis at Saab Automobile and had the honor of being employed there after graduating. Unfortunately, around one month after I started work, the company suffered a financial crisis and went bankrupt – I can only hope that the timing of the two events was coincidental. For the first time I had no plan and so started to look for new positions. I was offered a few assignments but the only one that felt right was an opportunity to study for a PhD in the combustion division at Chalmers University of Technology. I am deeply grateful to my examiner Professor Ingemar Denbratt and my supervisors Docent Petter Dahlander and Doctor Stina Hemdal for giving me this opportunity. During my time at Chalmers, I have met many people who have helped me on my way – so many that I will probably forget to mention some here despite my gratitude! I will first thank Professor Sven Andersson for all the inspiration he has given me during the various projects we conducted together at Chalmers. I also thank Professor Mikael Enelund for having faith in the idea of acquiring a vehicle simulator for the school of mechanical engineering. I and many students appreciate your valuable contribution to the educational program. Special thanks are owed to Gerben Doornbos, who I worked with during most of my time at Chalmers. In fact, we probably spent a bit too much time together conducting experiments with the engine rigs! Finally, without Elenor and Ulla the division would work far less smoothly than it does - they too deserve special thanks.

Without my colleagues in the engine lab the work described in this thesis would have been substantially harder, so I owe special thanks to Eugenio De Benito Sienes, Daniel Härensten and Anders Mattsson for their valuable assistance. I am also indebted to the colleagues who contribute to good working environment and all the students I have worked with, especially Håkan for his fantastic work on the simulator and all the projects we have undertaken together.

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Abbreviations and nomenclature

ATDC	After Top Dead Center
BMEP	Brake Mean Effective Pressure
BTDC	Before Top Dead Center
CAD	Crank Angle Degrees
CO	Carbon Monoxide
CO_2	Carbon Dioxide
DCI	Dual Coil Ignition
DI	Direct Injection / Injected
HC	Hydrocarbons
HCCI	Homogenous Charge Compression Ignition
HFI	High Frequency Ignition
HLCSI	Homogenous Lean Charge Spark Ignition
IMEP	Indicated Mean Effective Pressure
LIF	Laser Induced Fluorescence
LII	Laser Induced Incandescence
MFB	Mass Fraction Burned
NO_x	Nitrogen Oxides
O_2	Dioxygen
PIV	Particle Image Velocimetry
PM	Particulate Mass
PMP	Particulate Measurement Program
PN	Particulate Number
RPM	Revolutions Per Minute
SGDI	Spray-Guided Direct-Injected
TDC	Top Dead Center – The position of the piston when it is farthest from the crankshaft
VPR	Volatile Particle Remover

Contents

1 Introduction

Combustion engines have been the dominant power plants in vehicles during the 20th century. However, since the start of the 21st century there has been a shift towards alternative power sources. Although electric powertrains are gaining market share, the combustion engine is expected to play a major role for the foreseeable future. To remain competitive, modern combustion engines must be as inexpensive as possible while maintaining low fuel consumption, low emissions, and appealing power characteristics. The goal of this thesis was to find new ways of creating engines that satisfy these requirements more fully than existing options. The requirement for *low fuel consumption* primarily comes from market demands such as direct requests from end users, but is also driven by issues such as vehicle taxation based on CO_2 emissions. The requirement for *low emissions* derives primarily from legislation such as the emissions standards that have been introduced in many parts of the world. For example, the first European emissions standards were introduced in 1992, and the sixth version of these standards (Euro 6) came into effect in September 2016 for light passenger and commercial vehicles. Each new iteration of the regulations has imposed stricter standards on emissions of pollutants such as hydrocarbons, carbon monoxide and nitrogen oxides. The Euro 5 standards introduced regulations on particulate emissions as well: compression-ignited engines were required to adhere to limitations on particulate mass and number emissions, whereas spark-ignited engines were only subject to particulate mass emission limits. The more recent Euro 6 standards impose even tighter restrictions on particulate emissions from spark-ignited engines by adding a limit on particle number emissions to the previously established mass limits. Finally, the requirement for *appealing power characteristics* stems from the way in which drivers perceive cars. To satisfy this requirement, an engine should be able to operate over a large speed range and provide adequate torque at all points within this range. This is normally referred to as driveability. Efforts to reduce fuel consumption and emissions have spurred the development of new engine technologies, which may have to be combined in new ways to satisfy drivers' power expectations. One such new technology is stratified combustion, which can be utilized in Direct-Injected (DI) gasoline engines. This technology forms the basis of the work presented in this thesis.

1.1 Objectives

The objectives of this work were to find ways of improving stratified combustion, an alternative to the traditional homogenous combustion strategy used in four-stroke gasoline engines, which are the most common engine type in passenger vehicles. To this end, a series of experiments were performed using SGDI gasoline engines. State of the art stratified combustion is used at low to moderate loads and engine speeds. It is difficult to use at high loads and speeds, so an engine using stratified combustion must also be able to operate in homogenous mode. This project therefore utilized engines that were optimized for homogenous combustion. A key objective was to find ways of achieving low fuel consumption while avoiding the increased particulate emissions typically observed during stratified combustion. Using information drawn from the literature and results obtained during the experimental campaign, the following questions were addressed:

- What are the main characteristics of particulate emissions from engines operated in stratified mode?
- How can these particulate emissions be accurately measured?
- What are the main differences between particulate emissions emitted from engines operated in the homogenous and stratified modes?
- How are soot emissions and combustion characteristics affected by changing the following engine components or operating parameters:
    - Ignition systems
    - Forced induction
    - Exhaust gas recirculation
    - Fuel injection pressures
    - Turbulence levels
    - Engine load and speed

1.2 Limitations

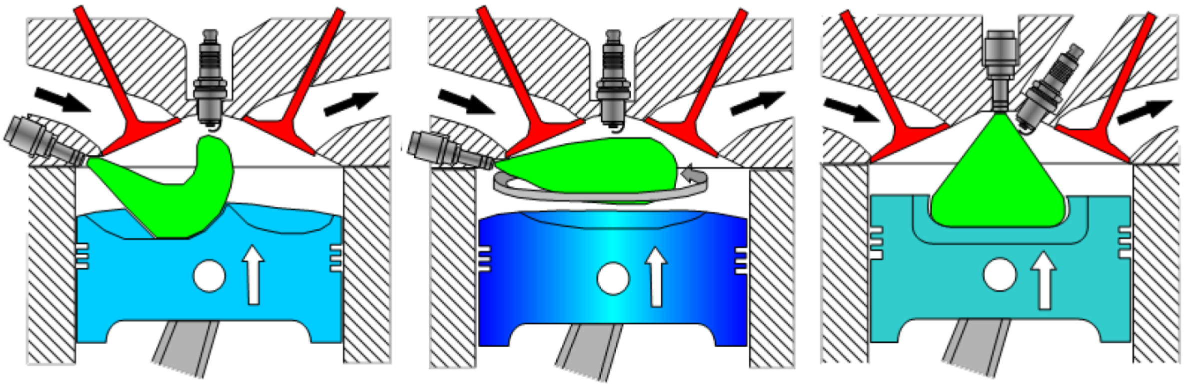
The project focused solely on combustion and engine-out emissions; the use of exhaust after-treatment was not examined at all. Additionally, some engine technologies that may be useful for reducing particulate emissions and enhancing combustion, such as piezo injectors, were not considered in this work because extensive information on their capabilities and usefulness was already available when the studies reported herein were conducted.

2 Background

Contrary to the common belief that Nicolaus Otto was the first inventor of the four-stroke engine, recent research has shown that it was patented in London as early as in 1854 by the Italian inventor Eugenio Barsanti and the Italian hydraulic engineer Felice Matteucci [68]. The four-stroke engine is based on a piston, a cylinder, an associated crankshaft mechanism, and a gas exchange system with an igniting mechanism [35]. Stored energy from the fuel is transformed into mechanical power in four phases, the so-called four strokes. It is assumed that the reader has at least a basic understanding of the four strokes and the operation of a four-stroke engine, so these topics are not discussed further in this thesis. The four strokes have remained relatively unchanged since their invention. However, there have been major changes in the way fuel is delivered to the combustion chamber. For a long time, *carburetors* were the most common devices used to mix fuel and air. A carburetor is mounted upstream of the intake channels and delivers fuel into a flow of air to form a homogenous mixture. In the late 1980s, mechanically or electronically controlled injectors started to replace carburetors. A carburetor can be replaced by a single injector mounted in the same position as the carburetor or by a set of injectors mounted a short way upstream of each cylinder's intake valve to create a so-called *port injection* system. Systems based on separate injectors yield better control over fuel mixing than is possible with carburetors, and the introduction of emission regulations in 1993 caused a rapid decline in the market share of carburetors across Europe. Port injected engines dominated the market for more than two decades, but have recently faced strong competition from *DI* engines. In a DI engine, the injectors are placed downstream of the intake valve rather than upstream, so they inject fuel directly into the combustion chamber. This has thermodynamic benefits and enables even greater control over fuel mixing. Importantly, it also enables the engine to be operated using combustion regimes other than the homogeneous mode that has been dominant since the four-stroke engine was invented.

2.1 Direct Injection

Direct injection technologies for four stroke engines can be subdivided into three different classes depending on how the fuel is mixed or directed inside the cylinder when operating in stratified mode. These classes are referred to as Wall-Guided, Air-Guided and Spray-Guided, and are illustrated in Figure 1.



a) Wall-Guided ***b) Air-Guided*** ***c) Spray-Guided***

Figure 1 - Illustration of the three dominant Direct Injection technologies. Image courtesy of S. Sloboda [58].

The *Wall-Guided* technique, which is depicted in Figure 1a, is the oldest of the three technologies. It involves directing the injected fuel towards a bowl in the piston, which is shaped such that the fuel spray is directed towards the spark plug. In principle, the spray should remain relatively focused without deteriorating or mixing too much with the air. A certain amount of mixing is desirable but not to the extent that the mixture becomes over-leaned and falls outside its ignitability limit. However, too little air dilution will cause under-mixing, which will result in heavy soot

Figure 2 - Image of combustion chamber and channels of a Mitsubishi GDI-engine operated without proper maintenance. Image courtesy of [59].

production and misfires in the worst cases. This phenomenon has been thoroughly studied by Sandquist et al. [67]. The technique therefore depends on striking a fine balance between over- and under-mixing to avoid extensive soot production or misfires. Injection must occur when the piston is in a specific position, so the spray must be injected within the same CAD interval independently of the engine speed, leaving little room for calibration and combustion phasing. The technique was commercialized in Mitsubishi's four-cylinder inline 4G93 engine for passenger vehicles in the late 1990s. These engines produced high soot emissions and had to be maintained carefully to avoid soot fouling. They were eventually discontinued, partly because of their high soot emissions. Figure 2 shows the combustion chamber of one of these engines; both the chamber and the channels are covered with a thick coating of soot that prevents the engine from functioning properly. This particular engine was operated for a long time without proper maintenance. However, even without regular maintenance, a regular DI gasoline engine operated under homogenous conditions would not be so badly contaminated. This image illustrates the problems of stratified combustion and the associated soot production. The major issue with the wall-guiding technique is that fuel is directed towards a surface where it may not evaporate as quickly as needed for efficient combustion. Unevaporated fuel presents a risk of incomplete combustion and has a high risk of forming soot [27]. The process of wall-guiding is difficult to control because of the diversity of operating conditions that occur in a combustion engine under the different load cases generated by normal driving. Some of the soot that remains after combustion has stopped will be emitted in the exhaust, but some will be deposited in the combustion chamber, valves, valve-guides, and exhaust system components including the silencers and catalysts. Thus, soot buildups can cause significant engine fouling if not taken care of properly.

The *Air-Guided* process illustrated in Figure 1b was commercialized by Volkswagen in the year 2000 with the introduction of the FSI/TFSI-engine. This engine design has been updated several times and remains in production using the Air-Guided approach. The fuel is injected into an air flow that is directed towards the spark plug. The injection timing must therefore be matched with the air flow. This design seems to have more potential than the Wall-Guided design because the fuel is not directed towards a surface from which it cannot evaporate fully. However, it is still difficult to create a fuel cloud within the ignitability limits in the vicinity of the spark plug due to cyclic variations in the air flow and fuel spray structure. Consequently, the fuel clouds are inhomogeneous, which could cause misfires in the worst cases. Although the technique has more potential than the Wall-Guided system, Volkswagen uses it only at low loads and engine speeds. This limitation is typical for stratified combustion: no engine currently in production uses stratified operation across its entire speed and load ranges. Because of this limitation, the engine must operate in other modes, preferably homogeneous, at high loads and speeds [2] [3] [12].

The third and newest technique for stratified operation is the S*pray-Guided* technique, which is depicted in Figure 1c. In Spray-Guided systems, the fuel spray created by the injector is directed towards the spark plug. This is achieved by mounting the spark plug and injector in close proximity, and engineering the injector and its placement so it injects the fuel spray in the desired direction. This makes the system's operation less timing-dependent than that of Wall- and Air-Guided systems. Spray-Guided systems are also less dependent on the piston position and airflow than Wall- or Air-Guided systems. However, if the injection timing is too late, the spray will probably impact on the piston crown, causing similar problems to those created by the Wall-Guided design. Spray-Guided engines have been commercialized by both BMW and Daimler, but BMW discontinued their engine; as of the time of writing, Daimler seems to be the only company producing a Spray-Guided engine that operates in true stratified mode [3]. Most new DI-engines such as the new VEP architecture from Volvo Cars are designed for Spray-Guided operation but do not utilize stratified operation [4], except for a short period during cold start. All results presented in this thesis are from experiments performed with Spray-Guided engines.

2.2 Stratified combustion

Most modern combustion engines utilize homogenous operation. Homogenous operation is achieved by injecting early during the intake stroke so the fuel can evaporate and mix homogenously with the air. This can be achieved with both port fuel injected and DI engines. The engine load is controlled by the amount of fuel injected in each cycle. To keep the mixture within its ignitability limit (which is determined by the injected fuel mass), the air mass must be controlled to maintain a suitable air to fuel ratio. This is generally achieved by using a throttle or a variable intake valve to restrict the intake air flow. Most engines operate at a stoichiometric or near-stoichiometric air to fuel ratio. The restriction of air flow is the major contributor to pump-losses, which directly increase fuel consumption. It is therefore desirable to reduce these losses. This can be done by leaning out the air to fuel ratio, i.e. by allowing more air-mass than necessary to enter the combustion chamber. However, above a certain lean air to fuel ratio, the mixture will exceed its ignitability limit, leading to misfires or partial burns. To lean out the mixture beyond this point, it would be necessary to either increase the fuel's ignitability or employ a different combustion mode such as stratified operation, which eliminates pumping losses over the air mass restrictor; in the simplest stratified combustion systems, the restrictor is removed entirely. However, this presents a problem: injecting the specified amount of fuel at such lean air to fuel ratios with the same injection timing as used in homogenous mode would cause over-dilution of the fuel and thus misfires. It is therefore necessary to concentrate the fuel around the spark plug by some means to create an ignitable cloud around the spark. This is achieved by injecting late in the compression stroke and igniting the mixture before the fuel cloud has become excessively diluted. In practice, injection occurs between 50 CAD BTDC and 20 CAD ATDC, and ignition happens a few CAD after injection. In principle, this will produce an ignitable cloud that is close to the sparkplug and surrounded by air or recirculated exhaust gases. There are three main techniques for guiding the fuel to achieve this, which are discussed in detail in section 2.1 Direct Injection. In addition, hybrid techniques that combine elements of the main techniques are possible, albeit less common.

Although nowadays stratified combustion is exclusively utilized in DI engines, some attempts have been made to achieve stratified operation in port-injected engines. In 1975, Honda released their CVCC (Compound Vortex Controlled Combustion) engine, which utilized a pre-chamber design that can be regarded as a semi-stratified system based on modern definitions. The system utilized two mixtures: a homogenous lean mixture distributed across the entire combustion chamber and a richer mixture that was injected via a pre-chamber and directed towards the spark plug. Additionally, in the 1970s Ford tested their PROCO concept - the Ford Programmable Combustion Engine. This was one of the first practical applications of Spray-Guided stratified operation in a somewhat modern DI gasoline engine. The rapid development of DI engine technology since those early days, together with the advent of simulations and mechanical tools has made it much easier to implement stratified operation.

2.2.1 Advantages

In general, stratified operation has four major advantages over homogeneous operation: reduced pumping losses, reduced wall heat losses [54], increased volumetric efficiency, and increased charge cooling. Only air is compressed during the compression stroke, so engines using stratified combustion achieve a higher specific heat ratio (κ~1.4) than for homogenous combustion, in which compression work is done on both the air and the fuel (κ~1.3). An engine operated in stratified mode reduces the need for a throttle to restrict air flow. As noted above, this significantly reduces pumping losses and thus fuel consumption. In practice, a stratified engine may use a throttle to create a pressure drop to allow for exhaust gas recirculation. However, the resulting pressure drop over the throttle is compensated by the EGR flow and does not directly affect fuel consumption. Wall heat losses are reduced because the ignitable air-fuel cloud is (in principle) surrounded by air. This produces a flame that is also surrounded by an insulating layer of

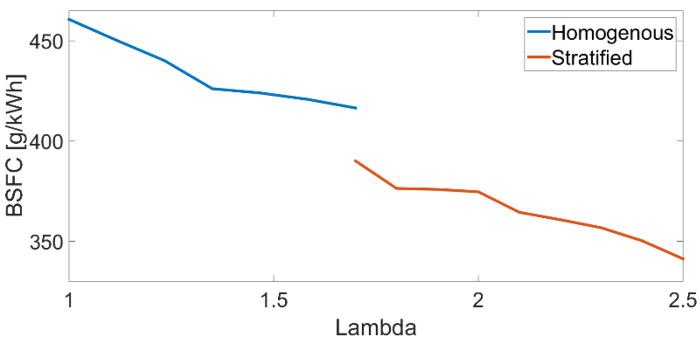


Figure 3 - Fuel consumption benefits of stratified operation compared to homogeneous operation in a research engine operating at 2000 rpm and 2 bar BMEP.

air or recirculated exhaust that prevents heat losses through the cylinder wall. Consequently, less of the heat generated during combustion is lost to the walls and more is used for expansion work. Increased volumetric efficiency is achieved because all the fuel can be injected after the intake valve has closed, maximizing the mass of admitted air. Because the fuel is injected after intake valve closure, its evaporation is driven by the heat of the air inside the cylinder and thus reduces the in-cylinder temperature. Together, these factors reduce fuel consumption. Stratified operating allows the engine to operate with higher compression ratio, mainly since the risk of knock is lowered due to non-existing or lean end gases. The increased compression ratio in turn, leads to even higher efficiency. Another advantage of stratified operation is the increased mass flow which aid spool up time for assessor pumping devices such as turbos. Figure 3 shows the reduction in fuel consumption for stratified operation compared to homogeneous operation. The fuel consumption in homogenous mode (shown by the blue line) ranges from roughly 480 g/kWh for a stoichiometric mixture to 420 g/kWh under lean homogeneous conditions (lambda = 1.7). This corresponds to the system's ignitability limit; the mixture cannot be leaned out further in homogenous mode. The brown curve representing stratified operation starts at lambda = 1.7 and a fuel consumption of roughly 390 g/kWh. With the throttle fully open under stratified conditions, the lambda value reaches 2.5 and the fuel consumption is 340 g/kWh. These results were obtained with a single cylinder research engine that has rather high internal friction. As a result, the BSFC values for both combustion modes are somewhat high. However, the trend for stratified combustion to provide greater fuel efficiency than homogeneous combustion is also observed in real operation. There have been few publications on engines developed specifically for stratified mode, but one was presented in 2012 [2] by the English company Ricardo, which has developed such an engine. That engine reached a lowest BSFC value of 206 g/kWh and thus achieved similar efficiency to a modern diesel engine. At low load (2 bar BMEP and 2000 rpm), the engine operated at 277.5 g/kWh in stratified mode compared to 370 g/kWh in stoichiometric homogeneous mode. This difference is substantial and clearly demonstrates the advantages of stratified combustion with respect to fuel consumption. Stratified operation with natural aspiration is impractical because it would make the engine's load limit too low. However, turbocharging [8] makes it possible to extend the load limit while reducing particulate emissions [9] [10] [11]. Figure 4 presents an inexact illustration of the typical operating areas for the various modes.

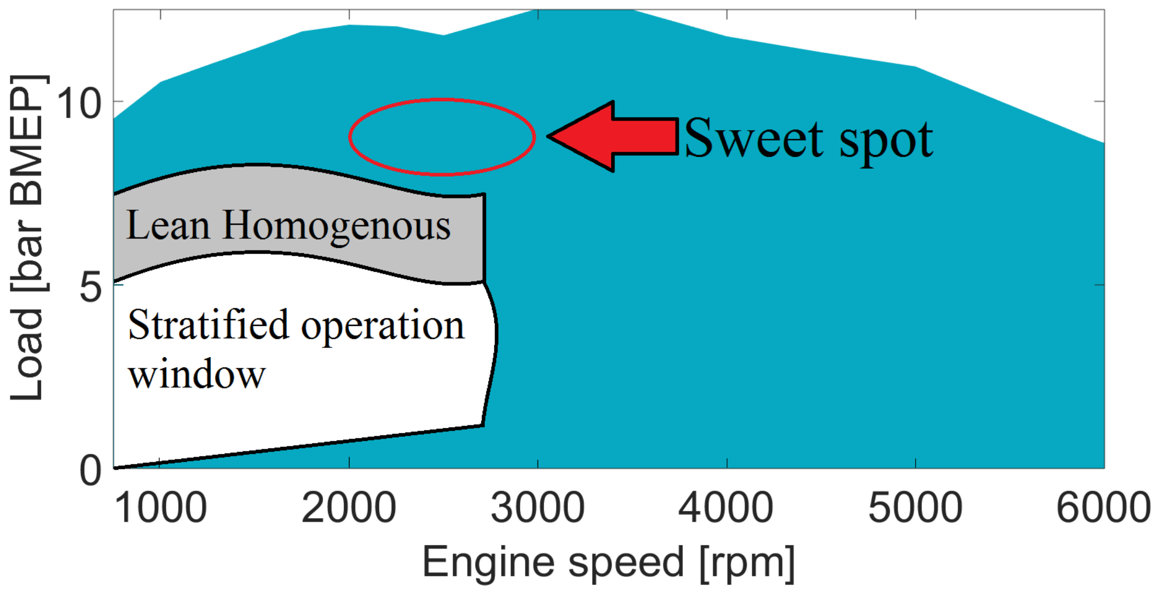


Figure 4 - Areas of operation for stoichiometric homogenous, lean homogeneous and stratified combustion.

The graph represents a typical gasoline engine's total operating window, with the x-axis showing the engine speed in revolutions per minute and the y-axis showing the load in bar. The blue area represents the engine's viable operating window. At 2500 rpm and around 8 bar BMEP, there is an ellipse representing a generic sweet spot region of most current engines. A sweet spot region is the operating window in which a gasoline engine operates most efficiently, i.e. where the fuel consumption is lowest in relation to the vehicle's power output. The sweet spot occurs at high load because high loads reduce throttling losses, and at an engine speed where the gas exchange dynamics favor high volumetric efficiency. However, the sweet spot is not generally the preferred region for operating a gasoline engine. Indeed, normal-sized engines rarely operate at this point during typical driving, which usually involves low or moderate loads and engine speeds. There are ways to shift the sweet spot into a more useful region, such as downsizing. Alternatively, the engine could be operated in stratified mode, which achieves similar fuel efficiency to that attained in the sweet spot. In addition to the blue region, which shows results for normal stoichiometric homogenous operation, the image shows results for two alternative operating regimes - stratified and lean homogenous operation. The operating windows are shown for illustrative purposes and their boundaries are inexact, but they show the general areas in which these two operating regimes are utilized. Importantly, these areas correspond to typical engine operating conditions: few drivers run their engines above 3000 rpm during daily driving. Consequently, reducing fuel consumption in these regions can be very beneficial.

As mentioned, downsizing is one way of shifting the sweet spot to more frequently used regions. During the writing of this thesis, downsizing has been the most popular way besides hybridization to achieve lower fuel consumption. However, downsizing comes with the disadvantage of higher fuel consumption in the area above the sweet spot, in terms of both load and speed, than for a corresponding normal-sized engine. The load related increase in fuel consumption is mainly due to increased knock sensitivity whereas the engine speed related increase in fuel consumption is mainly due to increased friction and pumping losses. At the end of this project several manufacturers started to once again develop normal sized engines for hybrid use where the area of high fuel consumption below the sweet spot region was addressed with an electric motor. In these engines the area of high load is not an issue to the same extent and the region where stratified operation is most beneficial for these engine types is also the region where the electric motor is utilized. Hybridization with electric motors in this area of operation is indeed a competitor to stratified operation. It is worth though to keep in mind that there might be cases where the electric motor cannot be used. Examples of such cases are operation during low temperature, discharged battery or hardware malfunctions.

2.2.2 Disadvantages

The greatest advantage of stratified combustion is that it reduces fuel consumption. However, stratified combustion has several challenges [22] [23]. The biggest drawbacks of stratified combustion are its tendency to produce high particulate emissions, reduce combustion stability, and interfere with exhaust aftertreatment. This thesis focuses on the combustion and particulate emissions; aftertreatment systems are not considered. However, in general no type of lean combustion is compatible with conventional three-way catalysts because lean combustion produces exhaust gas mixtures containing excess oxygen. Consequently, the three-way catalyst must be complemented with systems such as NOx traps or a Selective Catalytic Reduction apparatus. DI engines generally produce higher particulate emissions than comparable port injected engines. If the DI engine is operated in stratified mode, its particulate emissions may be over 100 times greater than they would be in homogenous mode [15]. The major reasons for this are discussed at length in the chapter on particulate emissions. Successful stratified combustion is also highly dependent on the mixture formation process: due to cyclic variation in air-flow and spark development, the fuel mixture will not form in exactly the same way in different combustion cycles, which may create ignitability issues.

In most cases, stratified combustion is only used at low to medium loads and speeds. If the engine speed is too high there will not be enough time for mixture formation or the turbulence level is too high, so the engine is normally switched to homogenous mode once the speed exceeds the upper limit for stratified mode [12]. The upper load limit is normally restricted by incomplete combustion. As the load increases, so does the injected fuel mass. If there is not sufficient time for complete oxidation of the injected fuel, combustion will be incomplete, leading to higher soot emissions and potentially increased HC emissions as well. On the other hand, if the injected fuel mass is too small, there is a risk of over-dilution. This is especially problematic at higher engine speeds where the turbulence level increases. The combination of high speed and low load is therefore more challenging than that of low load and low speed.

2.3 Different stratified operating modes

This thesis focuses on the conventional stratified operating mode, which uses a single injection event close to the ignition event. Other stratified modes are possible such as modes which involves a rapid series of injections performed close to the ignition event or modes which involves a more widely spaced series of injections occurring during the intake and compression strokes or combinations thereof. These modes are sometimes referred to as micro- or semi-stratification. Dividing the injection event into several smaller injections can improve mixing and reduce the spray penetration length, which may help to reduce particulate emissions and increase efficiency. Some stratified modes create lean homogenous mixtures by injecting early in the intake stroke. These mixtures are normally too lean to be ignited by a conventional spark plug, so a small injection is performed closer to the ignition to create a small stratified charge that assists the ignition of the main homogeneous mixture. Theoretically, these modes combine the advantages of homogenous (low particulate emissions) and stratified (low pumping losses) combustion. Conventional solenoid injectors open and close relatively slowly and so cannot usually perform multiple rapid fuel injection events. Therefore, modes that require such injections must be implemented using piezo injectors.

2.4 Alternatives to stratified operation

Several alternatives to stratified operation exist, each with their own advantages and drawbacks. This section describes the most common alternatives.

2.4.1 Lean homogeneous

Lean homogenous combustion, or Homogenous Lean Charge Spark Ignition (HLCSI) as it is sometimes referred to, is a field that has seen a lot of recent development. It uses air-fuel ratios that are leaner than stoichiometric to reduce the pumping losses associated with restricted air flow. It also has relatively low combustion temperatures, which reduces exhaust and cooling losses. Lean homogeneous combustion has been achieved at lambda values of up to 2 [37]. This mode has a favorable engine-out emissions profile: as long as the combustion remains stable, it yields low CO and HC emissions [38]. However, NOx emissions peak at a lambda value slightly above 1.1 and decrease above this point. Engine-out NOx emissions depend on two factors: the air mass and the temperature. Up to a lambda value of slightly above 1.1, increasing the air mass enhances NOx production, but above this point the cooling effect of the extra air mass becomes dominant and reduces NOx formation [35]. Lean combustion is “less forgiving” on the ignition system than stoichiometric combustion, and therefore benefits from the use of specialized ignition systems in terms of increased combustion stability and reduced combustion duration [39].

2.4.2 HCCI and SCCI

Homogenous Charge Compression Ignition (HCCI) utilizes an early injection to create a homogenous mixture that is ignited by compression ignition, as in diesel engines. The mixture should therefore not require a conventional ignition system. Compression ignition is impossible in an engine not designed for operation in this manner, such as a conventional SI engine. However, it is possible to create conditions favorable for self-ignition by increasing the compression ratio, pre-heating the induction gases, forced induction or exhaust gas recirculation [40]. The advantages of HCCI include low particulate emissions, fuel type flexibility, high efficiency and low NOx emissions. The latter two advantages are partly due to its high combustion speed. However, this rapid combustion also causes so-called “ringing phenomena”, i.e. pressure oscillations that create noise [43]. Ringing can be suppressed by increasing the combustion duration. One way of doing this is to perform an injection late in the compression phase in addition to the main injection during the intake phase. This reduces the rate of pressure rise and thus suppresses ringing [41] [42]. Another challenge in HCCI combustion relates to combustion phasing: there is no way to precisely control ignition in HCCI, so combustion phasing is difficult to control.

2.5 Emissions legislation

States have created a variety of laws and regulations to limit vehicles’ emissions. Such laws have a long history: the Motor Vehicle Air Pollution Control Act was passed in the USA in 1965 and the Japanese Air Pollution Act was introduced in 1968. Emissions legislation became more prevalent in the 1990s with the introduction of the European emission standards among others. As of today, there are three major regions of the world that are setting standards for emissions legislation: Europe, Japan and USA [36]. At the time of writing, emissions in these regions were governed by the Euro 6, Post New Long Term, and Tier 2 regulations, respectively. The USA has also published its Tier 3 regulations, which will be phased in from 2017. Modern emissions regulations specify the maximum quantities of different species a combustion engine may emit, and dictate how these emissions should be measured. The most heavily targeted emissions are HC, CO, CO2, NOx, PM and PN. Many regulations also impose requirements concerning emissions of sulfur and evaporative species, as well as the vehicle's self-diagnostic capabilities, but this thesis focuses on the first six pollutants mentioned above. Of these, particulate mass (PM) is regulated by all major emissions standards, but particle number (PN) is currently only regulated by the European standards. The particulate number limit of $6x10^{12}$ applies from September 2014 until September 2017 where the limit is reduced to $6x10^{11}$. While no limits on PN emissions are currently planned in US legislation, it is possible that by 2020 both China and India will have adopted limits on PM and PN emissions similar to those specified by the Euro 6 regulations. Small particles present a threat to human health, and can be subdivided into two size classes: PM2.5 comprising particles of less than 2.5 μm, and PM10 comprising particles of less than 10 μm. PM2.5 are considered to be the most dangerous because they become airborne more readily, are easily trapped in human tissues (which can cause problems in the airways), and are cancerogenic. Most of the particles emitted from gasoline engines are below 500 nm and thus belong to the PM2.5 class. It is therefore important to reduce PN emissions, and the European emissions standards are currently the most stringent in the world in terms of PM/PN.

2.6 Particulate emissions

Figure 5 – Berlin cathedral with soot-contamination, photographed in mid-September 2016. The two towers show clear contamination.

Particulates have attracted increasing interest and concern in recent years. This concern has been spurred by increasing smog levels in major cities around the world and a recently identified connection between small particulates and human health [17]. Smog levels in densely populated cities have increased and several cities have introduced bans on combustion vehicles. Most of these bans target vehicles with diesel engines because they have been identified as major sources of particulate emissions, although DI gasoline engines also emit smaller quantities of PM. Figure 5 shows an image captured in September 2016 of a building in Berlin whose upper structures exhibit severe soot contamination due to vehicular PM emissions. Additionally, Paris banned vehicles with diesel engines from the city for several days in 2014 and 2015. This reduced smog so much that tighter restrictions were introduced: as of July 2016, no car made before 1997 may be driven in inner Paris on a weekday. Several other countries have also passed laws to reduce particulate emissions: Germany has a three-tier system (red, yellow, and green) for classifying vehicles based on their emission levels, and Norway has banned diesel vehicles from being driven in inner Oslo when smog levels are high. Besides soiling buildings and reducing visibility, airborne particles cause health problems. Several studies [32] [33] [34] have found a relationship between nanoparticles and increased health risks such as issues with airways and an increased risk of cancer. Even substances that are harmless as µm-scale particles, such as TiO_2 [47] and Teflon [48], have adverse effects on human health when scaled down to the nm range. This is partly because smaller particles are more easily deposited in human tissue such as lungs and airways. A particle of 10 nm is considered ultrafine and is typically airborne for roughly 15 minutes. Given the number of vehicles in typical cities and the fact that a Euro 6 vehicle emits around $6*10^{12}$ particles/km, a typical city-dwelling human will inevitably inhale large numbers of potentially harmful particles.

To illustrate the size distribution of the particles generated during combustion and relate them to a more familiar source of particulate emissions, Figure 6 shows particulate sampling diagrams for stratified and homogeneous combustion, and cigarette smoke.

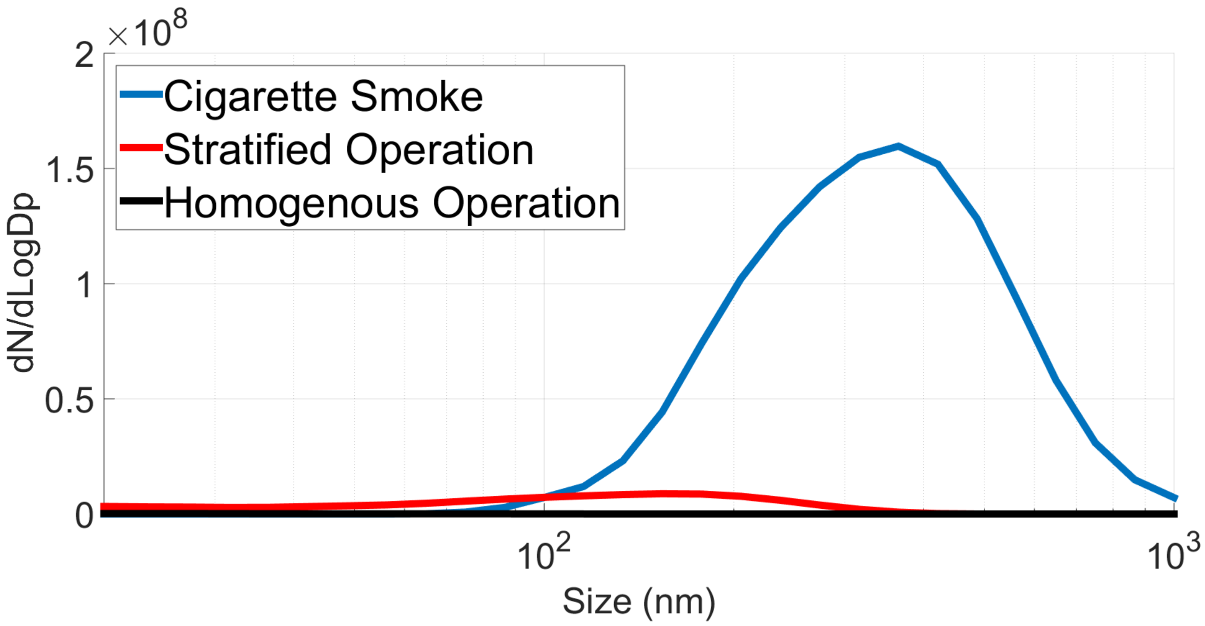


Figure 6 - Particulate Size Distribution from an unnamed commercial cigarette versus particulate emissions generated by an engine operated in homogenous mode while idling at 1000 rpm and 1.5 bar BMEP.

The x-axis of this figure represents the particle size in nm and the y-axis represents the number of particles emitted. The blue curve shows data for smoke sampled from a normal cigarette when smoked, the black curve shows results for an idling SGDI engine operated in stoichiometric homogenous mode, and the red curve shows particulate emissions from the same engine when operating in stratified mode. The cigarette smoke produces substantially more particulates than the engine in either mode. Notably, the curve for cigarette smoke features a substantial peak at around 3-400 nm while the curve for homogeneous combustion appears to lie on the x-axis. The red curve representing stratified combustion shows somewhat higher particulate emissions but without the large peak seen for cigarette smoke. It should be noted that increasing the engine load would substantially increase the particulate emissions for stratified mode. Although the particulates emitted from the cigarette are larger than their counterparts from the engine, they are still well within the size range that favors accumulation in human tissues. Particulates from cigarettes also have different chemical compositions to those from engines, so they are not fully comparable, but the graph illustrates their relative impact. As stated earlier in this thesis, particulate emissions from engines have important effects on society. Even though the impact of individual engines' emissions may be small compared to other sources such as cigarette smoke, there are almost 1 billion vehicles on the road today emitting particles, and their number is increasing. Therefore, vehicular particulate emissions are a big issue.

2.6.1 PM formation

Soot is formed via a series of steps [18] that are illustrated in Figure 7. The process begins with the formation of precursors when there is a shortage of oxygen and the temperature is within a certain range. Despite extensive investigation, this process is not fully understood but its outcome is believed to depend strongly on the fuel's composition. Two precursors are believed to be the main contributors to soot formation: acetylene and aromatic hydrocarbons [30] [31]. These precursors form aromatic rings that grow into larger kernels, which in turn form small soot particulates through coagulation. Agglomeration of these primary soot

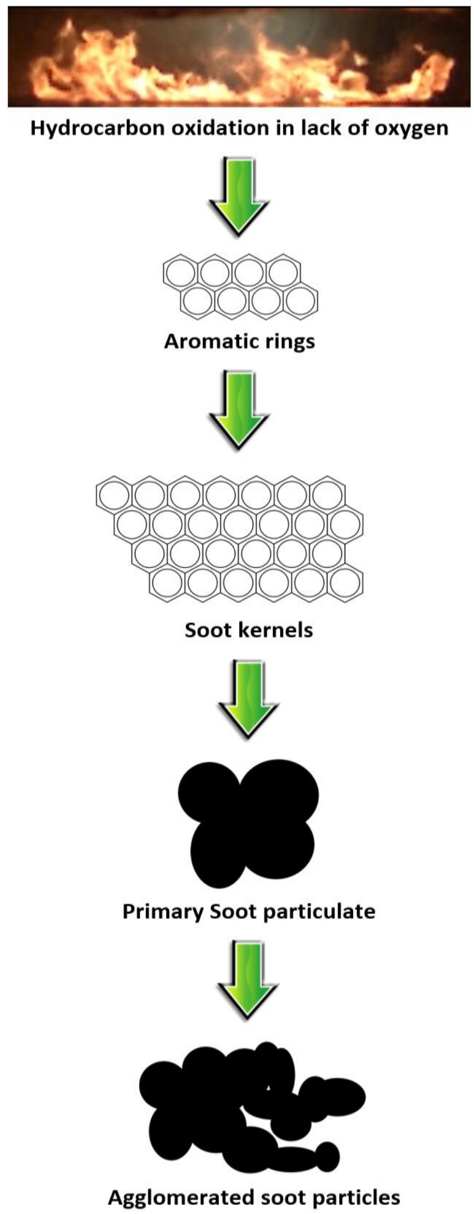


Figure 7 - Illustration of the soot formation process.

particles produces progressively larger particles, at a rate that depends on the total concentration of primary soot particles.

Soot formation occurs in parallel with soot oxidation. The final level of soot emissions thus depends on the balance between formation and oxidation. The oxidation rate increases with the temperature and requires the presence of an oxidant, which is typically O_2 or NO_2 in a GDI engine.

Under ideal conditions, fully evaporated fuel would mix perfectly with an oxidizer in a combustible ratio. This mixture could be homogenous rich, homogenous stoichiometric or homogenous lean. In the rich case, soot oxidation is hampered due to the overall lack of oxygen. In the stoichiometric and lean cases, there is enough oxidant present to oxidize all the soot. Perfect mixing is rare in real engines, so the mixture is generally at least somewhat inhomogeneous, consisting of small lean or rich regions. Soot oxidation is disfavored in rich zones, which therefore serve as sources of soot. The degree of inhomogeneity depends on how well the fuel mixes with the oxidizer, which in turn depends on several factors.

2.6.2 Particulate measurement

Much of the work presented in this thesis related to understanding the measurement of particulates. In addition to being formed and oxidized during combustion, particles are transformed in various ways in the cylinder and as they pass through the exhaust system. Therefore, the choice of sampling point strongly affects the results of particle sampling [13]. The rate of transformation increases as one gets closer to the site of combustion. The main factors governing particulate formation and oxidation are the temperature, humidity, pressure and time of residence. All these factors must be controlled or monitored to ensure reliable and reproducible sampling. When measuring particulates it is necessary to select a physical location for sampling. If the aim is to study particles entering a particulate filter, it would be sensible to collect samples at a point slightly further downstream of the exhaust system because that is where particle filters are typically placed. Conversely, if the aim is to study tail-pipe emissions, sampling should be performed downstream of all the components of the exhaust aftertreatment system. This work focuses on engine-out emissions, so the particle sampling point was located roughly 15 cm downstream of the exhaust valve. Excluding measurements directly inside the combustion chamber, this represents one of the most difficult places to perform sampling because of the environmental challenges presented by its large pressure fluctuations, high temperatures, and rapid changes in particulate formation. In ideal cases, the states of all particulates would be "frozen" and sampled by a measurement device as shown in Figure 8. However if the system is incorrectly conditioned, the particles will continue to transform while moving through the sampling pipes towards the measurement device. Much of the early work in these studies focused on finding a suitable method for measuring particle numbers and sizes. All the measurement techniques utilized in this project were intrusive because they rely on a metallic sampling probe in the exhaust flow. Intrusive sampling will always affect the measured results to some extent, but this approach was deemed to be the best option for the planned studies. A possible alternative would have been to utilize non-intrusive techniques based on laser techniques such as Laser Induced Incandescence (LII). LII uses a laser sheet or a laser with a random geometric form to heat soot particles to around 3500 °C, causing them to emit light. Their light emissions are proportional to their masses, so the particle mass and number can be calculated by measuring the emitted light. This technique has been successfully used in combustion research [57] but was deemed too complicated for this project.

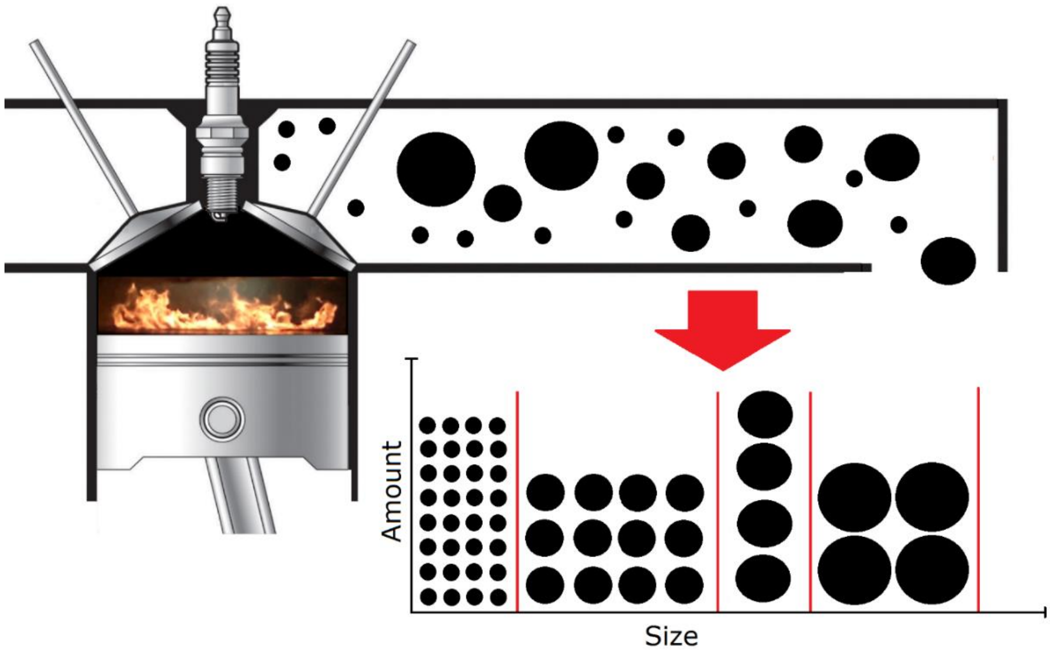


Figure 8 - Schematic illustration of the particle measurements used to generate a particle size distribution diagram.

When this thesis was written, no standard for measuring both particle size and number had been proposed. A decision was made to develop sampling systems for only solid, or non-volatile particles, in keeping with the procedures for certifying engines. Later in the project also a sampling system that did not account for the volatile species was used. Sampling of solid particulates requires the removal of the volatile fraction from the samples while minimizing losses. Several publications discuss sampling and conditioning of particulates, such as those of Giechaskiel et al. [24] [25]. When developing the system for particle sampling, inspiration were drawn from the procedures for measuring particle number and mass defined in the UNECE-R83 PMP (Particulate Measuring Program) standard [1]. The PMP method for measuring particle numbers was chosen as the basis for the method used to determine particle size distributions in this project because it was designed to cause minimal changes in the samples' particle numbers. In general, the method requires the sample to be taken from a "constant volume sample tunnel" as shown in Figure 9, forwarded to a cyclone that filters all particles larger than 2.5 μm, and then passed through a "volatile particulate removal" (VPR) system. The VPR system removes most volatiles by sample dilution. Dilution also prevents further agglomeration by reducing the particle concentration. However, it also makes it more difficult to achieve a high resolution; if the sample is excessively diluted, the level of certain particles may fall below the measurement instrument's detection limit.

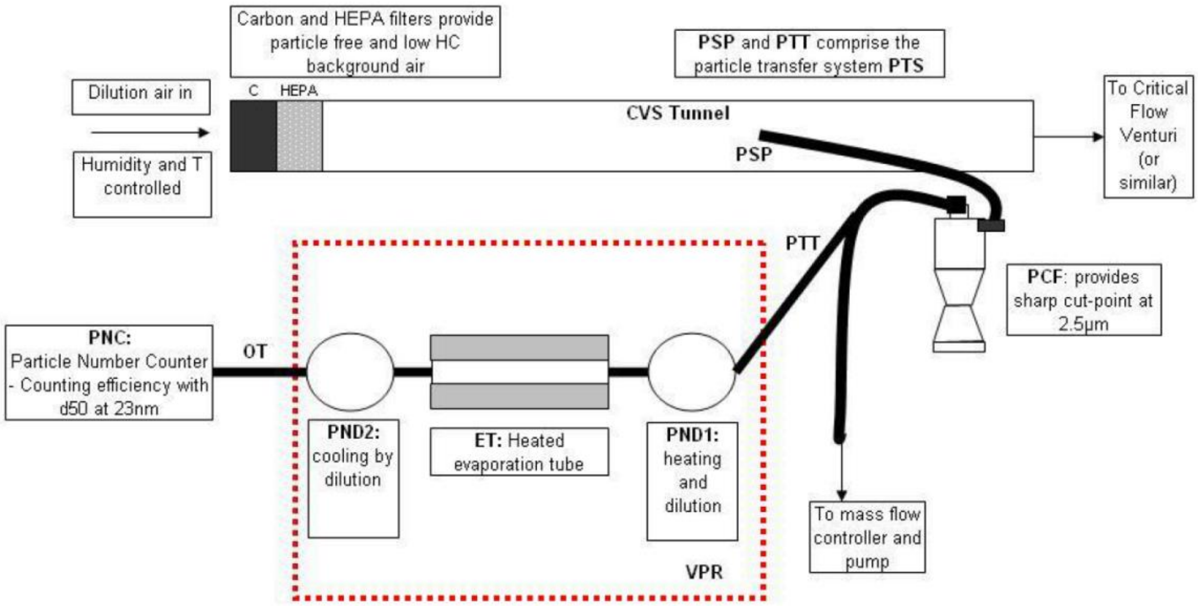


Figure 9 – Depiction of the system for measuring particle numbers detailed in the UNECE standard R83, Rev. April 2011. Image courtesy of UNECE, Particulate Measurement Program.

The VPR consists of a *primary diluter, evaporation tube,* and a *secondary diluter*. In the *primary diluter*, the sample is diluted with dry, heated, filtered air at a temperature between 150 and 400 °C. This reduces the partial pressure and the particle concentration so as to prevent the condensation of volatiles and agglomeration, respectively. Correct operation of the classifying unit requires that the inlet temperature be kept below a certain limit, so the primary diluter is also used to cool the exhaust sample. This could be done by diluting with cooled air but that would promote nucleation, so a secondary diluter is needed. The *evaporation tube* is a simple device that consists of a tube heated to between 300 and 400 °C. When the sample flows through the evaporation tube, volatiles that may have condensed on the particles are re-evaporated and prevented from condensing by the high temperature. The *secondary diluter* is used to further reduce the sample's partial pressure to prevent condensation of the remaining volatiles. Its dilution air is not heated to ensure that the sample outlet temperature remains below 35 °C. Because the first dilution step reduces the sample's partial pressure, the risk of nucleation in this step is low, so dilution can be performed with cooler air than in the first dilution step.

While this project was in progress, there were suggestions that future emissions standards will impose restriction on particulates as small as 5 nm whereas current regulations only apply to particles of 23 nm or above. The need to measure such small particles would increase the complexity of sampling and measurement because most small particles are nucleation mode particulates that form relatively quickly unless suppressed. Our experiences during this project suggest that there is a need for new methods of measuring solid particles of less than 23nm. This problem was also identified and discussed by Zhongqing et al. [19].

2.6.3 Interpretation of particulate size distribution graphs

Particle size distribution graphs are the most common way of presenting a combustion engine's particulate mass and number emissions. These graphs usually have a logarithmic x-axis and a linear y-axis. The x-axis represents the particles' size and usually ranges from 5 to 1000 nm, but ranges up to 2500 nm are also common. The lower limit of 5 nm is due to restrictions in the detection limit of the measurement device. However, the nucleation mode does contain particulates smaller than 5 nm. The y-axis represents the number of particulates in a unit, normalized to the range of the size bin of the classifying instrument. Measurements are presented in this way to facilitate comparisons between results from different types of classifying instruments. Particulate emissions from combustion engines are tri-modal: the first peak represents the nucleation mode, the middle peak represents the agglomerate mode, and the third peak represents the coarse mode. An exact definition of the size ranges for each mode has not been presented, but the usual size ranges for nucleation mode, agglomerated, and coarse particles emitted by gasoline engines are around 0-30 nm, 30-300 nm, and > 300 nm, respectively. Coarse particulates are not formed during combustion; instead, they derive from carbon deposits that form on a surface of the combustion chamber over several cycles and are occasionally released. Figure 10 shows an illustrative particulate size diagram with the typical nucleation and agglomeration mode peaks. The nucleation peak is not always larger than the agglomeration mode peak, and their relative heights can be used to characterize the combustion process. A high nucleation mode peak indicates relatively efficient soot oxidation, whereas a high agglomerate peak indicates poor soot oxidation that may be accompanied by abundant rich burn areas. The coarse mode peak is not visible in this diagram; few measurements acquired in this project exhibited a coarse mode. When a coarse mode peak was observed, it was found that the engine had malfunctioned and become contaminated with oil during the corresponding experimental runs.

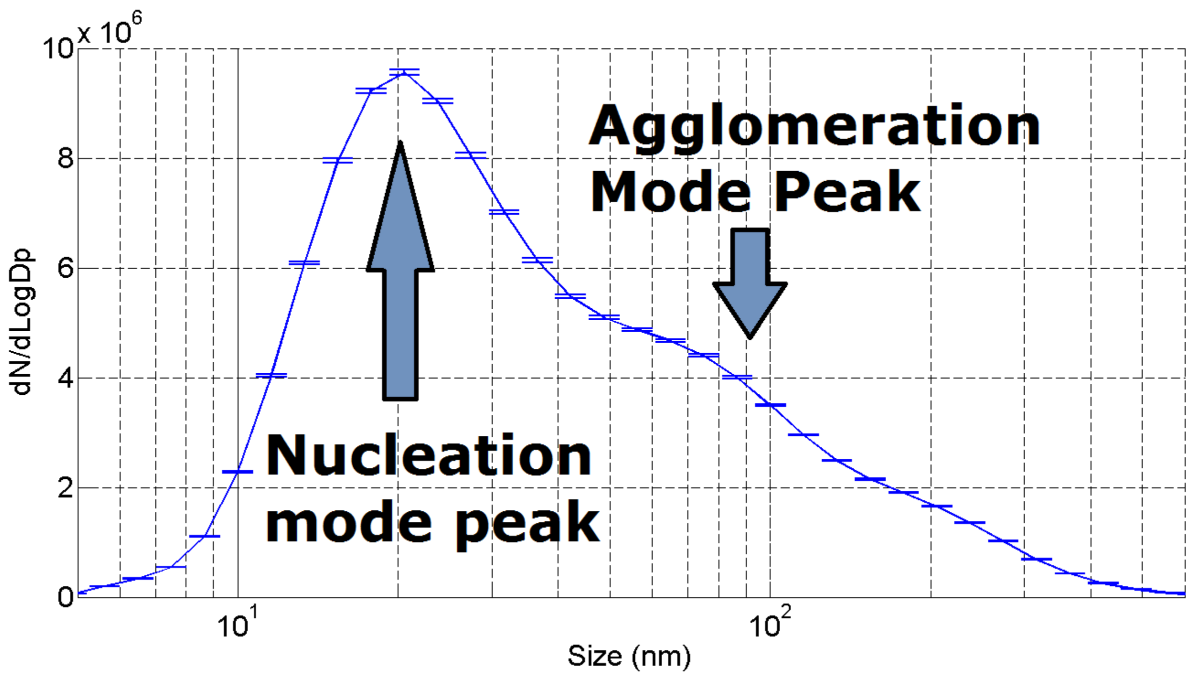


Figure 10 - Typical Particulate Size Diagram (PSD) with nucleation and agglomeration mode peaks.

2.6.4 Soot in stratified combustion

Figure 11 - Image of typical flame from stratified combustion as seen through piston.

As previously discussed, soot is formed during combustion under fuel-rich conditions. Rich gasoline combustion is easily distinguished by its yellow flames. Conversely, well pre-mixed homogenous gasoline combustion generates blue flames. Stratified combustion in the engines utilized in this study generated a mixture of pre-mixed blue flame and fuel-rich yellow flames. Figure 11 shows an image of a flame, seen through the piston, from a high speed video recording of an optical single cylinder engine. The characteristic yellow flame of a rich mixture is clearly in the center and upper part of the image, with a blue flame corresponding to well-mixed combustion on its periphery. This blue flame is formed by the burning of fuel that was injected early and had sufficient time to vaporize and mix with air. No flame is present in the lower part of the image where the intake valves are located. This corresponds to a volume of unused air, which is present because the engine's gas exchange system was designed to generate a strong tumble motion towards the exhaust port.

In the beginning of this project, it was suggested that the yellow flames may result from the burning of partly vaporized fuel and fuel in liquid flame, which would generate a diffusion flame. The rate of burning in a diffusion flame is controlled by the rate of mixing of the reactants, the fuel and the air. Consequently, diffusion flames tend to burn more slowly than pre-mixed flames, and create more soot due to the risk of lacking oxygen. There is no evidence of such diffusion flames being present in the free air volume inside the combustion chamber. In fact, experiences from previous experiments performed by the research group have suggested that most of the fuel is vaporized before the start of combustion. It is possible that a diffusion flame may be present on some surface of the combustion chamber that is subject to fuel wetting, such as the piston top.

Figure 12 - Image of flame as seen through cylinder liner.

Figure 12 depicts combustion in an optical single cylinder engine as seen through the cylinder liner, with a flame that follows the piston top. This is due to fuel impingement on the piston top during the injection phase. Several studies have identified such pool fires as the largest contributor to soot formation in stratified operation [50] [51] [52]. Velji et al. [50] studied the influence of injection timing on soot emissions in stratified operation and found that the injection timing did not significantly affect engine-out particulate emissions. They also found that earlier injection increased the penetration length because of the lower back pressure in the cylinder at earlier injection timings. Delaying the injection reduced the penetration length but increased the penetration angle due to the greater back pressure. However, the earlier injections caused greater fuel impingement overall. Experiments conducted within this project at various back pressures support these conclusions (see section 6.2 Influence of forced induction for details).

It is important to note that soot is not only formed in pool fires. Due to the nature of stratified combustion, the mixture of air and fuel is inhomogeneous, with a rich kernel and a lean perimeter zone. Soot forms in the rich zone, but because the overall air-fuel ratio is lean, there is sufficient oxygen for complete soot oxidation. Consequently, the presence of locally rich mixtures is not a problem if there is sufficient time for oxidation, and the availability of sufficient time is a prerequisite for adequate soot oxidation. Soot oxidation becomes more efficient at higher temperatures; during the expansion phase, both the temperature and the rate of soot oxidation fall. Therefore, soot formed early in the combustion process is more likely to be oxidized than that formed at later stages. Late-formed soot accounts for most of an engine's particulate emissions. In addition to soot formed from combustion of fuel, soot can also be formed in the combustion of lubrication oil. A small amount of lubrication oil is always present in the combustion chamber.

However, in comparison to the soot emissions formed from combustion of fuel, the soot emissions formed from oil are negligible in a healthy engine.

2.7 Standard emissions

This chapter describes the group of emissions that are frequently considered together and referred to as standard emissions. While there is no established definition for this term, it is commonly understood to mean those pollutants that were subject to legal limitations before legislation concerning particulate emissions was introduced.

2.7.1 Nitric Oxides

Nitric oxide (NO) is an indirect greenhouse gas because it forms ozone upon decomposition. Combustion engines are a major global source of atmospheric NO; it is formed when nitrogen and oxygen react at high temperatures such as those that occur during combustion [90]. Nitrogen dioxide is formed by the reaction of NO with oxygen. The two gases are normally referred to collectively as NOx. Because stratified combustion uses a heterogeneous air-fuel mixture with lean regions and long burn durations in some cases, it creates favorable conditions for NOx formation. As shown by Akihama al. [49], NOx emission formation peak in regions where the lambda value is between 0.9 and 1.25 for gasoline. If the equivalence ratio of the fuel mixture could be kept outside this range, NOx emissions could be reduced. However, the inhomogeneity of the fuel mixture and the short time delays in stratified combustion make it hard to control the equivalence ratio in this way. NOx emissions from stratified combustion are normally controlled by introducing recirculated exhaust gases into the combustion chamber [44] [45] [46], which has the drawback of increasing particulate emissions. This is discussed in more detail in section 6.1 Influence of EGR.

2.7.2 Hydrocarbons

Engine out hydrocarbon emissions consist of fuel or lubrication oil that has not undergone combustion. Crevices are important sources of hydrocarbon emissions in homogeneous combustion, but over-dilution is a more important source in stratified combustion [56] [67].

Over-dilution can occur at the outer periphery of the fuel-air mixture because of its heterogeneity, which hinders complete combustion. The risk of over dilution can be reduced by igniting earlier, by utilizing more suitable ignition systems (as discussed in this thesis), or by optimizing the fuel spray and its interaction with air movements in the cylinder.

2.7.3 CO_2 and CO

Carbon dioxide is the primary greenhouse gas emitted by humans, largely because it is formed by the combustion of fossil fuels. Carbon monoxide is also a greenhouse gas, albeit a weak and indirect one. Both are formed by the reaction of hydrocarbons with oxygen, and the level of carbon dioxide production is a direct measure of an engine's fuel consumption. Stratified combustion generally produces relatively low levels of CO_2 and CO because it is more fuel-efficient than homogenous operation.

2.8 Important parameters in stratified combustion

Successful combustion in stratified mode depends on several interdependent factors [29]. Unlike in homogenous operation, most of the fuel-air mixing in stratified mode occurs shortly before the ignition sequence. The result of the fuel mixing and ignition process determines the outcome of the combustion. In general, stratified mode combustion occurs via the sequence of events outlined in the list below. The process is affected by several factors such as the geometry of the injector [29], the type and properties of the fuel, the injection pressure, air motion, air properties, and the engine timing [14] [15] [16]. This subchapter briefly describes some of the relevant processes.

Stratified combustion involves the following steps:

1. Air is inhaled in the intake stroke and flows into the chamber in a pre-defined way based on the design of the intake channels and the interaction between the piston and the combustion chamber.
2. Fuel is injected through the injector late in the compression stroke.

3. Part of the injected fuel atomizes and evaporates shortly after injection due to the high injection pressure
4. In parallel with points 2-3, the air motion induced in step 1 affects the evaporation process and the physical position of the fuel cloud.
5. The spark is initiated. If the mixing process has successfully generated an ignitable mixture around the spark in the time window when the spark is present, combustion proceeds.

Even if combustion is successfully initiated, the continued success of the combustion process will depend on whether the injection and mixing processes have formed a combustible mixture at the flame front. If the fuel has been over- or under-diluted with air, combustion will be incomplete and the engine will produce elevated hydrocarbon or particulate emissions.

2.8.1 Air motion

Air motion in a combustion engine serves two main purposes: to speed up the flame speed and to improve the mixing of the fuel and air. Most gasoline engines utilize a tumbling motion where the air flows around an axis parallel to the crank shaft. Light duty diesel engines normally utilize a swirling motion in which the air flows around an axis parallel to the cylinder liner; the air motion is generated during the intake stroke and then transformed during the compression stroke. Some designs disrupt the air flow to create undefined turbulence while others aim to retain some of the original flow during the expansion stroke. The initial air motion is primarily established by the design of the intake channels and valves, but also by the interaction of the flow with the combustion chamber walls and squish volumes. Interactions with the walls occur when the air flow impinges upon a wall and changes direction. Conversely, squish volumes are designed to “squeeze” the air-fuel mixture between two surfaces during the compression stroke. This causes movement from the “squeezing” region towards areas with more volume, and is normally used to direct an air-fuel mixture towards specific areas of the combustion chamber or to increase turbulence in specific regions. The advantages of a squish area come at the cost of an increased overall surface area and the risk of creating hot zones. Higher surface areas result in greater heat losses, and hot zones can become knock initiation zones. It should be noted that high levels of air movement are not purely beneficial: designing intake channels and valves in a way that increases air motion generally incurs a cost in terms of pumping losses. Increased air motion also increases the loss of heat to the combustion chamber and cylinder walls. Finally, excessive tumble levels during ignition can potentially deflect and extinguish the spark. All these factors must carefully be considered when designing of a combustion system.

While the mean air flow will always be the same in a given engine, the flow of air in the cylinder varies somewhat in both intensity and direction from cycle to cycle for several reasons including changing boundary values and temperature variations. These variations are normally referred to as cycle variations and in the worst cases they can cause misfires or partial burns. Air-fuel mixing in stratified operation occur shortly before ignition and is therefore more sensitive to variations in air flow than is the case in homogeneous operation.

2.8.2 Initial flame kernel development

The development of the initial flame kernel has a profound impact on the overall success of the stratified combustion. Zeng et al. [60] studied initial flame kernel growth and correlated it with the occurrence of good, partial, and slow burn cycles. They found that partial or slow burn cycles were characterized by a slow early flame kernel development. They also found that once the flame kernel reached a radius of ~5 mm, the flame expanded rapidly at the same rate as in good burn cycles even if the initial flame development was poor.

Another factor that affects the initial flame kernel development is the possibility of spark restrikes, which can occur with conventional coil-based ignition systems. A restrike occurs once the spark length exceeds a certain threshold. When this occurs, the arc deflection process is interrupted and a new voltage break down takes place, causing the formation of a new spark between the electrodes with a much lower spark length. A requisite for a new voltage break down process is that there is sufficient energy left in the coil. If the initial spark had created an embryo flame kernel before being lost, the restrike will cause the flame kernel to become detached, reducing its supply of energy [61].

Peterson et al. [62] examined the correlation between flame kernel development and fuel cloud properties by using high-speed Planar Laser Induced Fluorescence (PLIF), Particle Image Velocimetry (PIV), and Mie scattering in an optical engine to analyze the spatial and temporal evolution of the fuel cloud and the flow velocity of the flame kernel. They used nitrogen dilution to destabilize the combustion process and force misfires, and found that all of their cycles (regardless of the conditions) yielded a fuel cloud within the flammable limit in the vicinity of the spark plug. Measurements of the spark energy also revealed that all the cycles had sufficient energy for successful ignition. It was therefore be concluded that the misfires were related to the interaction between the ignition process and the fuel cloud. All cycles that exhibited misfires were characterized by an initial flame kernel growth that failed to develop into full combustion. These cycles had small flame kernels in rich mixtures that never reached the bulk of the flammable mixture, which is what caused the misfires. Similar observations were made during the studies reported in this thesis. Figure 13 and Figure 14 show piston and side views, respectively, of two cycles studied in this project, one of which misfired and one of which did not. The images were taken from high speed video footage captured in an optical engine operated at 2000 rpm and 4 bar BMEP. In both cases, the sequence of images starts at the moment of ignition and ends at 45 CAD after the ignition event, with steps of 15 CAD. The horizontal line in the images is a measurement artefact caused by a faulty camera sensor and can be disregarded. Although the images represent only one cycle rather than an average of all misfire and misfire-free cycles, they clearly illustrate the phenomenon described by Peterson et al. The ignition sequence look similar in both cases. However, clear differences are apparent at 15 CAD: the misfire-free cycle has started to develop a flame that covers almost a third of the projected area by this point, whereas no visible flame is present in the misfire cycle. At 30

CAD, the misfire-free cycle has ignited the bulk of the fuel cloud, giving rise to intense yellow luminosity as the flame burns in fuel-rich zones. Conversely, only small yellow flame spots are visible in the misfire cycle; the bulk of the fuel is not ignited. At 45 CAD, the misfire-free cycle still exhibits active combustion whereas the misfire cycle shows only a small amount of burning, with unburned gases occupying most of the chamber. Fansler et al. [63] reported similar findings in studies on early flame kernels in locally rich regions. However, they also found that the flame kernel moved upward rather than downward relative to the bulk of the fuel, which was located in a bowl in the piston in the studied combustion system. This motion was attributed to convection and the local burning velocity. The air movements described in the previous chapter probably affect the flame kernel's movement, and conclusions from one combustion system cannot be transferred directly to one with different airflow patterns. It can however be concluded that a weak interaction between the initial flame kernel and the fuel cloud is a major source of misfires.

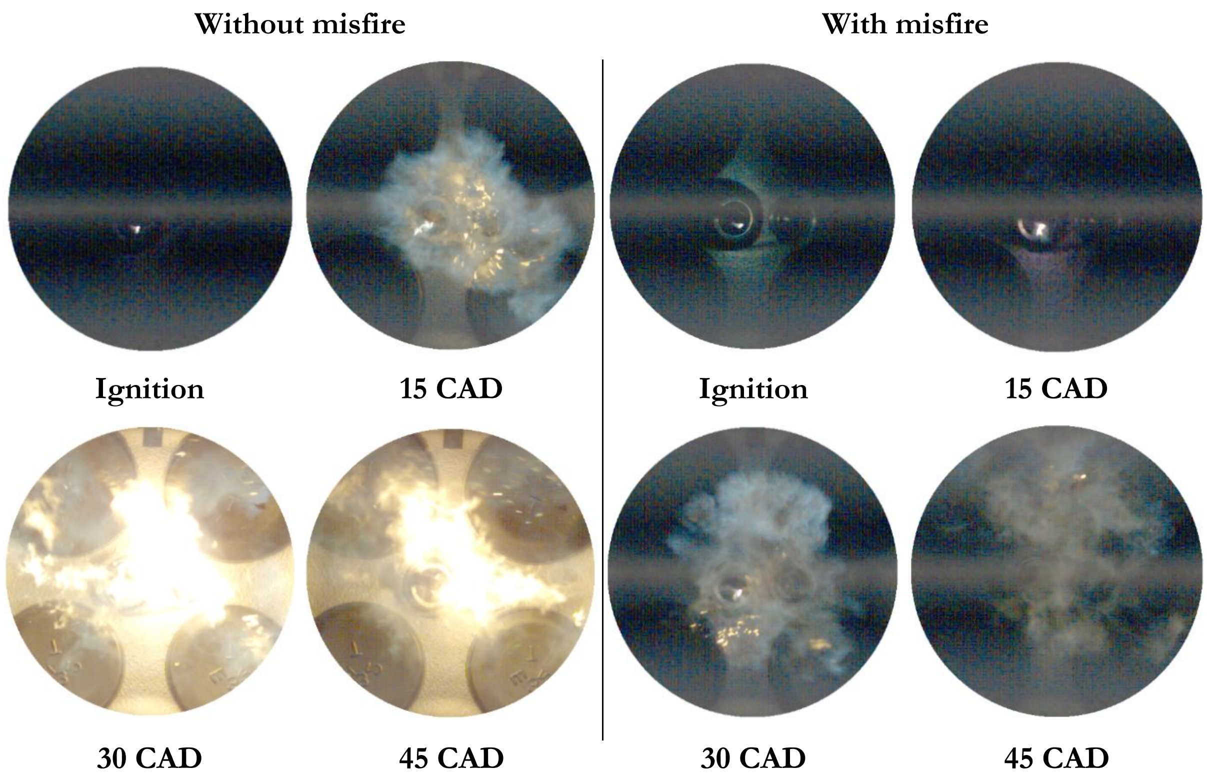


Figure 13 - Image sequence showing characteristic flame behavior (viewed through the piston) during a misfiring cycle (right) and a misfire-free cycle (left) at 2000 rpm and 4 bar BMEP.

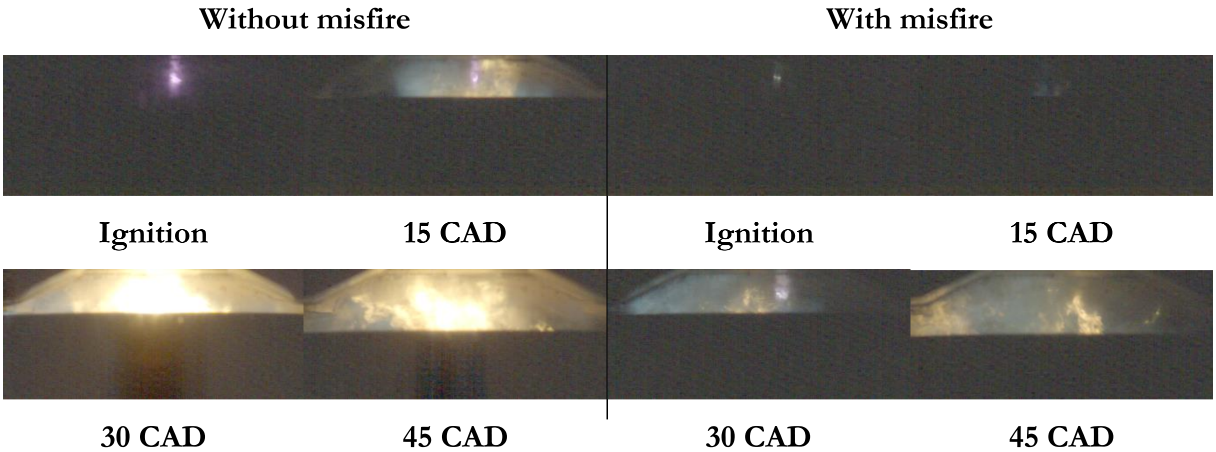


Figure 14 - Image sequence showing characteristic flame behavior (viewed from the side) during a misfiring cycle (right) and a misfire-free cycle (left) at 2000 rpm and 4 bar BMEP.

2.8.3 Fuel injector type and characteristics

Most DI engines use either solenoid or piezo-actuated injectors. The former injector type uses an inward-opening needle while the latter uses an outward-opening needle. Solenoid-actuated injectors are substantially cheaper than piezo actuated systems but they also open and close more slowly and are therefore unsuitable for combustion regimes requiring multiple injections in quick succession. Piezo injectors can create more homogenous fuel clouds because they produce conical sprays as shown in Figure 15, in which the left-hand image shows a fuel cloud from a piezo-injector and right-hand images show one from a multi-hole solenoid injector. The upper and lower images show sprays with fuel temperatures of 293 K and 363 K, respectively. The rays and plumes from both injectors are less distinct at the higher temperature, demonstrating that higher fuel temperatures facilitate evaporation. The lower images represents flash boiling conditions which occurs when the fuel exits the nozzle in a supercritical state. Characteristic vortices are clearly visible in the image of the fuel cloud from the piezo-injector at 363 K. These vortices are formed by drag forces [64] and occur towards the bottom of the fuel spray, where they facilitate air entrainment. This region of the spray is well-suited for achieving stable ignition. Because of their rapid opening and closing times, piezo injectors are favored in systems requiring multiple injections. Multiple injection can reduce the spray's penetration length and hence the risk of piston-wetting. It is also possible to divide the combustion into several phases, for example one where the flame kernel is initiated by a pilot injection and a second where a fuel-rich mixture is delivered towards and into the flame kernel. Modern piezo injectors are four to five times faster than conventional solenoid injectors and can inject fuel up to ten times during a combustion cycle. The fuel injection event creates turbulence, which can be utilized to reduce the overall sensitivity of the ignition event to exogenous airflow fluctuations.

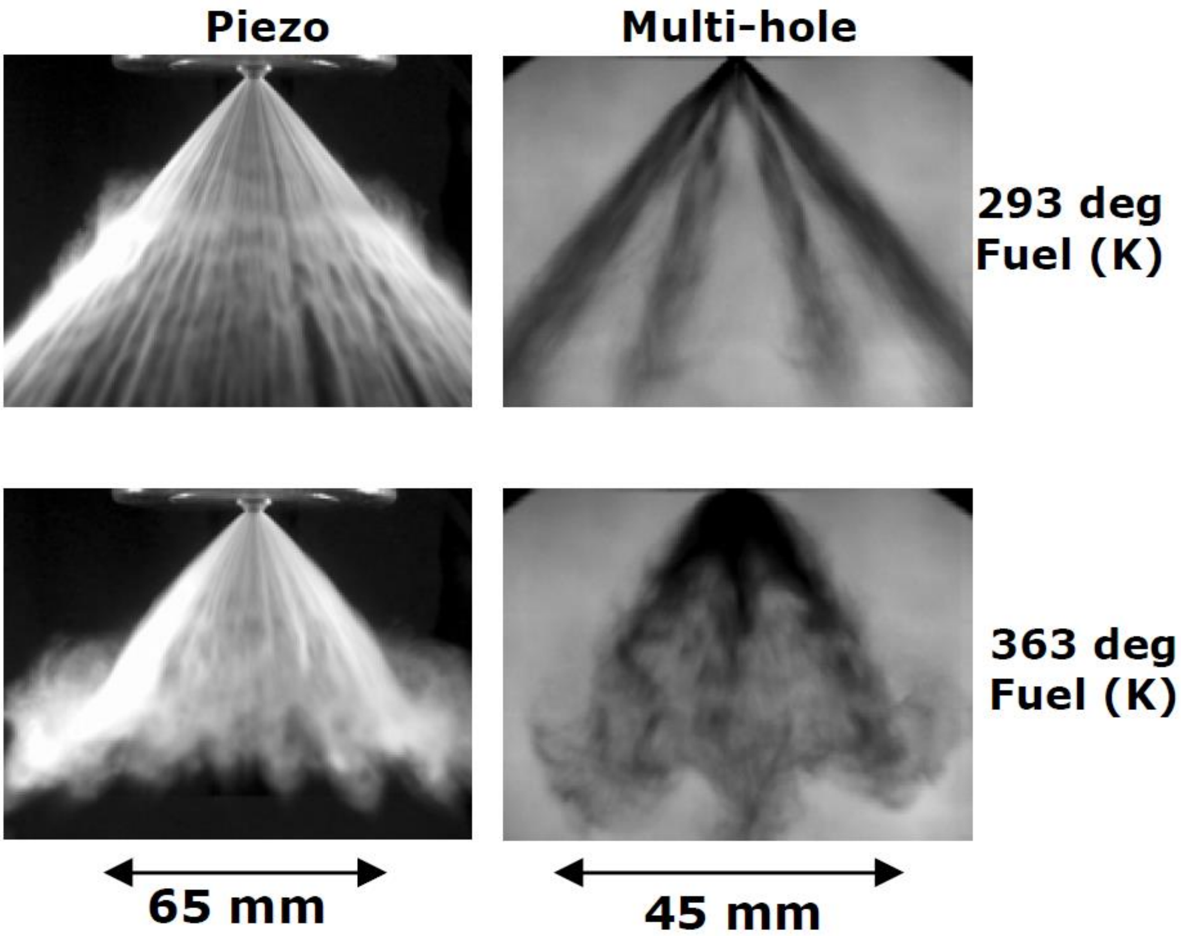


Figure 15 - High-speed images of single injections from a multi-hole injector (right) and a piezo injector (left) at two different fuel temperatures. Image courtesy of [64].

2.8.4 Ignition system and spark plugs

The spark exhibits cycle to cycle variation, primarily because of variations in the airflow that can change its orientation and position. During the firing event, the spark dances around rather than taking the shortest path between the electrodes. Figure 16 shows a spark filmed through the piston during a combustion event. As can be seen in the image, the spark has been deflected sideways. Moreover, the spark is in continuous motion and normally follows the movement of the air.

Figure 16 - Spark in combustion engine filmed from piston-view.

Conventional coil-driven spark plugs have similar designs with a center electrode and one ground electrode. The spark is fired between these two electrodes. Spark plugs are designed to operate optimally within a specific temperature range. Because of this, they are also designed to permit a certain amount of heat loss through their metal parts. This leads to heat losses from the flame kernel at the moment of its initiation. If these losses are too large, flame kernel development is hampered. Fansler et al. [63] studied flame kernel development with a conventional single electrode plug and a three-electrode plug, and found that the three-electrode plug was more susceptible to early flame kernel development. However, these experiments were performed in a system designed around the single-electrode plug and no attempt was made to modify or optimize its design for use with the three-electrode plug. In a three-electrode plug, the spark can be formed between the center electrode and any of the three ground electrodes. Consequently, the volume potentially exposed to the spark is greater than for a single electrode plug, and the spark may potentially encounter a wider range of air-fuel ratios and local air flows. It is also worth noting that the ground electrodes can physically hinder flame propagation after initiation. In contrast to the results of Fansler et al., experiences from studies within this project has shown that a three electrode spark plug could improve the combustion. This contradictory result is probably due to differences in fuel distribution between our test engine and theirs that result from their different designs. It is worth stressing again that every combustion system has distinct properties, and phenomena that are important in one system may not even occur in another.

One way to increase the likelihood of successful ignition is to improve the ignition process. The simplest way to do this is to increase the power of the spark, but more refined systems are available. The environment around the spark plug is harsh, with high air velocities, liquid fuel, and sometimes also wet electrodes. All these factors hamper spark development, and may extinguish it. Single coil systems are usually incapable of reigniting an extinguished spark because of the low residual energy levels in the coil. One way to overcome this problem is to use multi-spark systems. For example, Daimler [3] use a multi-spark system in their DI gasoline engines that can operate in stratified mode. The definition of multi-spark ignition systems encompasses both systems that fire several sparks in succession and those with extended spark durations. Systems of both kinds are intended to increase the time during which a spark is present to increase the chance of successful ignition. As discussed in sections 2.8.1 Air motion and 2.8.6 Mixture formation, the fuel cloud's composition and distribution change constantly as it shifts and moves. Consequently, there is no guarantee that the spark will be close to an ignitable mixture at any given point in time. Increasing the spark's duration increases the likelihood that it will come into contact with an ignitable fuel mixture at some point. Delphi [70] has published a study comparing single spark ignition systems to more advanced multi-spark ignition systems which suggests that multi-spark systems increase robustness during stratified operation. Another useful technology is so-called Corona discharge systems. These systems differ substantially from conventional spark plugs: whereas a spark plug generates a spark between a center and a ground electrode, a Corona ignition system fires into free air. In a normal spark ignition system, the spark is the result of an electric breakdown, which requires a lot of energy to create. The Corona ignition system instead ionizes the surrounding air-fuel mixture with an alternating current, which creates an electric discharge. This allows the system to deliver more energy to the igniting mixture and to maintain a spark over a long period. Perhaps more importantly, it permits the formation of multiple ignition points because the spark does not jump between electrodes as in a conventional spark plug. This means that successful ignition can be achieved even if the fuel cloud is spread across a relatively large volume, as is normally the case during stratified operation. Detailed studies on corona ignition systems are presented in Paper 5. Similarly, Pineda et al. [53] studied Corona Discharge Ignition systems in a boosted DI gasoline engine. While their engine was operated in homogenous mode, their results are consistent with the results for stratified combustion presented in this work. They observed that compared to a conventional sparkplug, the corona system reduced cyclic variability, yielded more complete combustion, and extended the knock-limit. The last result is unimportant for stratified combustion because the combustion phasing is rarely limited by knocking. However, the increased knock-limit is probably due to a higher burn rate resulting from multiple flame front initiation. In stratified combustion, this would probably reduce the risk of excessive dilution in the peripheral parts of the fuel cloud. An improved ignition such as the one offered by a Corona ignition system can in turn be used to increase EGR tolerance as well as reducing fuel consumption due to the increased combustion efficiency following the increased burn rate.

2.8.5 Pre-cycle effects

Pre-cycle effects are phenomena in which the outcome of one cycle affects the combustion process in a subsequent cycle. They have been studied by Peterson et al. [62], who investigated the coupling between successive cycles at various dilution ratios and spark timings. They found that in general, a partial burn or misfire was preceded and followed by a successful cycle. It was rare for two failed cycles to occur in a row. They also found that the successful cycle after a misfire or partial burn was usually stronger than normal (i.e. produced a higher IMEP). This was attributed to the retention of unburned fuel from the previous cycle in the cylinder as a result of poor scavenging.

2.8.6 Mixture formation

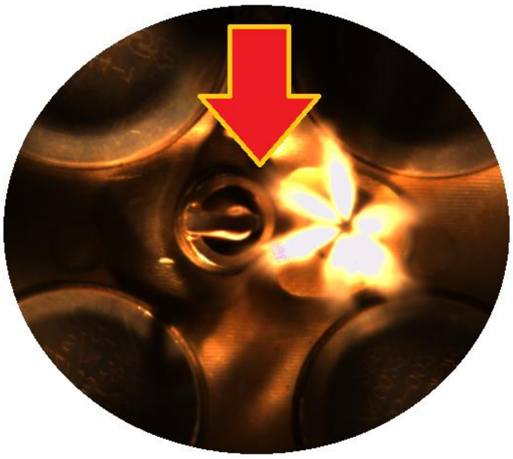


Figure 17 - Image from piston view showing spray injection during compression stroke.

Figure 17 shows an image captured through the piston of the optical engine during an injection event, with the fuel spray plume being clearly visible. The orientation of the spray and its rays are normally discussed in terms of spray targeting. Spray targeting depends on several factors including the injection timing, physical design constraints, fuel pressure and air motion. It is important to ensure that the spray does not impact on any surfaces such as intake valves or spark plugs because that would hinder evaporation and could give rise to diffusion flames. The image in Figure 17 shows that the spray is targeted so that two rays are being fired towards different sides of the spark plug, which is of the conventional single-electrode type. The arrow indicates the direction of the tumble flow in the horizontal plane just below the spark plug. Mixture formation occurs in an interaction zone between the air and the fuel spray, where the fuel atomizes, evaporates and mixes with the air. The process is complicated and depends on several factors that will not be discussed further in this thesis. However, it is important to note that variations in the spray pattern and air motions give rise to variation in the fuel distribution between cycles. To achieve stable combustion, the goal is to consistently form a combustible mixture in close proximity to the spark. This requires careful coordination of the timing of the ignition and injection. It is important to avoid over-dilution of the spray while still allowing it to mix to an extent that avoids excessive soot production. There are three main ways by which this can be done in stratified operation: igniting the head of the spray, igniting in the middle of the spray, or igniting the tail of the spray. The later the ignition timing, the more time there is for fuel mixing, which increases the proportion of pre-mixed combustion and reduces soot formation. The third approach was used in all experiments conducted in this project. Fansler et al. [65] [66] have studied the local fuel concentration around the spark by measuring the concentrations of diatomic carbon (C_2) and cyano radicals (CN). They found that the local liquid fuel concentration around the spark remained quite high across a fairly wide range of ignition timings. They also found that the vapor phase around the spark was generally fuel-rich, with an equivalence ratio of up to 2.8, but that there was considerable cyclic variation in the equivalence ratio. Peterson et al. [62] used LIF to measure the equivalence ratio distribution of the fuel cloud and obtained similar results, but were also able to show that the fuel-air mixture in the outer part of the fuel cloud is lean, which is consistent with the results obtained in this project. Figure 18 presents high speed video images showing the initial flame propagation in an optical engine operating at 1000 rpm and 1.5 bar BMEP. The images show the evolution of the flame from the moment it first becomes visible to the point that the fuel-rich flame appears. The first image (i.e. that in the upper left corner) was captured a few CAD after the end of the fuel injection, when the spark ignited the tail of the fuel ray. A small pre-mixed flame in a stoichiometric or lean mixture is visible to the right of the spark plug. This flame continues to develop up until roughly 8.4 CAD, when a fuel-rich flame appears. Combustion was observed to begin with a stoichiometric or lean flame in all the experiments conducted in this work, which is consistent with the findings of Peterson et al.

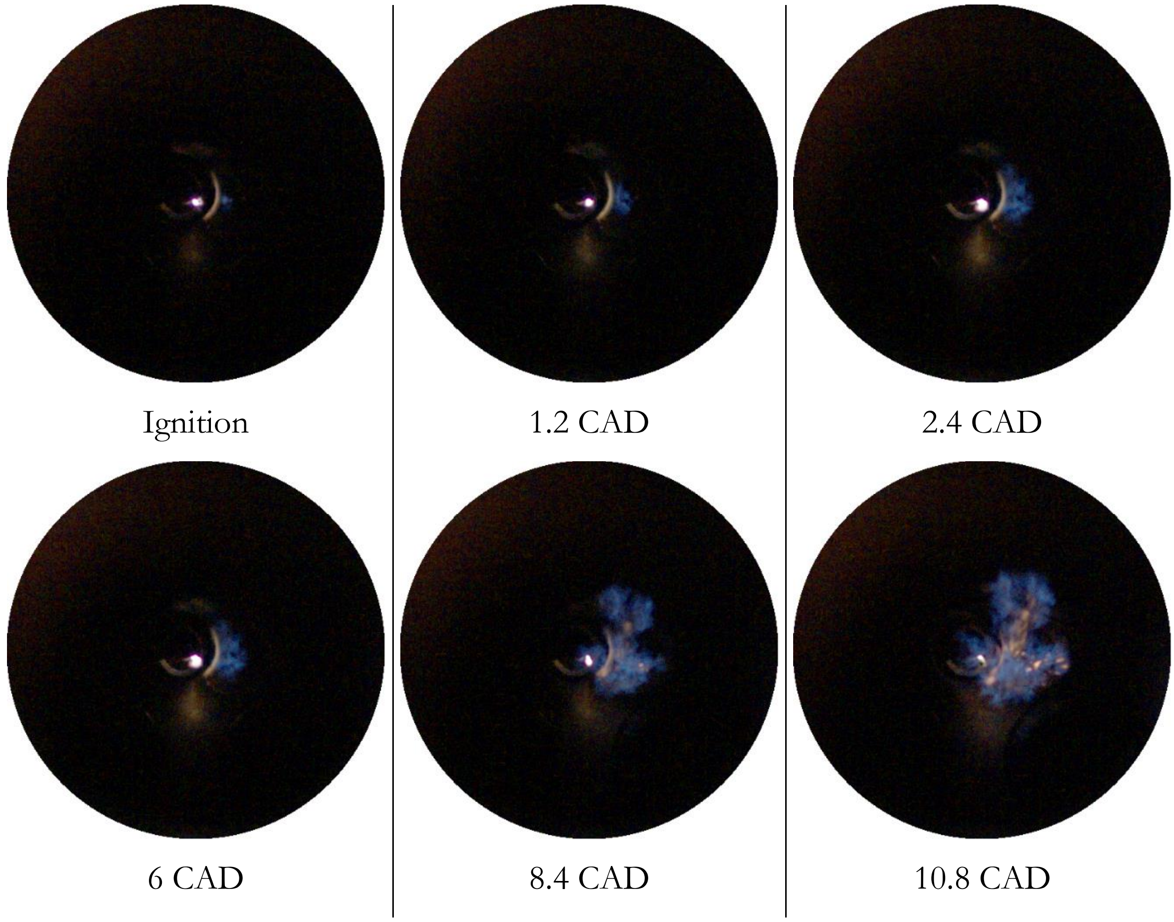


Figure 18 - Image sequence showing the initial flame front development shortly after ignition. The labels indicate the timing of each image in CAD since ignition.

2.8.7 Fuel injection pressure

Higher fuel injection pressures accelerate fuel evaporation. The injection pressure is important in all DI engines but especially in those using stratified combustion due to their short mixing times. As discussed above, it seems that combustion is always initiated by the formation of a lean or stoichiometric flame. A flame within this range in a DI-engine can always be considered to be pre-mixed in comparison to a yellow flame that could also be a diffusion flame (as well as rich pre-mixed). A pre-mixed flame consists of evaporated and well mixed fuel. Their formation is facilitated by high injection pressures, which produce smaller fuel droplets that evaporate quickly. Since the flame originates from this mixture, high injection pressures favor ignition. A high injection pressure is also beneficial in the later stages of combustion because it increases the amount of time in which the fuel can evaporate and mix. Moreover, pre-mixed flames of stoichiometric to lean mixtures produce low levels of soot. The results of an investigation into the effect of varying the fuel injection pressure in stratified combustion are presented in chapter 6.3 Influence of fuel injection pressure).

2.9 Characteristics of stratified and homogenous flames

The difference between stratified and homogenous operation can be hard to understand without visualization. During this project several videos were recorded from both stratified and homogenous combustion. Figure 19 presents several images from such high speed videos that show what combustion looks like in both operating modes. All the images are taken from videos filmed through the piston of the optical engine and correspond to a single cycle in each operating mode.

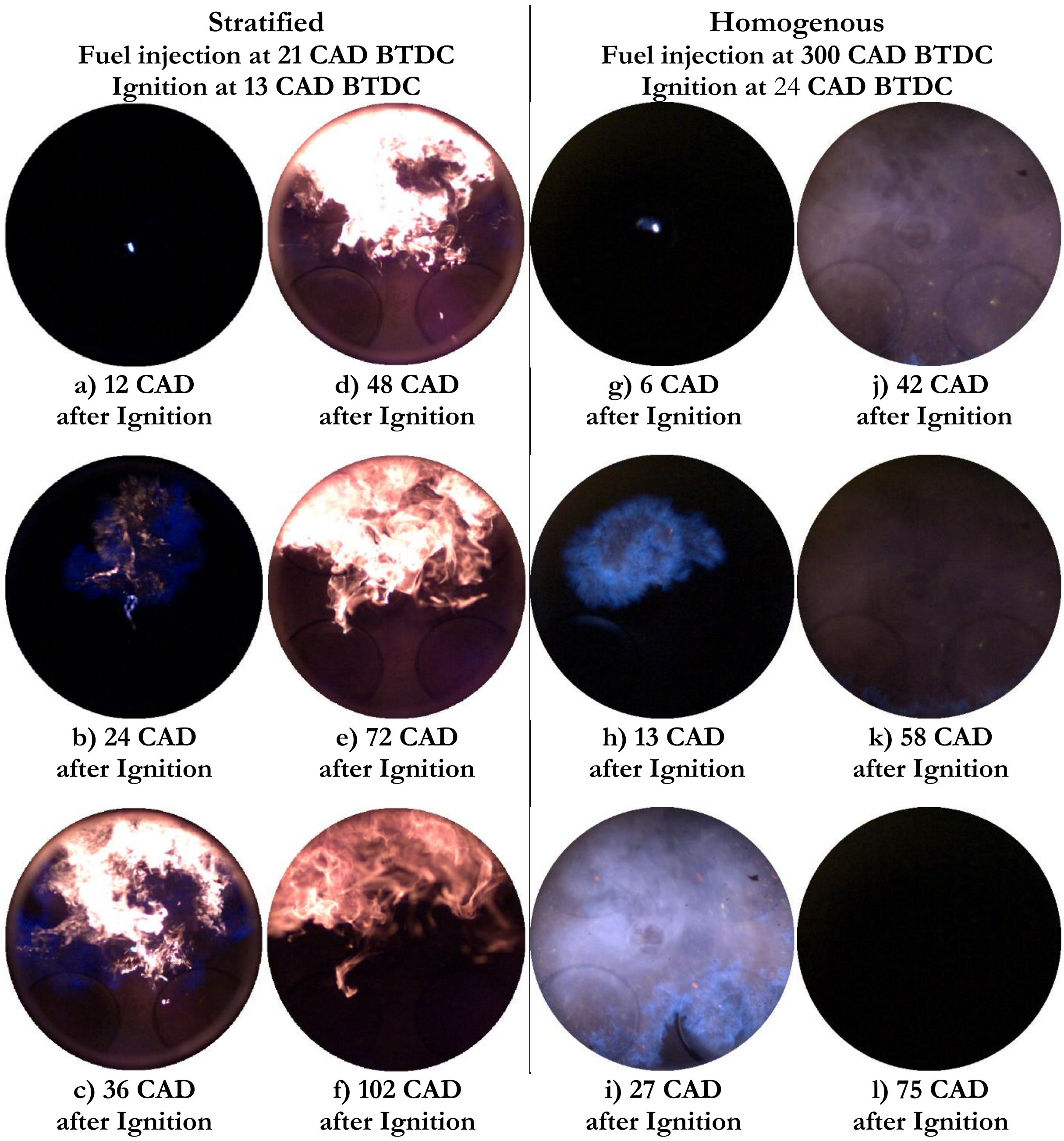


Figure 19 - Visualization of flame development in stratified mode (left) and homogenous mode (right).

In this example, the injection for stratified combustion was initiated at 21 CAD BTDC. Ignition was initiated at 13 CAD BTDC and the spark was maintained up to roughly TDC, as shown in image (a). At this time, the spark was visible but the flame was not. Image (b) was acquired at 24 CAD after start of ignition, when the flame was more developed. A clear blue flame is visible, resulting from pre-mixed combustion of a locally stoichiometric or lean mixture. The fuel burning in this phase is therefore the material that was injected right at the start of the injection, which has had time to mix with air. The next image (c) features a more prominent yellow flame. The flame still has a relatively large blue part but the yellow part has grown. The yellow flame consists of burning fuel that was injected later and is burned under rich conditions. In the following image (d) the blue flame is almost completely gone and the yellow flame occupies most of the combustion chamber. This remains true in the later images (e and f); a yellow flame can be distinguished as late as 72 CAD after start of ignition. The last image (f) is presented to show how slow stratified combustion can be. The ignition phase is slow, the middle part of the combustion is relatively fast, and the end is slow again. This slow combustion towards the end is sometimes referred to as the "tail" of the combustion and is easily identified in heat release curves. Stratified combustion does not necessarily have to suffer from this late slow phase, but the combustion system studied here did. Its occurrence is probably due to pool fires resulting from fuel impingement on the piston top. Figure 19 g-l shows flame development during homogenous combustion under similar operating conditions as for the stratified case. After the moment of ignition (g), the flame develops in both directions under the influence of the air motion in the cylinder. As in the early stages of stratified combustion, the flame is blue, indicating that the combustion is pre-mixed. However, in the homogeneous stoichiometric case the blue flame persists throughout the combustion process (images h-l) and there is no yellow fuel-rich flame. The early stages of flame development proceed more rapidly than in the stratified case: the extent of flame development 13 CAD after ignition in the homogeneous case (image h) is comparable to that seen after 24 CAD in the stratified case (image b). The flame also burns out more quickly in the homogeneous case: it is extinguished by 58 CAD (image k) whereas the stratified flame is still burning at 102 CAD (image f). The combustion system utilized in this study tended to form pool fires resulting from fuel impingement in stratified mode, which was the reason for the persistence of the yellow flame in that case; pool fires did not form in the homogeneous combustion experiments because injection occurred earlier in the cycle so there was more room for the spray to evolve and ignite without impinging on the piston or cylinder walls.

3 Experimental Setups

This chapter presents the experimental hardware used in the experiments discussed in the thesis. The intention is to give the reader an understanding of the hardware and the engine operating points at which it was used.

3.1 Engines

Four engines were used in the studies presented in this thesis, as shown in Table 1. Most of the research was performed on single cylinder research engines with various engine heads. However, one study on particulate sampling systems was performed with a complete engine (Engine 4) operating in homogeneous mode. Two types of single cylinder engines were used, one for optical access and one made entirely out of metal. In one study, two combustion heads with identical configurations were used for both the full metal engine and one of the optical engines (Engines 1 and 2).

Table 1 - Engine configurations utilized in this thesis.

	Engine 1	**Engine 2**	**Engine 3**	**Engine 4**
Type	Single cylinder metal	Single cylinder optical	Single cylinder optical	Four-cylinder metal
Description	AVL 540	AVL 540	AVL 540	Volvo VEP MP
Gas exchange system	Spray-Guided, tumble with variable swirl	Spray-Guided, tumble with variable swirl	Spray-Guided, high tumble	Spray-Guided, tumble
Bore [mm]	83	82	82	82
Stroke [mm]	90	92	92	93,2
Compression ratio	9-10:1	10.6:1	10:1	10.8:1
Boosting system	Roots blower	Compressed air	N/A	Single turbo
Injector	200 bar, multi-hole, solenoid	200 bar, multi-hole, solenoid	350 bar, multi-hole, solenoid	200 bar, multi-hole, solenoid
Piston type	Bowl with valve clearance pockets	Flat	Flat	Bowl with valve clearance pocket

Figure 20 shows the engine head of Engine 1, which is representative of the heads used in this project. The orientation of the engine head in the image is the same as in all the combustion images presented in this thesis. The two upper valves are exhaust valves and the lower valves are intake valves. The injector and spark plug are also visible. It should be noted that the spark plug is oriented such that the gap faces towards the injector. Experience from engine tests conducted in this project showed that orienting the spark plug like this increased combustion stability because the ground electrode does not cover the spray. The channel used for the cylinder pressure sensor is also indicated in the image.

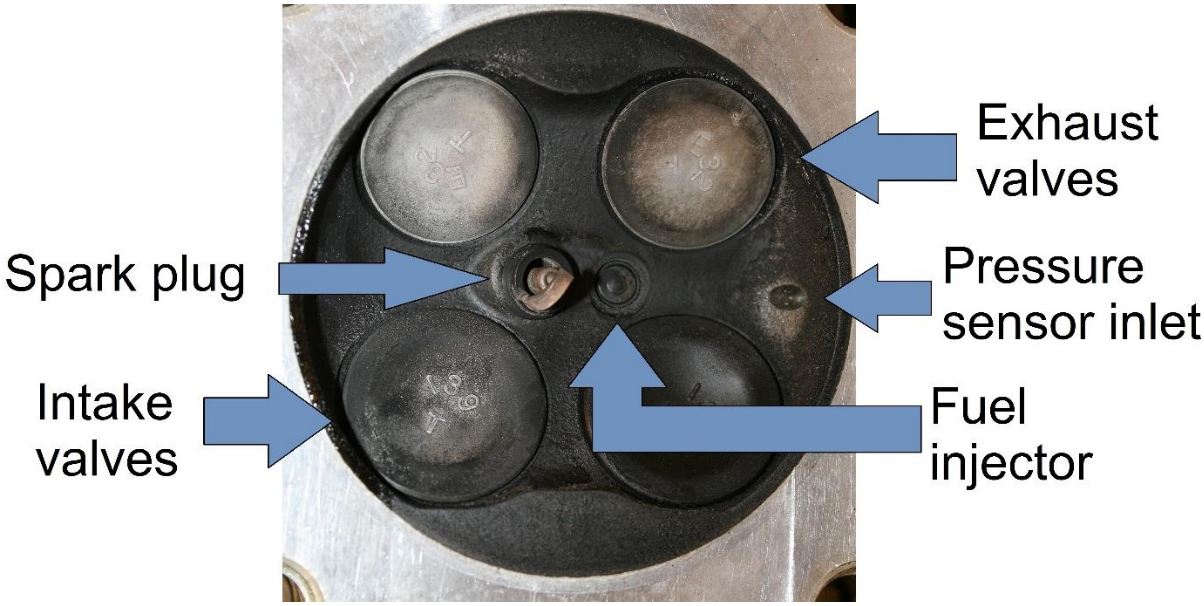


Figure 20 - One of the engine heads utilized in this study.

3.2 Ignition systems

Three types of ignition systems were utilized in this project. One standard coil on plug system with a single ignition coil, a Dual Coil Ignition (DCI) system, and a High Frequency Ignition (HFI) system. The first of these systems is the type most commonly found in modern four stroke Otto engines. It consists of a regular spark plug with a single coil that fires a single spark and will not be discussed further in this thesis. The second system is based on a standard spark plug but has two coils working in parallel. When one coil is discharged to create a spark, it is only discharged to half charge, then the second coil takes over and the first coil recharges until the second coil has depleted half its charge. This process continues, enabling the creation of a continuous spark that persists for roughly 7-8 times longer than a normal spark. This is beneficial in stratified operation because it increases the length of time for which the spark is in contact with the air-fuel mixture, increasing the likelihood of successful combustion. The third ignition system uses a different spark plug and a different control system. Instead of discharging a current between two electrodes as in a normal spark plug, the HFI system uses a spark plug with no ground electrodes. The ground is instead the air-fuel mixture around the spark plug. The control system creates a corona discharge using an alternating current with a frequency of around 50 Hz. The Corona discharge ignites the air fuel mixture, leading to combustion. The spark plugs for both ignition systems were designed to be bolted into the stock mounting hole of the engine head. Figure 21 shows images of the spark events with the HFI and DCI systems.

Spark event generated with the HFI system used in this study. Image courtesy of [69].

Spark event generated with the DCI system used in this study.

Figure 21 - Images of spark events, HFI system (left) and DCI system (right).

One benefit of the HFI system is that it utilizes multiple ignition points, so combustion is initiated in several places, which should lead to faster flame propagation. Another advantage is that the flash-over that occurs when a spark is created in a regular spark plug never occurs in the plug of the HFI system if the system is operated correctly. Flash-over results in high electrical losses, so the HFI system is less power demanding than traditional coil-based systems.

3.3 Emission measurement equipment

This section discusses the measurement equipment utilized in the studies. The discussion begins with a thorough description of the particulate conditioning systems developed at the beginning of the project, and ends with a description of the optical measurement tools that were used.

3.3.1 Standard emissions

In the metal engines, standard emissions were measured using the equipment listed in Table 2. All standard emissions were sampled roughly 15 cm downstream of the exhaust valve and fed through a heated hose to the exhaust pump. The heated hoses were electronically maintained at a temperature of 120 °C to avoid condensation. The samples were then transferred from the exhaust pump to the various emission measurement devices.

Table 2 - Standard emissions measurement equipment.

HC	Jum Engineering Model VE7
CO	Maihak Unor 611
CO_2	Maihak Unor 6N
NOx	Eco Physics CLD 822 M hr
O_2	Maihak Unor 611
Exhaust pump	Jum Engineering Model 222

3.3.2 First particulate sampling system

The PMP-system presented in [1] was modified and used to measure the solid particulate size distributions in the exhaust of the single-cylinder engine. The most significant components of the original design were retained, but some parts were removed. The system was designed to prevent nucleation and agglomeration by using a two-stage dilution process. In the first step, the sample was diluted with dry air at 350 °C. This reduced the partial pressure enough to prevent condensation of hydrocarbons; hot dilution air was used to prevent nucleation. Downstream of this first dilution step, the samples passed through an evaporation tube consisting of a tube and a surrounding heater. This slightly increased the sample's residence time to encourage evaporation of the few volatiles that did not evaporate in the first dilution step. Downstream of the dilution tube, the sample was again diluted, this time using air at a maximum temperature of 35 °C. During this step, the partial pressure was further reduced and the sample was cooled to a temperature suitable for the classifying instrument. If the classifier registered large amounts of volatiles (i.e. if the standard deviation of the mean value for particulates of <20 nm was large) even after two-stage dilution with the evaporation tube, a thermodenuder was employed. The thermodenuder consisted of a tube surrounded by active charcoal heated to a maximum temperature of 250 °C to adsorb any volatiles in the sample as it passed through the tube. After the dilution system, the sample was passed into a cyclone to filter out particulates larger than 2.5 µm at the inlet of the classifying instrument. The system thus consisted of the following components:

1. Sampling probe
2. Primary diluter
3. Evaporation tube
4. Secondary diluter
5. (Thermodenuder)
6. Cyclone
7. Particulate classifier

The dilution steps and evaporation tube (1-4) were combined in a single device, a particulate sample conditioner manufactured by Dekati (FPS4000). The thermodenuder was manufactured by the same company.

3.3.3 Second particulate sampling system

In some tests a simplified version of the previously described sampling system was used. This system was based on a two-step diluter and a heated cyclone. The emission sample was first passed into the cyclone to remove particulates larger than 2.5 µm and then diluted at a ratio of 5:1 with heated air before being transported through a heated pipe to a rotary disc diluter and diluted further. The degree of dilution in this second step depended on the sample's particulate content.

3.3.4 Size distribution classifier

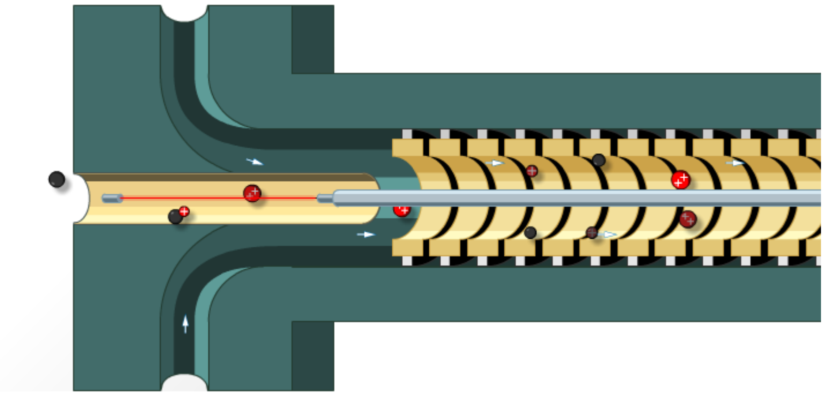


Figure 22 - Schematic view of the DMS500 classifying column.

A DMS500 mk2 particle analyzer was used to classify particulate size distributions in the engine tests. The DMS500 is an electrical mobility based measurement instrument capable of classifying particulates of between 5 nm and 2.5 µm. Particulates are fed into a column fitted with several electrometers, as shown in Figure 22. The particulates become electrically charged as they pass the corona charger upstream of the electrometers. In the center of the electrometers, a positively charged piston repels the particulates and drives them toward the electrometer rings. When the positively charged particulates strike an electrometer, their electrical charge is registered. The sums of the electrometer values are then sampled and converted into a particulate size distribution that is determined from the particles' electric charge and aerodynamic drag. The DMS500 has a response time of 200 ms and is therefore suitable for fast measurements.

3.3.5 Particulate mass measurements

Particulate mass measurements were acquired with an AVL Microsoot unit. The Microsoot unit utilizes photoacoustic technology in which particulates are excited with a laser to create a sound wave that is detected by a sensitive microphone. The intensity and strength of the sound wave is related to the mass of the particulates. This instrument detects the total mass of all particulates in a sample flow rather than the mass of the individual particulates. Water condensation was found to be a problem when the Microsoot was operated without exhaust dilution, so the instrument was always operated with samples diluted to at least 6:1.

3.4 Optical measurement equipment

Optical engines were used to visualize combustion behavior. Optical access was obtained through the cylinder liner and piston. Two high speed color cameras, a Phantom Miro V310 and a Vision Research V7.3, were used for simultaneous but not synchronized filming. In some cases, the cameras were fitted with filters that were placed in front of the camera to capture specific wavelengths. Two filters were used, a 308 nm narrow-band filter and a 610 nm high-pass filter. The former was used to capture OH luminescence and the latter to capture black body radiation generated by burning soot. The 610 nm filter was chosen based on experiences from previous studies in this project, which showed that combustion generated a peak at around 589 nm due to Na radiation. A filter of a slightly higher wavelength was used to exclude this peak. The filter of 308 nm was utilized in conjunction with a Hamamatsu C9548 image intensifier. Figure 23 is a depiction of the optical engine setup utilized in this project. Optical access is visualized through the cylinder liner and piston.

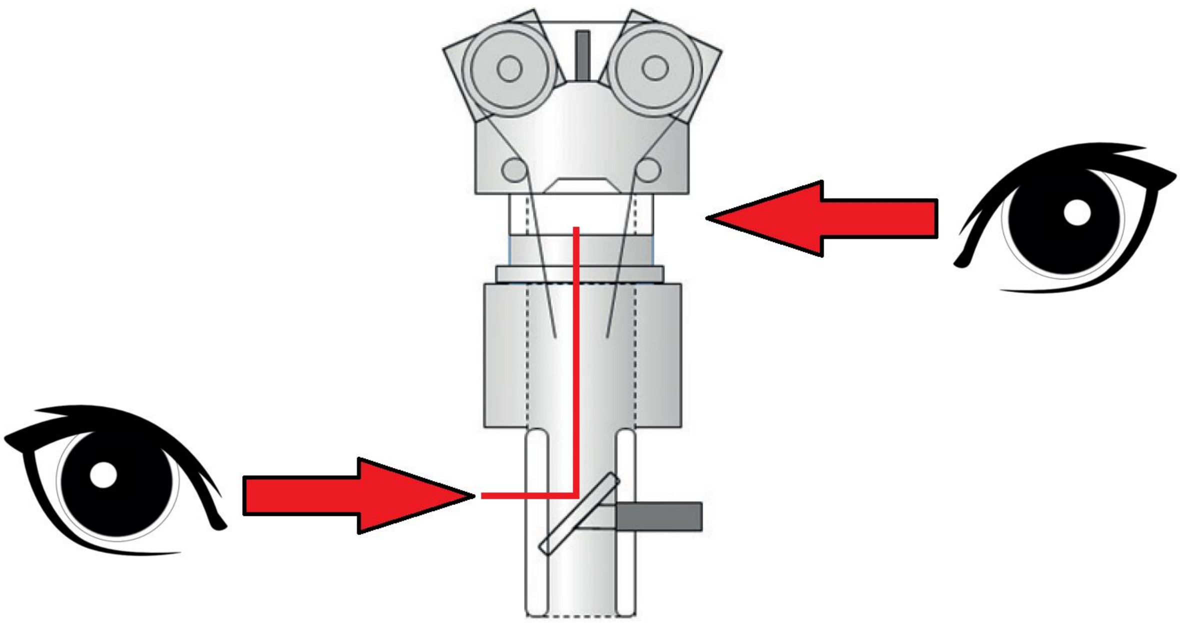


Figure 23 - Depiction of optical engine setup.

4 Experimental methodology

This chapter describes how the experiments were performed and what engine operating points were used. The chapter also gives the reader a brief introduction to the operational differences between metal and optical engines.

4.1 Engine operating points

All engines were operated at specific engine operating points depending on the purpose of the measurement campaign. The first five engine operating points were developed using a mini map provided by Volvo Cars. All engine operating points used in this work are presented in Table 3.

Table 3 - Engine operating points utilized in this project.

No	BMEP [bar]	Speed [rpm]	Exhaust Pressure [kPa]
1	1.5	1000	101.8
2	4	1000	102.4
3	1.5	2000	105.5
4	4	2000	107
5	2.5	1500	104.1
6	2.63	1200	Ambient
7	2.63	1500	Ambient

Engine operating points 1 to 5 were utilized in the beginning of this project. They were chosen to represent a typical (but not extreme) window in which a GDI engine could be operated in stratified mode. Engine operating point 6 was used in experiments with Engine 3, the single cylinder optical engine; the engine speed of 1200 rpm was selected to complement the frequency of an Nd-YAG laser utilized in the study. Engine operating point 7 was also used with Engine 3 in a study that was not dependent on the frequency of a laser, so there was more freedom with respect to the choice of engine speed. Most of these engine operating points were utilized in both stratified and homogenous mode. Stratified mode experiments were performed with the throttle wide open or with forced induction. In homogeneous mode, the engine was operated at lambda values between 0.95 and 1.4 depending on the study's focus.

4.2 Metal Engines

This section describes the methods used in the experiments performed with metal engines. To ensure stable and repeatable results, the standard emissions measurement instruments were calibrated with span gases each morning before running tests. During calibration of the measurement equipment, the engine was heated to an oil and coolant temperature of approximately 90 °C. The apparatus for measuring particulate emissions was cleaned and conditioned regularly depending on the contamination level. In the worst cases, when using stratified combustion conditions that produced large amounts of soot, the apparatus was cleaned up to three times daily. All engine operating points were considered stable if they satisfied the criteria in Table 4.

Table 4 - Engine stability criteria.

std (net imep) for net IMEP < 3.3 bar	0.1 bar
CV (net imep) for net IMEP > 3.3 bar	5 %

Before each run the engine was operated at a relatively high load of 5 bar BMEP at lambda = 1.1 to burn any soot particulates and oil residues in the engine; failing to include this step significantly reduced the accuracy of particulate measurements in some cases. This engine operating point was also used as a reference point to check for deviations in real time data that could indicate potential errors. Real time data was sampled using an AVL Indicom system with fast pressure sensors in the intake, cylinder and exhaust. Additional sensors such as temperature sensors and pressure sensors were installed at appropriate sampling points.

4.3 Optical Engines

Because optical engine components are sensitive and break if subjected to high temperatures and pressures, optical engines cannot be operated under the same steady-state conditions as metal engines. The optical engines utilized in this project were therefore restricted to an engine speed of 2000 rpm. This limitation did not hamper the experiments because none of the planned engine operating points involved a speed above this value. However, the optical engine's heat sensitivity did pose problems. Optical engines have very different heat loss profiles to metal engines because the comparatively low heat transfer capacity of glass means their surface temperatures are generally higher. The optical engines used in this study were never operated for more than 150 consecutive combustion cycles before being shut down and allowed to cool. Consequently, the engine never reached steady state operation. All optical engine tests were performed according to the following routine:

1. The engine was revved up to the specified speed with the ignition system active but the injection system disabled.
2. The injection and ignition settings were adjusted to the appropriate values for the planned experiment.
3. The engine settings were verified by briefly activating the injection system.
4. All measurement equipment was initiated and prepared for use.
5. The injection system was turned on, causing combustion to begin. Combustion was allowed to proceed for roughly 50 cycles to heat the engine's components.
6. Measurements were conducted for at most 100 cycles, after which the engine was turned off.
7. If the engine's exhaust temperature reached 350 °C, combustion was immediately halted to protect the engine.

Step 5 was performed because experience indicated that allowing the engine to heat up before beginning measurements stabilized combustion. Because of the operational limitations discussed above, results from optical engines cannot be directly compared to those from metal engines. However, they are useful for showing trends and can therefore be valuable complements to metal engine data or useful as stand-alone results. It is possible to perform heat release calculations based on cylinder pressure traces from an optical engine, but the different heat transfer characteristics of glass and metal necessitate the use of some assumptions when doing so [5] [6] [7]. The heat release calculations performed in this project were based exclusively on cylinder pressure traces from the metal engines.

5 Evaluation methods

Several methods for evaluating data were developed during this project. The evaluation of particulate and standard emissions measurements proved to be straightforward, but considerable effort was required to identify reliable methods for evaluating data from the optical engine. In cases where the raw measurements of a particular variable were reliable, only minimal post-processing of the raw data was deemed necessary. For example, the only post-processing of the particulate measurements involved averaging and computing the standard deviation of the mean using a script implemented in the Mathworks Matlab software package.

In contrast, the videos recorded with the high speed cameras required extensive post-processing. Each recording covered 70-100 cycles with almost 7000 frames in each file. The recordings were evaluated with respect to flame area and flame intensity. To this end, the videos were converted into an RGB-format in which each frame is represented by three matrices that record the RGB values for each pixel in that frame. Videos captured using the 308 nm narrowband OH-filter captured blue-emitting flames, which were readily distinguished using the data in the RGB matrices because the red and green matrices contained little significant information; most of the information needed to determine which pixels corresponded to a flame was found in the blue matrix. Conversely, in videos recorded using the high-pass filter or without a filter, there was important flame information in all three of the RGB matrices. Analysis of these images was further complicated by reflections in glossy parts of the combustion chamber or cylinder liner due to the high intensity of the non-premixed flames. Reflections were difficult to deal with because of the risk of misidentifying a reflection as a flame. This could potentially have been managed by reducing the aperture size, but that would have caused the loss of too much information on pre-mixed low intensity flames. Flame identification was therefore performed with an algorithm programmed in Matlab. Figure 24 shows a typical image before post-processing. Three areas of the image in which it was difficult to determine whether the observed brightness

Figure 24 – Cylinder regions that were difficult to analyze during the post-processing of raw combustion images.

corresponded to a flame or a reflection are highlighted. Most of the flame is in the center of the combustion chamber, and the flame's intensity is high enough to saturate the picture. Arrow 1 points to a real flame that is not connected to the main flame. The surface to the left of the flame is reflecting the flame's light, which must be accounted for when estimating the flame area. Arrows 2 and 3 point to areas that are just as yellow as the flame in the beginning of the combustion process. To deal with these difficulties, the threshold used to identify flames was adjusted throughout the cycle based on the maximum flame intensity. Specifically, the maximum intensities in each of the RGB matrices in the frame of interest frame were identified, and all pixels within a tolerable deviation of that intensity were taken to represent flames. The reflection problem is most severe when the combustion is most intense, and is thus less severe during the early and late stages of combustion. Consequently, the higher the peak intensity, the narrower the tolerable deviation. Figure 25 shows three images illustrating the progress of the post-processing algorithm for an unfiltered video. The raw image is shown on the left, while the center and right images show the blue and yellow flame images extracted by the algorithm. Everything not identified as a flame in these images is black, so the valves and combustion chamber are not visible. Any pixel identified as a flame can be used to count the flame area, to assess the intensity in that part of the flame, or to compute the total flame intensity.

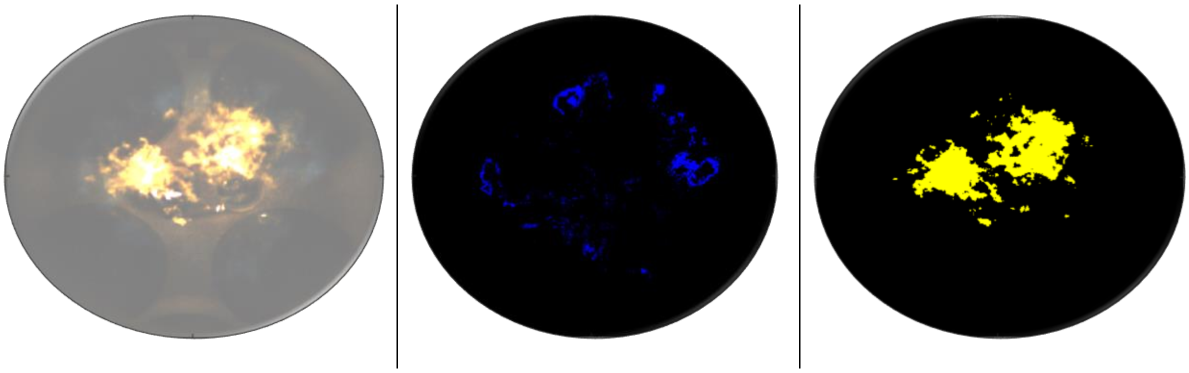


Figure 25 - Three images illustrating the post processing procedure. Left: raw image of a fuel-rich stoichiometric flame. Middle: blue flame pixels identified from the raw image. Right: yellow flame pixels identified from the raw image.

The algorithm was tuned for each batch of experiments until a good agreement between the raw-image and post-processed image was achieved. When a pixel was identified as belonging to a flame, the cumulative maximum intensity or flame area for that particular frame was updated accordingly and the resulting data were used to evaluate the overall combustion process.

6 Summary of results

This chapter briefly summarizes the main results obtained in the project. A clear progression can be discerned upon looking at the work as a whole. The project began with a study on the influence of various parameters on particulate emissions. This revealed several parameters that could potentially influence particulate emissions, namely the addition of exhaust residuals, forced induction, increased fuel injection pressure, ignition system, and the influence of air motion. This chapter discusses the progress made during the project and how it contributed to an improved understanding of particulate emissions.

6.1 Influence of EGR

One of the studies conducted in the project was an investigation into the effect of recirculating exhaust gases on combustion. Unfortunately, some of the planned study components were not completed and so the study was never published. However, some interesting results were obtained. Exhaust gas recirculation is typically performed to reduce NOx emissions. It can be used for this purpose in stratified combustion, but the focus of our investigation was to determine its effect on particulate emissions. As noted earlier in this thesis, particulates form and grow by agglomerating with other particulates. The rate of this process depends on the number of available particulates. Adding extra particulates by recirculation of exhaust gases would thus be expected to increase the number of particulates and so it would be reasonable to anticipate that larger agglomerates would be formed. As shown below, larger particulates was not always formed.

The study was performed with a single cylinder metal research engine (Engine 1) that was fitted with an external routing system used to adjust the amount of recirculated residuals (more commonly referred to as external EGR) admitted to the combustion chamber. The engine was operated at engine operating points 1-5 as defined in Table 3, with 0, 5 and 10% external EGR. Figure 26 shows the particulate size distributions obtained at 1000 rpm and 4 bar BMEP with various external EGR levels.

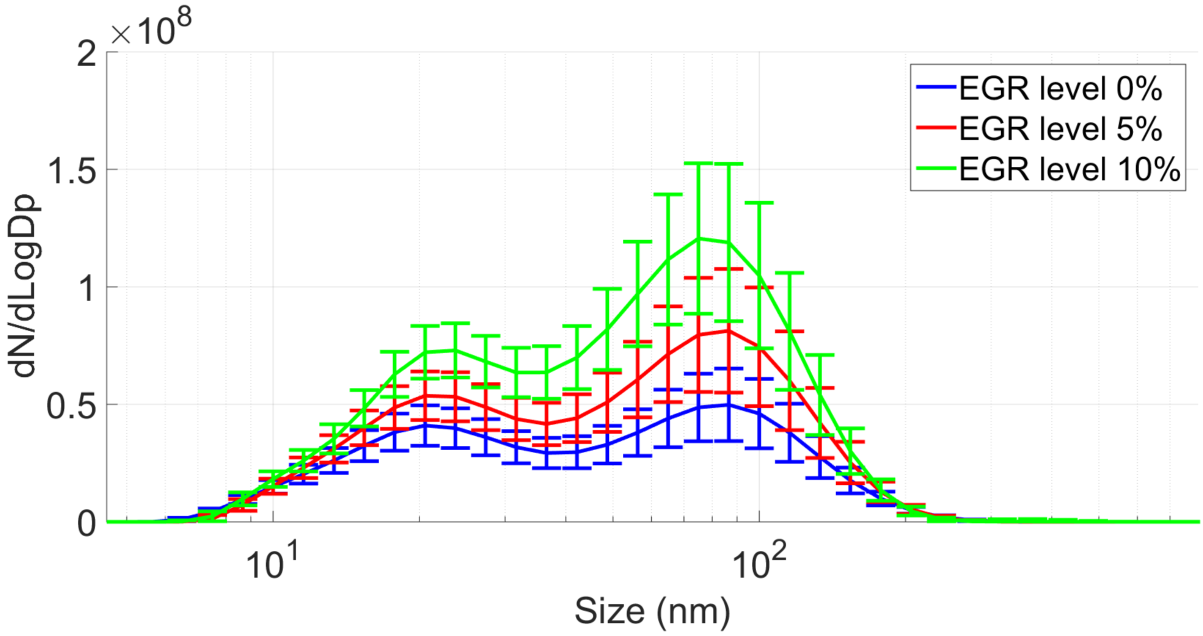


Figure 26 - Particulate Size Distribution with various levels of external EGR at 1000 rpm and 4 bar BMEP.

The particulate size distribution plots show that the number of particulates increased with the EGR rate. However, in contrast to expectations, the particles did not get larger, only more numerous. This suggests that there may be a maximum size beyond which particulates cannot grow by agglomeration. All published studies on stratified combustion have reported a maximum agglomerate size of around 100 nm. The bars on each curve show the standard deviation of the mean value. The standard deviation appears to be higher for the agglomerates than for the nucleation mode particulates; this is probably due to the increased time needed for agglomerates to form, indicating that their formation is cycle-dependent. Further, increased levels of EGR increased the overall standard deviation. This is probably due to a combination of a reduction in combustion stability and the previously discussed cycle dependence. Figure 27 shows the nitric oxides and hydrocarbon emissions generated at the same engine operating points.

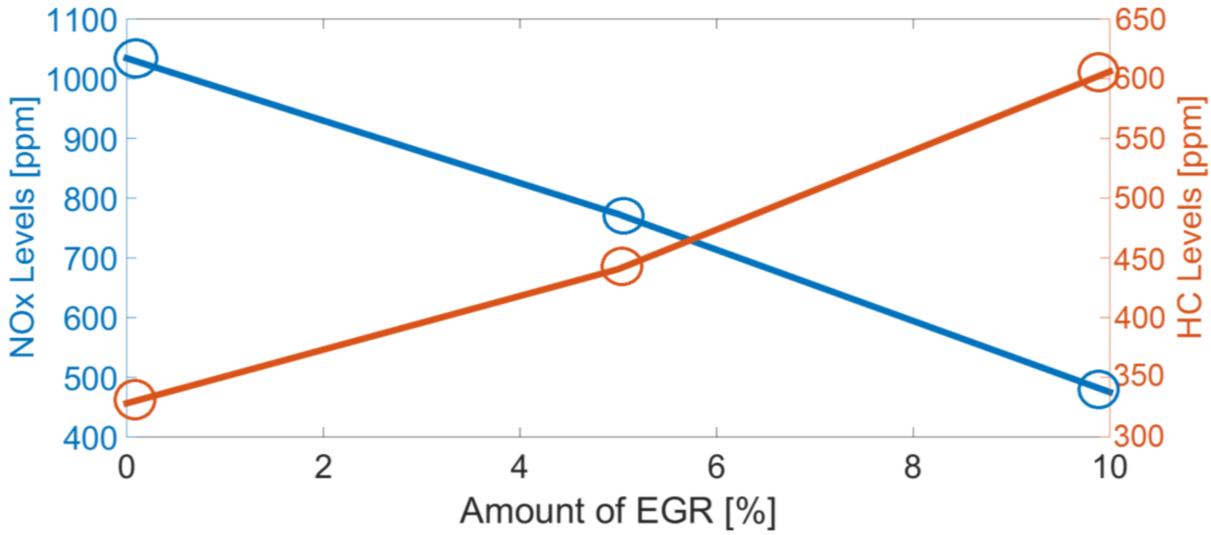


Figure 27 - Nitric Oxides and hydrocarbon emissions generated in a metal engine operated at 1000 rpm and 4 bar BMEP with various levels of external EGR. Measurement points indicated with circles.

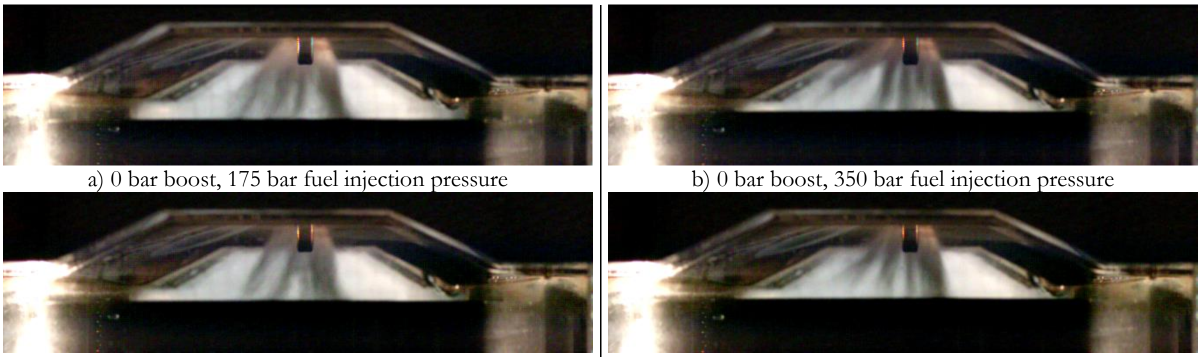
The graph shows the levels of nitric oxides in ppm on the left y-axis and the levels of hydrocarbons in ppm on the right axis. The x-axis represents the amount of EGR, ranging from 0 to 10 %. Increasing the level of EGR clearly reduces NOx emissions at the cost of increasing those of hydrocarbons and particulate matter. The increase in hydrocarbon emissions is probably due to a more severe over-dilution because the outer perimeter of the fuel cloud is mixed with exhaust gases instead of air as would occur in the absence of EGR. The dilution of fuel with exhaust gases creates an environment in which the fuel can no longer be oxidized during the combustion process.

6.2 Influence of forced induction

This study was performed to see how increasing the air mass in the cylinder would affect combustion and particulate emissions. Experience from previous work with the studied engine indicated that the fuel spray tended to impinge on the piston crown during stratified combustion. This reduced fuel evaporation, leading to diffusion flame formation and combustion in excess air. The consequences of this are shown in Figure 28, which depicts the top of a piston used in earlier stratified combustion experiments conducted within this project.

Figure 28 – The piston used in Engine 1 (as defined in Table 1) after operation in stratified mode. Extensive soot deposition has occurred on the piston's upper surface.

A layer of soot is clearly visible on the piston's upper surface with six distinct craters formed by the impingement of fuel spray rays. The hypothesis of this study was that reducing the penetration length of the fuel sprays would reduce this impingement. As a beneficial side-effect, forced induction should also increase the amount of oxygen in the cylinder, which was expected to promote soot oxidation. This study was performed with two engines, one optical and one metal. The optical engine enabled optical access through a window in the piston and a glass liner. The metal engine utilized the same gas exchange system, fuel injector and ignition system as the optical engine; aside from the modifications to permit optical access, both engines had identical hardware setups. This made it possible to obtain steady state combustion data for each operating point with the metal engine and then study the corresponding optical data from the optical engine. The optical engine was used to record spray and combustion patterns while the metal engine was used to measure emissions and combustion characteristics. Figure 29 shows side-on images of fuel sprays formed at injection pressures of 175 and 350 bar, and overpressures of 0 and 0.4 bar. The images were sampled at an engine speed of 1500 rpm with a fuel mass that would correspond to 3.62 bar BMEP in a metal engine. The images correspond to the occasion of maximum spray penetration length and not the exact same CAD. In absolute terms of CAD they vary by 2-3°.



a) 0 bar boost, 175 bar fuel injection pressure
b) 0 bar boost, 350 bar fuel injection pressure
c) 0.4 bar boost, 175 bar fuel injection pressure
d) 0.4 bar boost, 350 bar fuel injection pressure

Figure 29 - Fuel spray patterns formed at injection pressures of 175 and 350 bar, with boost pressures of 0 and 0.4 bar.

The results from the optical engine do not reveal any discernible effect of overpressure on the penetration length. These results are consistent with the findings of Maligne et al. [28], who studied the fuel film thickness generated on the piston surface by various injectors at three different back pressures. They found that the back pressure did not significantly affect the film thickness, but the ambient temperature in the test volume had some influence. Increasing the fuel injection pressure appeared to concentrate the fuel spray, as shown by comparing Figure 29(a) and (b). Additionally, Figures 28(d) and (b) suggest that one of the rays diverged more from the rest of the spray. The engine's spray pattern was optimized for homogenous operation, with the rays being directed in the axial direction of the cylinder. In stratified mode, the clearance in this direction is reduced, so most of the rays hit the piston with the exception of one that appears to be directed more towards the intake side of the engine (i.e. the left side). This suggests that modifying the injector spray pattern for stratified operation could potentially reduce spray impingement on the piston crown and thus soot emissions.

Figure 30 shows the particulate size diagram for four boost levels and natural aspiration at an engine operating point of 2000 rpm and 1.5 bar BMEP. The x-axis represents the particulate sizes and the y-axis represents the number of particulates. The blue curve shows the particle size distribution produced with natural aspiration. This mode clearly generates the highest particulate emissions. Raising the boost pressure reduces particulate emissions; the agglomerates become substantially fewer in number but remain unchanged in terms of size, whereas the nucleation mode particulates become more abundant at higher boost pressures. The measurement technique used in this work did not include a volatiles removal step, so many of the observed nucleation mode particulates are likely to have been volatiles such as hydrocarbons in actuality. For further information and more conclusions from this study, see Paper 3 and Paper 4.

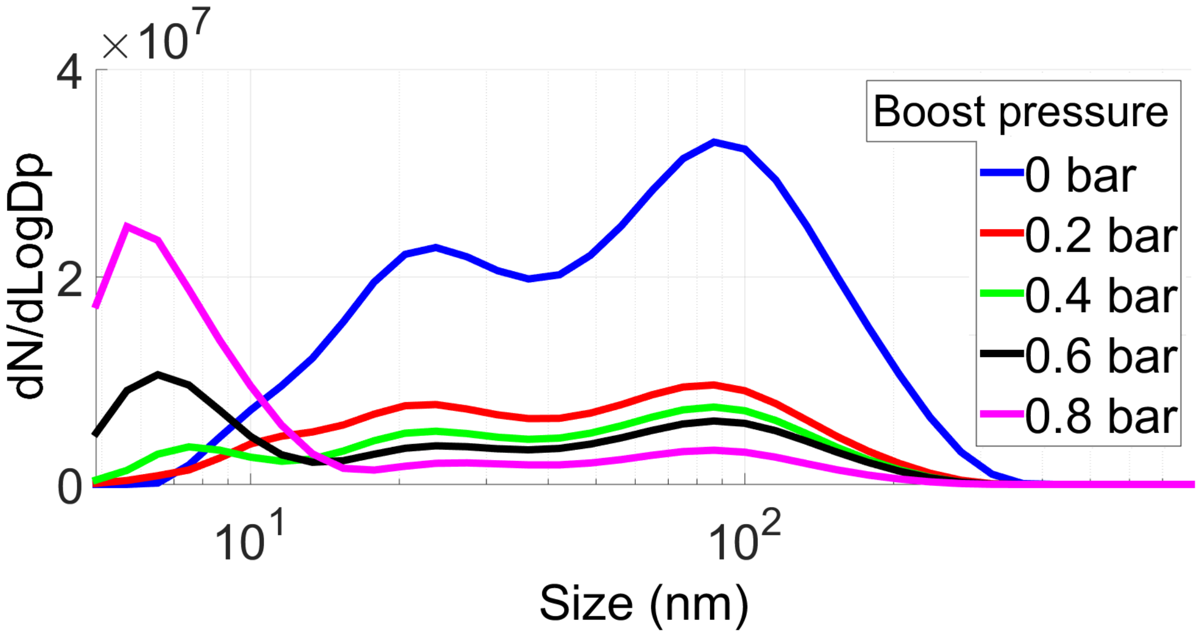


Figure 30 - Particulate Size Diagram showing the impact of forced induction at 2000 rpm and 1.5 bar BMEP.

6.3 Influence of fuel injection pressure

The influence of injection pressure was investigated in two separate studies that are discussed in Paper 1 and Paper 5. The first study was performed with a metal engine during the early stages of the project and focused on particulate measurement, whereas the second study used the optical engine to characterize the combustion process. In the first study, a solenoid-actuated multi-hole injector with a maximum injection pressure of 200 bar was used, whereas the second study used a different solenoid-actuated multi-hole injector with a maximum injection pressure of 350 bar. This increase in injection pressure tolerance is broadly consistent with the improvements in fuel injector performance that have occurred in recent years; as of the time of writing, injectors capable of handling pressures of up to 600 bar are available. It is likely that the fuel injection pressures used in gasoline injectors will follow the same trend seen with diesel injectors; modern diesel engines typically use fuel pressures in the range of 2500 bar. Figure 31 shows results from the first study.

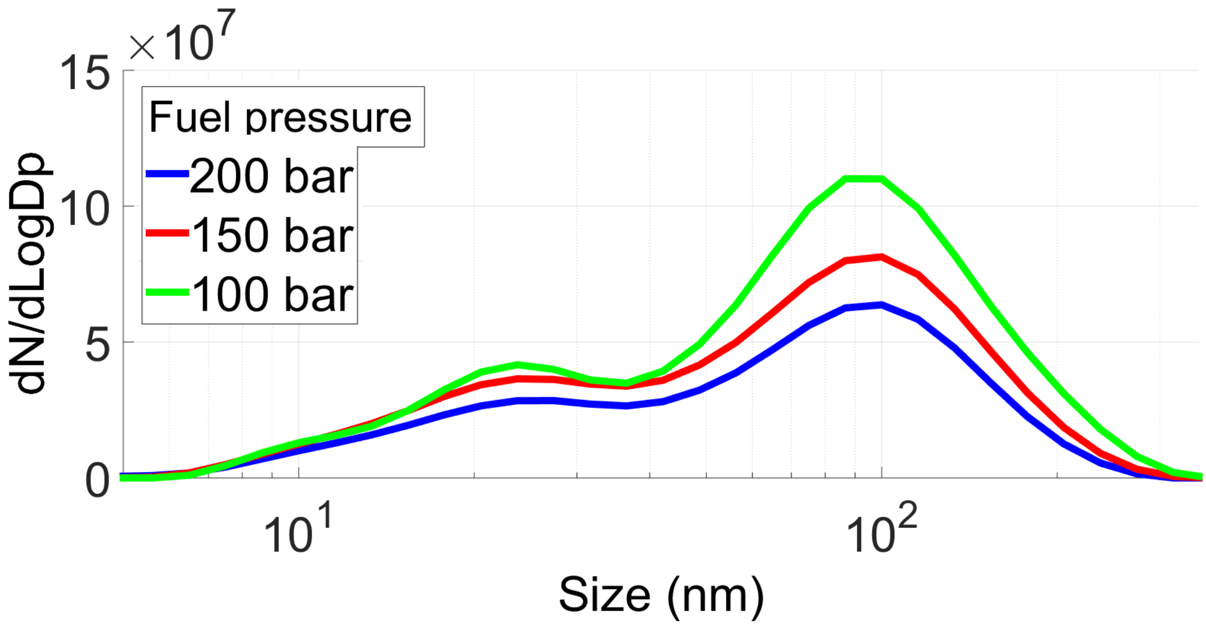


Figure 31 - Particulate Size Diagrams for three different fuel injection pressures at 2000 rpm and 4 bar BMEP.

Data were gathered at an engine speed of 2000 rpm and a load of 4 bar BMEP. The plot suggests that raising the fuel injection pressure reduces the number of particulates because there is a continuous decrease in particulate number on raising the injection pressure from 100 to 200 bar. However, there is no discernible effect on particle size. These results are similar to those achieved with forced induction, as discussed in the preceding section. However, raising the injection pressure also reduces the abundance of nucleation mode particulates, suggesting a more efficient fuel evaporation process. This result is consistent with previously reported findings for both stratified and homogenous combustion. For example, Piock et al. [26] investigated the impact of fuel pressures up to 400 bar on a gasoline engine and found a corresponding reduction in particulate number.

Another way of studying soot formation is to utilize optical filters in an optical engine. This approach was adopted in the second study on the effect of varying the injection pressure. The optical engine was fitted with a different gas exchange system to that used in the earlier study, and a fuel injector that was operated at injection pressures of 175 and 350 bar. Particulate measurements in optical engines do not yield reliable results, so another method was used to measure soot formation. The combustion process was recorded using a high speed video camera fitted with a high pass 610 nm filter that only transmits radiation with a wavelength above 610 nm, i.e. soot-derived black body radiation. This method excludes radiation from most non-soot species, making it easier to study the soot alone. Figure 32 shows the results obtained in these experiments.

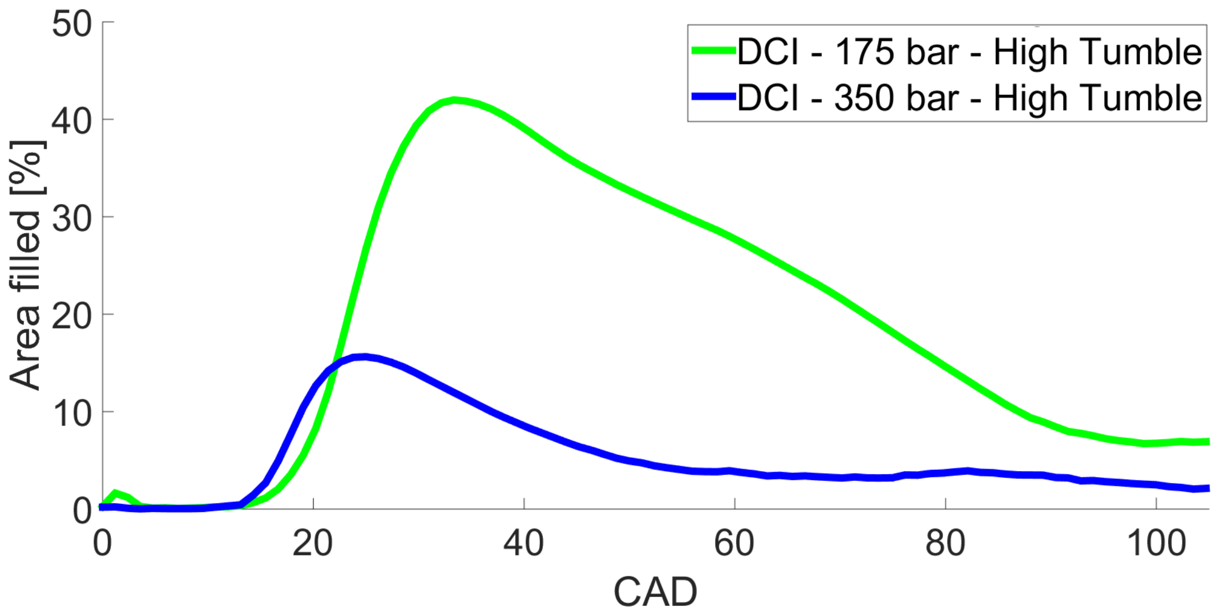


Figure 32 – Area illuminated by soot as seen from piston view versus CAD, sampled at 175 and 350 bar fuel injection pressure at 1200 rpm and 2.63 bar BMEP.

The y-axis values in this graph correspond to the proportion of the cylinder containing burning soot when viewed through the piston, while the x-axis values represent crank angle degrees after the start of ignition. The green and blue curves show results obtained at injection pressures of 175 and 350 bar, respectively. It is clear that the degree of soot illumination is reduced at the higher injection pressure. It should be noted that these results only provide direct information on soot oxidation, not soot formation. However, they clearly imply that increasing the injection pressure reduces soot formation, in keeping with the results presented in Figure 31.

6.4 Influence of the particulate sampling system

Two separate studies on the influence of the sampling system used to measure particulate levels were conducted, as discussed in Paper 1 and Paper 2. The first study was performed at the beginning of the project to gain more knowledge about the available sampling systems, while the second was performed during an attempt to develop ways of measuring low particulate levels. Figure 33 show particulate size diagrams based on data acquired with the stock DMS500 sampling system and the two-step FPS4000 sampling system with the denuder, respectively (see the measurement equipment section for details of these two systems).

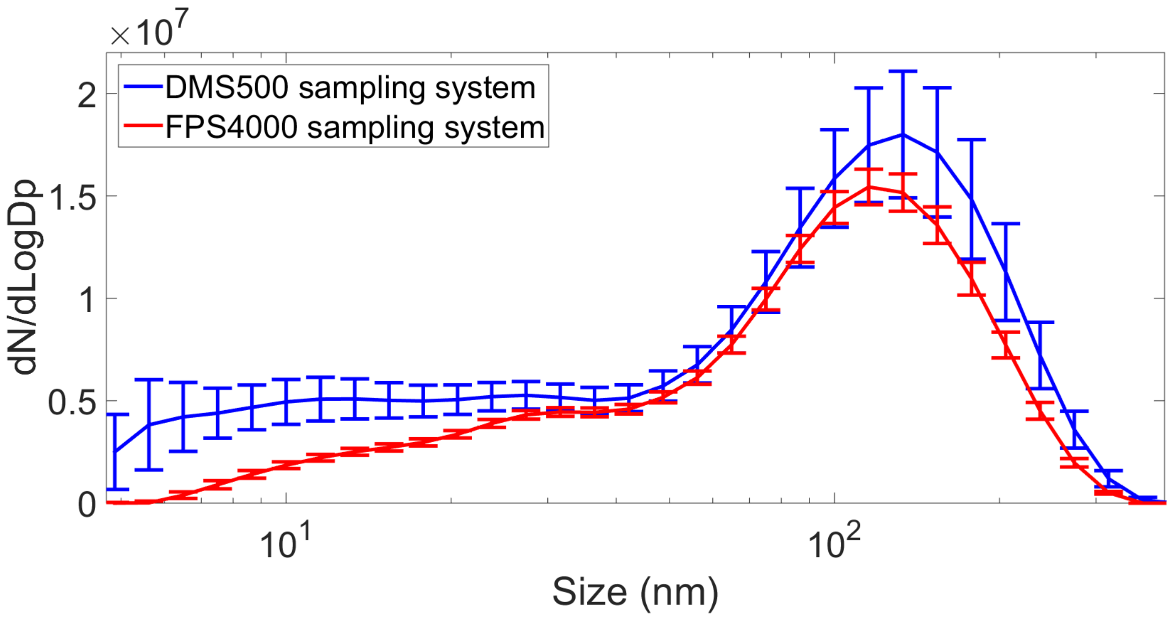


Figure 33 - Size distribution of particles (means and standard deviations) emitted in tests at 1000 rpm and 1.5 bar BMEP in stratified mode recorded using the FPS4000 sampling system with thermodenuder (red) and the DMS500 sampling system (blue).

The size diagrams clearly show how changing the sampling system changes the results obtained. The FPS4000 system, which uses a two-stage high temperature dilution process and a thermodenuder seems to produce more stable results in terms of the standard deviation of the mean. Additionally, it appears to reduce the number of detected nucleation mode particulates. The purpose of the thermodenuder is to remove volatile species, a majority of which are small and occur in the nucleation mode region. It thus appears that most of the smaller particulates emitted by this engine are volatiles. Overall, however, the particle number estimates produced by the two systems are quite similar, particularly for agglomerates. Swanson et al. [20] performed a similar study in which thermodenuders and catalytic strippers were used to remove volatiles. Their results suggested that the use of thermodenuders could produce semi-volatile particle artifacts with sizes of 3-10 nm due to incomplete removal of evaporated compounds in the thermodenuder. No such effect was found in our investigations.

6.5 Influence of ignition systems

The first studies conducted in this project were all performed with conventional ignition systems. However, because the engines' gas exchange systems were optimized for homogenous operation, the ignition systems were not properly tailored for stratified operation. Consequently, the ignition window was narrow and the system was highly sensitive to cyclic variations. If the operating conditions drifted outside the ignition window, misfires occurred. Of the five chosen engine operating points, the one combining high engine speed and low load caused the most misfires because the high speed reduced the available mixing time, and the small mass of injected fuel created an unstable mixing process. The conventional ignition system was a coil on plug system with a single coil mounted directly on top of the spark plug, capable of firing one spark per combustion event. This spark was only present for a short period, so misfires were common because of the unreliable fuel mixing process. To alleviate the problem of unstable ignition, two alternative ignition systems were tested: a DCI system and a HFI system, as described in the chapter on experimental equipment. Both these systems improved combustion stability and increased the ignition window, with the HFI system giving the best performance.

The DCI system was tested in both a metal engine (Engine 1) and an optical engine (Engine 3), whereas the HFI system was only tested in an optical engine (Engine 3). The first study was conducted to study the DCI system and its impact on particulate size distribution, while the second was performed with the optical engine to characterize the two systems' effects on combustion. Because the optical engine was used, no particulate size distribution data were generated. Figure 34 shows the results obtained using the DCI system in the metal engine with various current levels.

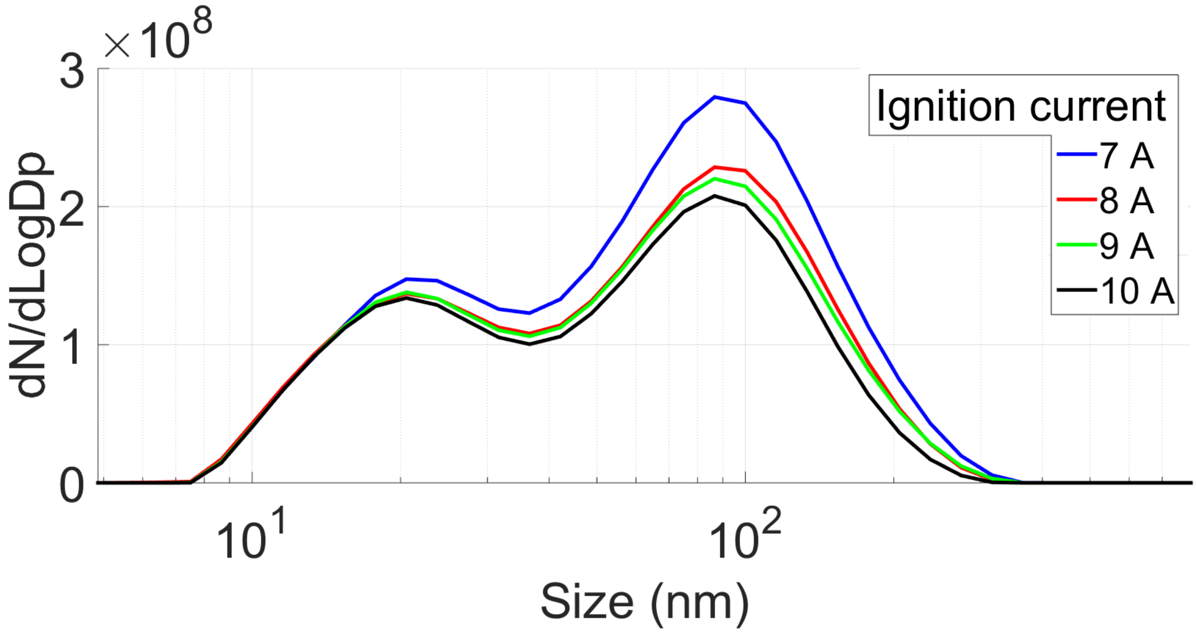


Figure 34 - Particulate Size Distribution at various current levels for the DCI system at an engine speed of 1000 rpm and 4 bar BMEP.

Up to a point, the combustion stability increased with the spark duration, although this effect plateaued at higher durations. Additionally, increasing the power reduced the number of emitted particles. This was attributed to a faster flame initiation, which reduced the combustion duration and thus left more time for soot oxidation. The results of this study were never published.

The second study compared the flame front development achieved with the DCI and HFI systems in an optical engine (Engine 3). High speed filming was performed with an OH filter of 308 nm and a high-pass filter of 610 nm; the results obtained on OH illumination are presented in Figure 35.

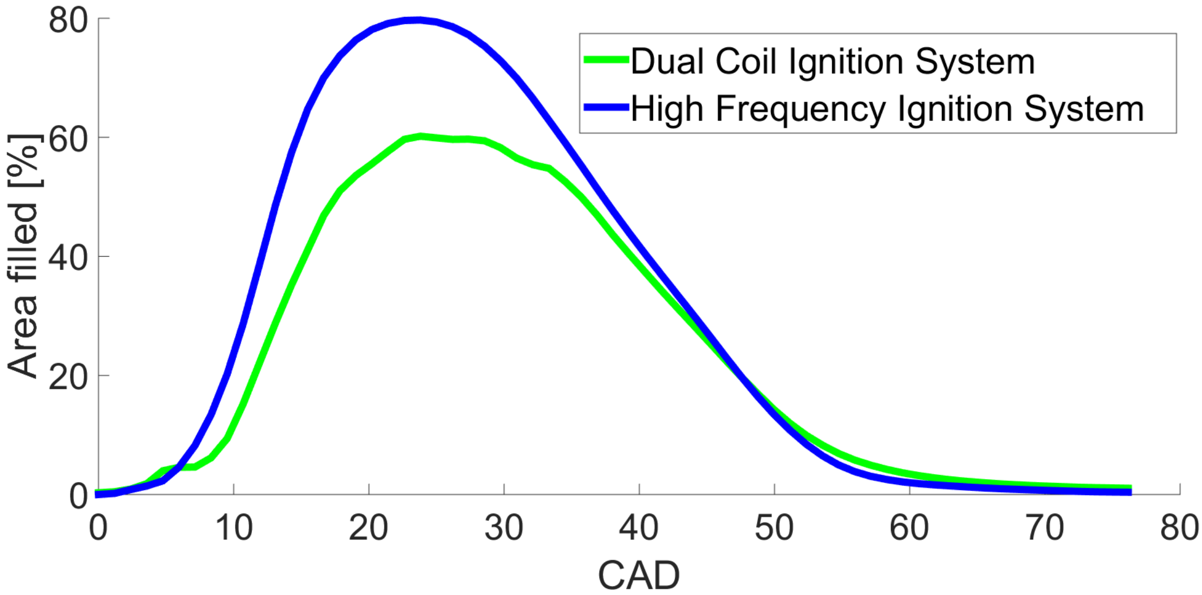


Figure 35 - Area illuminated by OH radicals as seen through the piston for the DCI and HFI systems.

The graph represents the area of the cylinder illuminated by OH radicals when viewed through the piston. The x-axis represents the time since ignition (in CAD), and the y-axis represents the percentage of the cylinder area with detectable OH illumination. Measurements were performed at a fuel injection pressure of 350 bar at 1200 rpm and 2.63 bar BMEP, and ignition was initiated at the same CAD value in both cases. The flame area for the HFI system was around 20% greater than that for the DCI system, probably because the HFI system delivers more power during the ignition phase, leading to faster combustion and a reduced risk of over-diluting the end gases. In addition, the DCI system appeared to yield comparatively slow combustion during the early stages of the ignition phase: by around 7 CAD after ignition, the flame created by this system had begun to grow, but its growth rate then declined sharply before increasing again at around 10 CAD. This behavior was not seen with the HFI system. The results of this study are discussed further in Paper 5.

6.6 Influence of load and speed

Figure 36 shows a particulate sampling diagram of engine-out particulate emissions from Engine 1 measured at engine operating points 1-5. Notably, increasing the load appears to increase the number of emitted particles. This is probably related to the piston wetting observed during the experiments: as the mass of injected fuel increases, so does the level of piston wetting and thus the amount of soot produced by flames burning on the piston surface. A less readily explained effect is that increasing the engine speed appears to reduce the size of the agglomerates: at constant load, increasing the speed from 1000 to 2000 rpm appears to make the agglomerates 20-50 nm smaller. This may be because higher speeds reduce the time available for agglomerate growth.

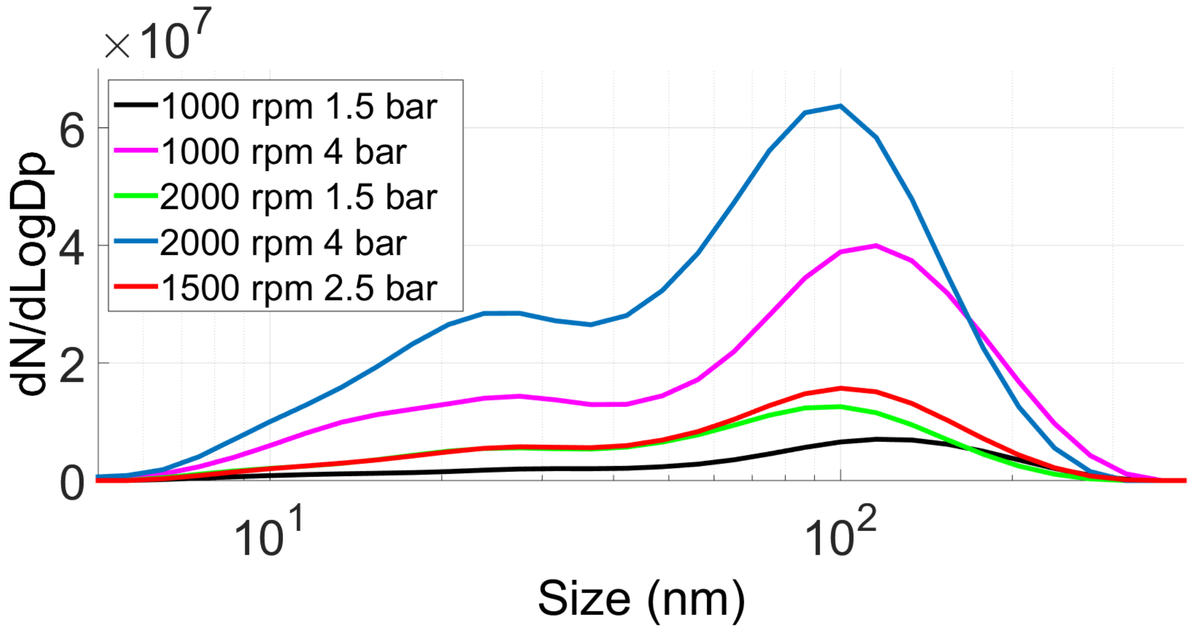


Figure 36 - Particulate size diagram showing the influence of engine load and speed at five various engine operating points.

7 Discussion and conclusions

This thesis describes work conducted during a PhD project on stratified combustion in SGDI engines at Chalmers University of Technology. The studies were primarily experimental, and aimed to identify ways of reducing particulate emissions. At the start of the project, the work focused on finding suitable ways of measuring particulate emissions. Several types of measurement methods were investigated, and it was found that a protocol derived from a PMP standard and another based on a manufacturer-specific sampling system both gave good results. Consequently, both methods were used in the later stages of the project. Subsequent investigations focused entirely on stratified combustion rather than particulate measurement per se. One of the first studies performed in this project examined the influence of various operating parameters on soot formation. It was during this work that the pronounced difference between particulate emissions generated during homogenous stoichiometric and lean stratified combustion was discovered. Stratified operation resulted in heavy soot emissions with a large agglomerate peak, whereas homogenous mode showed the opposite behavior, generating few agglomerates and large quantities of volatiles. At the start of the project, it was hypothesized that this difference was largely because stratified combustion generated a greater proportion of non-premixed flames as a result from the short mixing times. However, when the engine was dismantled after the experimental campaign, it was found that the piston was covered in soot and showed clear marks from impinging jets from the fuel injector (see Figure 28). The increased soot levels could therefore indeed be derived from a greater portion of non-premixed flames, not mainly in the free air as originally supposed but as pool fires on the piston. The engines utilized in this project were all optimized for homogenous operation, and were designed to produce spray patterns optimized for that mode. It was therefore difficult to reduce this piston impingement, so efforts were focused on minimizing the impact of the wall wetting. Consequently, efforts were focused on finding ways to manage the formation of non-premixed flames and the resulting particulate emissions rather than preventing the formation of such flames altogether.

To this end, the effects of varying the fuel injection pressure, operating parameters, boost pressure, ignition systems, exhaust gas recirculation and air motion were investigated. The boost pressure investigations provided a minor breakthrough because they showed that increasing the boost pressure reduced stratified mode particulate emissions more effectively than any previously tested change in operating parameters or fuel injection pressure. The effect of EGR on particulate emissions in stratified mode was then investigated (although the results of this study were not published), revealing that introducing 10% external gas recirculation approximately halved NOx emissions but increased particulate emissions. A subsequent study on boost in an optical engine revealed that higher boost pressures did not significantly reduce wall wetting of the piston. Moreover, there was no apparent correlation between the rate of heat release and boost. The boost-related reduction in particulate emissions was therefore attributed to an increase in the cylinder's temperature or oxygen content. Despite the promising results obtained by boosting, combustion stability remained problematic; it seemed that the engine's conventional single coil ignition system was incapable of supporting misfire-free stratified combustion. Therefore, tests were performed with a DCI system that delivered good results. A later study comparing the DCI system to a new HFI system revealed that the latter achieved a significantly higher burn rate. In addition, the effects of combining the new ignition systems with an elevated injection pressure were tested. Increasing the injection pressure improved mixing and significantly reduced the occurrence of non-premixed flames. The same study revealed that the impact of raising the injection pressure was reduced when the turbulence level was high. This study was supposed to be run in a metal engine to verify that these conditions reduced particulate emissions. Unfortunately, the corresponding tests were never completed. However, one would expect that combining an increased fuel injection pressure with the HFI system would significantly reduce particulate emissions. Towards the end of the project, it was reported that injectors capable of tolerating injection pressures as high as 600 bar were in development [55]. Based on the results achieved with a fuel injection pressure of 350 bar, it is likely that such high injection pressures would substantially reduce particulate emissions. Thus, the results obtained in this work strongly suggest that it will be possible to develop a viable stratified combustion system that can produce acceptable particulate emissions using commercial hardware.

This project has contributed to the understanding of stratified combustion in SGDI engines by presenting a series of experimental datasets acquired using engines designed for homogenous operation. New methods for reducing soot emissions have been identified, along with some strategies that are ineffective for this purpose and should be excluded from future investigations in this area.

7.1 Further work

Many of the ideas that were proposed during this project could not be investigated due to a lack of time. I believe that a natural next step would be to conduct a combined study using the most advanced available hardware to see how well the engine operates in stratified mode when boosting is combined with a HFI system, exhaust gas recirculation and an elevated fuel injection pressure. It would probably be sufficient to run these tests in a metal engine, but optical engine studies would provide interesting complementary information. Ideally, these experiments should be performed using boost pressures that are realistic given the chosen engine operating point; some of the boost pressures used in this project would be unlikely to be used in a production vehicle operating at the tested load and speed, although the results obtained clearly illustrated an interesting general trend. The HFI system would probably perform better if the injection and ignition systems were optimized to utilize all five ignition electrodes instead of only 2 or 3 as was usually the case in this project. Additionally, although the effect of exhaust gas recirculation on particulate emissions during stratified combustion was investigated in this project, the results obtained were never published. It would be desirable to repeat these experiments with more precisely controlled temperatures of the recirculated exhaust gases because it is likely that heated and cooled EGR would have different effects. Finally, when this project was begun, the engines were fitted with a prototype injector that could tolerate injection pressures of up to 200 bar. While it was state-of-the-art at the time, injectors capable of tolerating 350 bar became available towards the end of the project, and were tested with good results. The ongoing development of injector technology suggests that injectors capable of tolerating still higher pressures will become available soon. The results obtained in this work suggest that it would be beneficial to use the highest possible injection pressure in stratified mode. Moreover, it would be interesting to perform tests with injectors capable of delivering multiple injections per combustion cycle because it has been shown that multiple injections can reduce the spray penetration length and hence piston wetting. At the time of writing, the best multi-injection systems are based on piezo crystals. Piezo-injectors also produce beneficial umbrella-like spray patterns because of their outward-opening needles. The engine studied by Ricardo as discussed in the introductory section utilized a Piezo injector capable of up to 4 openings per cycle [2].

Once the studies suggested above have been completed, it would be sensible to study soot production and oxidation during stratified combustion using laser-based CAD-resolved methods. Most of the information on stratified combustion in the open literature is based on research performed on combustion systems that were optimized for stratified combustion to varying degrees. Conversely, the engines examined in this work were optimized solely for homogenous operation. Consequently, their airflow patterns, spray targeting, and combustion chamber design were all presumably suboptimal for stratified combustion. Consequently, results obtained in earlier studies on stratified combustion cannot be directly compared to those presented in this work. Laser-based methods such as LII and LIF could be used to obtain instantaneous information on where in the combustion chamber soot is formed and where fuel is present. Additionally, Mie scattering could be used in conjunction with LIF and PIV to study the ignition sequence and the relationships between early flame kernel development, air motion, and fuel distribution. By acquiring a more thorough understanding of the ignition sequence in this way, it may be possible to find methods for accelerating the ignition sequence. I hypothesize that the rate of soot formation would not be a problem if the rate of soot oxidation could be increased to match it. Accordingly, speeding up the ignition sequence leaves more time for soot oxidation and thus reduces particulate emissions. Efforts were made at the end of this project to fit an optical engine with a PIV-system and apparatus for LIF and Mie scattering. Despite almost four months of work, no practical setup was identified. However, large amounts of knowledge were gained during these four months, which will hopefully be useful some day.

8. Summary of papers

Experimental Investigation of Soot in a Spray-Guided Single Cylinder GDI-Engine Operated In a Stratified Mode

This article discusses the first study performed in this project, which was undertaken to acquire a basic understanding of stratified combustion in the engine provided for the project. The study utilized a single cylinder metal engine with a predominant tumbling air motion that had a small degree of swirl. The engine was operated with a fuel injector capable of a 200 bar injection pressure. The influence of several parameters on soot formation was examined, including the fuel injection pressure (100, 150 and 200 bar), combustion phasing (+/-2 CAD for 50% MFB), homogenous and stratified modes, the sampling equipment used, and the effects of load and speed. The main finding of this study was that stratified mode produced substantially more soot than homogeneous mode. Increasing the fuel injection pressure reduced the particulate number but did not change the size distribution. The combustion phasing had little influence, probably because the range of phasings studied was too narrow. Particulate number emissions was found to increase with increasing load but to decrease in size with increasing speed. This investigation also provided insights into the challenges of particulate measurement. The results were presented by the author of this thesis in 2013 at the 11th International Conference on Engines & Vehicles in Capri, Italy, SAE Naples section.

The author designed and set up the particulate measurement systems, conducted the experiments, post-processed the data, and wrote the paper.

Measurements of Particulate Size Distribution from a GDI Engine Using a Nafion Dryer and a DMS500 without Sample Dilution

The study described this article is the “black sheep” of this thesis. It was performed on a four cylinder engine operated in homogenous mode, and its main aim was to identify a sampling system that could successfully measure low levels of particulate emissions generated by engines operating in lean homogenous mode. Normal sampling systems rely on sample dilution to prevent volatiles and water vapor from condensing because condensation can cause erroneous measurements or even electronic failures in the classifier. However, dilution reduces the number of particulates per unit volume by at least a factor of five, which will present problems if the sampling device is operating near its detection limit. The novel sampling system tested in this study utilized a Nafion-dryer in combination with a thermodenuder. Nafion-dryers are normally used to extract water vapor from a flow and thermodenuders are used to extract volatiles from a flow. The system worked as expected, but large losses occurred via a heated hose that formed part of the sampling system. This article was presented in 2014 at the Fisita World Automotive Congress in Maastricht.

The author designed and set up the particulate measurement systems and hardware, planned and conducted the experiments, post processed the data, and wrote the paper.

Experimental Investigation on the Influence of Boost on Emissions and Combustion in an SGDI-Engine Operated in Stratified Mode

This study was performed in the same metal engine as was used in the first paper. The engine's configuration was unchanged, but this time it was operated with forced induction. The study's objective was to investigate the influence of boost on particulate emissions, and the main finding was that particulate emissions decreased with increasing boost. The paper was presented in 2015 at the 12th International Conference on Engines & Vehicles in Capri, Italy, SAE Naples section.

The author designed and set up the hardware, planned and conducted the experiments, post processed the data and wrote the paper.

Experimental Investigation on the Influence of Boost on Combustion and Particulate Emissions in Optical and Metal SGDI-Engines Operated in Stratified Mode

Given the findings in the previous study, it was decided to explore the effects of boost in stratified mode using an optical engine. The hardware used in this study was identical to that used in the metal engine tests, but with optical access. This allowed the results from the metal engine to be explained visually without the interference of factors such as discrepancies resulting from gas motion. The main findings of this study were that there is a trend towards faster flame development with increasing boost and that the flame area decreased as the boost is increased. The paper was selected as a journal paper and presented in 2016 at the SAE World Congress and Exhibition in Detroit, USA.

The author designed and set up the hardware in co-operation with Gerben Doornbos, planned and conducted the experiments, post processed the data and wrote the paper.

Reduction of Soot Formation in an Optical Single-Cylinder Gasoline Direct-Injected Engine Operated in Stratified Mode using 350 bar Fuel Injection Pressure, Dual Coil- and High Frequency-Ignition Systems

This study was the last one conducted within the project. It examined the influence of DCI and HFI ignition systems, a higher fuel injection pressure (350 bar) than had previously been accessible, and various tumble levels on particulate formation during stratified combustion. The plan was to run these tests in both a metal engine and an optical engine; the metal engine would allow for more exact heat release calculations and measurement of emissions and combustion characteristics in the steady state, while the optical engine experiments would provide visual data to help understand the results of the metal engine experiments. The hardware to be used was state of the art (for the time), and the combination of metal and optical engine tests was expected to provide a good level of scientific depth. Unfortunately, only the optical experiments were completed. The main findings of the study were that the relative abundance of diffusion flames was drastically reduced by raising the fuel injection pressure from 175 to 350 bar. The HFI system provided better combustion stability and utilized a larger area of the combustion chamber than the DCI ignition system. This was explained by suggesting that the HFI system minimizes over-dilution by enabling faster combustion. Another finding was that of the HFI spark plug's 5 electrodes, only 2-3 were used in most of the combustion cycles. The study was performed in an optical engine in co-operation with Volvo Cars Corporation.

This paper was intended to be more extensive – in addition to the planned metal engine experiments, there were also plans to incorporate PIV-measurements, Mie scattering experiments, and LIF imaging. A total of four months were spent on this sub-project, but unfortunately the complications of the PIV system consumed all the available time and so the Mie scattering and LIF studies could not be conducted. Moreover, legal restrictions prevented the use of the high pressure fuel injector in the metal engine tests, which were intended to be performed using the engine rigs at Chalmers University of Technology.

The author worked together with Gerben Doornbos and Stina Hemdal in planning, setting up and conducting the experiments. The author post-processed the data and wrote the paper.

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