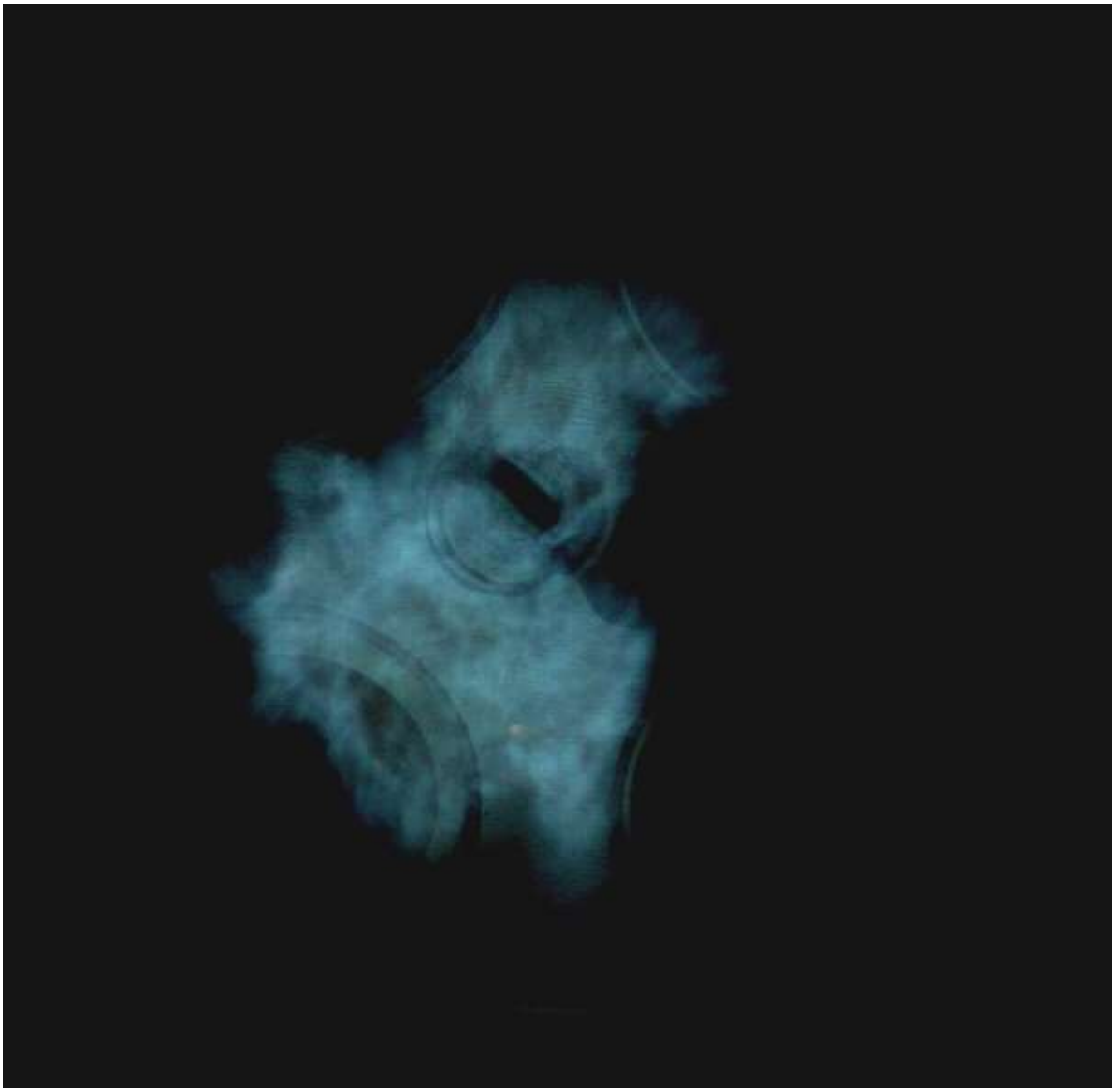

Optical Study of Tumble

Spray and Combustion

Master's Thesis in the Automotive Engineering Master's Program

Robin Ryrholm (Chalmers University of Technology)
Martin Eriksson (Lund University, Faculty of Technology)

Department of Applied Mechanics
CHALMERS UNIVERSITY OF TECHNOLOGY
Göteborg, Sweden 2015

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Picture of early combustion within the optical engine used in the study.

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Abstract

The automobiles of today are getting more fuel efficient in order to meet future emission standards. It is also important to have more fuel efficient engines than the competitors. The consequence is that engine development is intensified and more ideas needs to be tested thoroughly. One of the ideas at Volvo cars is a tumble flap. The tumble flap is controlled to close a predefined part of the intake port in order to increase the tumble motion in the combustion chamber. A tumble motion is a circular motion with it's axis of rotation parallel to the engine head. To make the effects of the tumble flap clearly visible an optical engine is used. An optical engine has a cylinder and piston made of glass which makes it possible to record what happens in the combustion chamber while the engine is running. The fuel is injected directly to the combustion chamber, so called direct injection engine. This gives excellent opportunity to see how the spray is affected by an increased tumble motion. The effect on the combustion could be seen when changing parameters. The events appear in a fraction of a second why a high speed camera is used to capture the events. It has been shown that the spray is mostly effected in a late phase of the intake stroke. The spray is stable and relatively small in the early stage and is therefore only mildly effected by the ongoing gas motion. With the tumble flap activated, it is possible to observe that fuel is moving back to the valves before it is evaporated. Sometimes it is possible to see a tumble motion within the spray. If the injection pressure is increased, the initial spray stability will increase, the fuel reaches further before it is effected of the ongoing gas motion. The combustion takes place considerably faster with the tumble flap restriction. A problem with the tumble flap is that diffusion flame combustion may occur due to an increased risk of wall wetting.

Keywords: Optical engine, Tumble flap, Direct Injection Spark Ignition

Optisk studie av tumble rörelse
Bränslespray och Förbränning
Examensarbete inom Mastersprogrammet Fordonsteknik
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Avdelningen för Tillämpad Mekanik
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Sammanfattning

Dagens bilar blir allt mer bränsleeffektiva och det blir allt viktigare att visa att ens bil drar mindre bränsle än konkurrenterna. Följderna av detta är att motorutvecklingen intensifieras och fler nya idéer ska testas utförligt. En av dessa idéer är en så kallad "tumble-flap". En tumble-flap kan regleras till att stänga av en förutbestämd del av insugsporten för att förstärka tumble-rörelsen, en cirkulär rörelse vertikal mot topplocket, i förbränningsrummet. För att tydligt se vilka effekter som en tumble-flap genererar har bränslesprayen och förbränningen undersökts i en optisk motor. En optisk motor är en motor där cylinder och kolv är i glas, detta ger möjlighet att filma förbränningsrummet medan motorn går. Den studerade motorn är en så kallad direktinsprutad motor. Bränslet sprutas direkt in i förbränningsrummet vilket ger goda möjligheter att se hur bränslets spray påverkas av en ökad tumble-rörelse. Påverkan på förbränningen kan också ses då olika parametrar ändras. Händelseförloppen utspelar sig under bråkdelar av en sekund, därför används en höghastighetskamera för att fånga fenomenen. Det har visats att sprayen påverkas mest i ett sent skede i insugslaget. Sprayen är relativt stabil och relativt smal i ett tidigt skede och påverkas därför marginellt av omgivande gasrörelse. Med aktiverad tumble-flap hinner man observera att bränsle rör sig tillbaka upp mot ventilerna innan det förångas. Det går även i vissa fall se en tydlig tumble-rörelse på sprayen. Ökar man bränsletrycket så förlängs sprayens initiala stabilitet, bränslet når längre innan det påverkas av gasrörelsen. Förbränningen sker markant snabbare om insugsporten har tumble flapen aktiverad. Ett problem med tumble-flap är att mer bränsle förbränns med syreunderskott efter den huvudsakliga förbränningen, så kallad "pool fire". Problemet minskas med ökat bränsletryck. Efter finjusteringar av olika parametrar så som ventilinställningar och bränsletryck bör den markant snabbare förbränningen som en tumble-flap medför innebära ökad effekt med samma mängd bränsle, alltså ökad bränsleeffektivitet.

Nyckelord: Optisk motor, Tumble flap, Direct Injection Spark Ignition

Preface

This Master's thesis studied the effect of a tumble inducing system using an optical engine. The investigations were carried out at Volvo Cars Corporation from March 2015 to May 2015. Before this, from January 2015 a literature study about the subject was carried out in Lund at Lund University. At Lund University Faculty of Engineering one of the Thesis supervisors gave the authors an introduction to Optical Engines.

The experiments were carried out by the authors with the help of the supervisors from Volvo Cars Corporation Roy Ogink and Chalmers Petter Dahlander.

Göteborg Juni 2015
MARTIN ERIKSSON
ROBIN RYRHOLM

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Martin Eriksson Robin Ryrholm
Maj 2015 Göteborg

Nomenclature

BDC	Bottom Dead Centre, the piston is in its lowest position
CA 50%	Crank Angle for the point where 50 % of the energy is released
CAD	Crank Angle Degree, rotational angle of the crankshaft
CI	Compression Ignition engine (diesel)
CoV	Coefficient of Variation
DI	Direct Injection engine, fuel injection inside the cylinder
DISI	Direct Injection, Spark Ignition engine
EGR	Exhaust Gas Recirculation
GDI	Gasoline Direct Injection, engine running on gasoline with fuel injection inside cylinder
ICE	Internal Combustion Engine
IMEP	Indicated Mean Effective Pressure
LDV	Laser Doppler Velocimetry
LIF	Laser Induced Fluorescence
LII	Laser Induced Incandescence
MBT	Maximum Brake Torque
MBT-Timing	Maximum Brake Torque Timing, optimum spark timing
PFI	Port fuel injection
PIV	Particle Image Velocimetry
SI	Spark Ignition engine (ordinary gasoline engines)
SMD	Sauter Mean Diameter
SOI	Start of injection
TDC	Top Dead Centre, the piston is in its uppermost position
TDCF	Top Dead Centre Firing, like TDC but marks that combustion takes place
TKE	Turbulent Kinetic Energy
VVT	Variable Valve Timing
WOT	Wide Open Throttle
UHC	Unburned HydroCarbon, basically fuel that is not used in the main combustion

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1. Introduction

This project investigated the influence of varied port restriction in a direct injection spark ignition., DISI, engine. The motivation, delimitations and problem definitions is presented in this chapter. The responsibility was divided between the students. Martin Eriksson was responsible for the test rig operation and spray analysis meanwhile Robin Ryrholm was responsible for the post processing and combustion analysis. Excluded parts were performed together with equal efforts.

1.1 Background

Volvo Cars is aiming for lower fuel consumption and emissions from their cars. Which is necessary in order to meet future emission legislations. Increasing the level of tumble seems to be beneficial in a DISI engine. It is due to increased levels of turbulence generating faster and more stable combustion. That is why Volvo Cars would like an investigation of the in cylinder flow and the combustion incorporating a tumble flap.

1.2 Purpose

The purpose of the master thesis was to study the effect of a variable tumble system in a DISI engine. The study investigated how the fuel spray is affected by an increased charge air velocity entering the combustion chamber. A tumble inducing mechanism was investigated to see if it gave improved part load characteristics in terms of low fuel consumption. The potential problem of wall wetting was investigated. The study also investigated the effect of tumble on flame propagation in terms of direction and area increase. Direction is important because of knock sensitivity.

1.3 Delimitations

Only one tumble inducing method was investigated, with different input parameters. The combustion chamber structure was predetermined, meaning effects on tumble due to geometrical shape was not studied. The combustion study operating points were homogeneous charge with no stratified charge operating point. For the fuel spray investigation the optical engine was motored. For the combustion study a skip fire scheme was used. This to lower the thermal stress on the cylinder liner and piston. The optical method implemented in the study was high speed video using a xenon lamp for illumination of the combustion chamber.

1.4 Problem definition

How is the fuel spray affected by different levels of tumble? Is wall wetting a problem due to increased intake air velocity? The effect on the spray stability due to increased fuel pressure have to be investigated. The normal flame propagation of Volvo's combustion system have to be investigated. This to be able to see the differences when introducing the tumble flap and changing injection pressure.

2. Theory

2.1 Engine

Internal combustion engines are the most common power supply for propulsion systems. In its different configurations, internal combustion engines are used in areas such as backup generators, ships, any type of land-going vehicles and in some aeroplanes among others [15]. The theory chapter will explain the Otto-Cycle and especially the DISI engine.

2.1.1 General design

A piston engine has a piston moving transversally inside a cylinder. The piston is connected by the piston pin to the connecting rod. The connecting rod is attached to the crankshaft which transforms the transverse motion to rotational motion. Torque is extracted from the crankshaft via a transmission that adapts the torque and rotational speed to the connected application. The piston is moving due to a combustion that takes place when the piston is close to top dead centre, TDC. TDC is the highest piston position. Via the intake port and the intake valve(s) the fresh air enters the engine. Depending on the fuel system in a gasoline engine, fuel is added either in the port (port fuel injection, PFI) or in the cylinder (direct injection, DI). To ignite the mixture a spark plug is used. In a diesel type engine, the mixture auto ignites due to high compression and high temperature. The rest gases from the combustion are let out via an exhaust valve to the exhaust port and then the exhaust system. The valves are controlled by camshafts that open and close the valves in a predefined pattern. To ensure that the valves open and close when the piston is at the same position in every cycle, the camshafts are driven by the crankshaft via a timing belt or a timing chain. Figure 2.1 shows a cutaway drawing of a spark ignition engine. A variant of a DI gasoline engine, is shown in figure 2.2.

An engine can have different number of cylinders with different configurations. These are e.g. inline, like in figure 2.1 and V shaped engine.

2.1.2 Four stroke principle

The four stroke principle consists of four different strokes, intake, compression, expansion and exhaust stroke. The different strokes are shown in figure 2.3 and explained in later paragraphs.

The intake stroke starts when the piston is at TDC. The intake valve(s) are open and the exhaust valve(s) are closed. The piston is moving down and the intake mixture is sucked in to the

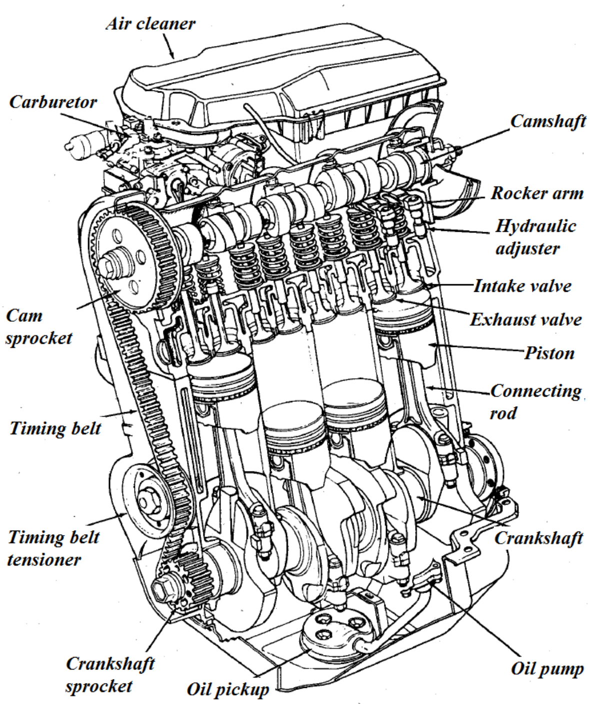


Figure 2.1: Cutaway drawing of a Chrysler 2.2-litre displacement four cylinder spark ignition engine [15].

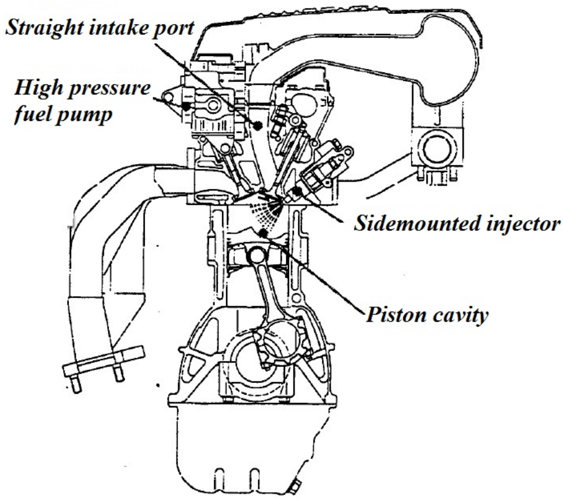


Figure 2.2: Mitsubishi GDI engine with the wall guided type of combustion chamber [26].

cylinder. When the piston has reached bottom dead centre, BDC, the intake stroke ends and the compression stroke begins.

When the piston is at bottom dead centre after the intake stroke the valves closes and the compression stroke starts. The high temperature inside the cylinder will automatically ignite the gas mixture close to TDC firing, TDCF, in a diesel engine. In a gasoline engine a spark from a spark plug is needed to ignite the mixture.

As the piston starts moving down after the compression stroke the expansion stroke begins. All valves are closed. In the expansion stroke the actual work is done. The forces from the combustion push the piston down which spins the crankshaft. The produced torque is often described in relative terms such as indicated mean effective pressure, IMEP. The definition for IMEP can be seen in equation 2.1.

$$IMEP = \frac{P_i * n_R}{V_d * N} \tag{2.1}$$

Where P_i is the indicated power output, n_R is number of revolutions per power stroke, (2 for a 4-stroke engine). V_d is the displaced volume and N is the engine speed [15].

Close to BDC, the exhaust valve(s) opens and while the piston moves up, the exhaust gases are pushed out of the cylinder therefore it is called the exhaust stroke. Since each stroke is half a rotation, each cycle will need two full rotations on the crankshaft to perform. The valve timing of a real engine differs from the theoretical timing described here.

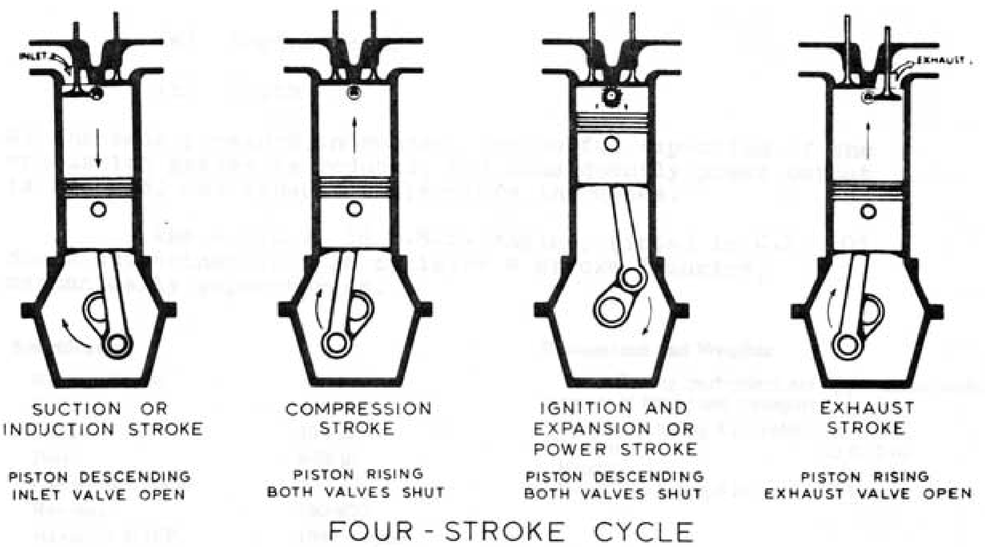


Figure 2.3: The four stroke principle [19].

2.1.3 Direct injection spark ignition

The direct injected spark ignition engine or DISI engine uses the four stroke Otto-cycle. In a DISI the fuel is added to the air mixture inside the cylinder instead of in the intake port, which can be seen in figure 2.2. The DISI engine may have benefits in both fuel consumption and in maximum power if designed properly.

Homogeneous charge

A homogeneous mixture of air and fuel is typically used in spark ignition, SI, engines. It is to get a combustible charge in the whole combustion chamber and the injection is early to give time for mixing. It means the level of mixing should be the same for the whole mixture entering the combustion chamber. For a port fuel injection system, the mixing is achieved partly in the intake port and partly in the combustion chamber before start of combustion. In a DISI engine, the mixing only takes place in the combustion chamber. The fuel is injected early in the intake stroke to get sufficient time for mixing. Since the fuel is added in the cylinder it is possible to get more air through the intake system and thereby increase the volumetric efficiency. Equation 2.2 shows how the volumetric efficiency is calculated. Higher volumetric efficiency means that more air is coming in to the cylinder. It means more fuel can be injected to achieve stoichiometric condition, allowing a higher power output.

$$\eta_V = \frac{\dot{m}_a}{\rho_{IN} * \dot{V}_D} = \frac{\dot{m}_a}{\rho_{a,0} * V_d * \frac{N}{2}} \tag{2.2}$$

Where $\dot{m}_a$ is the air mass flow, $\rho_{a,0}$ is the density of the air at the intake manifold, V_d is the displaced volume and N is the rotational velocity.

Homogeneous charge mode is used at high load in DISI engines to get as much power as possible and also at certain part load conditions. The fuel droplets from the injector are, ideally, fully evaporated before the combustion begin. In a port injection engine, the fuel is evaporated by the heat of the incoming air, intake valve, intake manifold and other relatively hot parts. In a DI engine, the fuel is instead evaporated by the heat from the intake air, cylinder walls and piston. The temperature decrease of the intake air, due to evaporation of the fuel, will be higher in a DI engine than a PFI engine. Cooler incoming air increases the density of the air and the resistance to knock. Lower knock sensitivity, due to lower temperature before combustion, makes it possible to increase the compression ratio in a DI engine. Increasing the compression ratio has a direct positive effect on the overall efficiency of the engine [28, 24].

Stratified charge

The direct injection engine gives the opportunity to run the engine in stratified charge mode. This is mainly beneficial at low load and low engine speeds. For stratified charge fuel is added is relatively late, just a few crank angle degrees, CAD, before the ignition. The combustion will be globally lean with a stoichiometric or rich mixing zone close to the spark plug. The excess of air needed to operate lean implies that the throttle will have to be more open than with a homogeneous charge at the same load. A more open throttle lowers the flow and heat losses, thereby increases the overall efficiency. To use exhaust gases in the combustion also improves the fuel consumption in this case, it is called exhaust gas recirculation or EGR, by the need of less throttling [28].

The fuel injector

The fuel injector in a DISI engine could either be positioned on the side or in the top centre of the combustion chamber. If positioned at the side, the injection charge is said to be wall guided or air guided depending on engine design. Wall guided if the spray needs the piston to get the fuel close to the spark plug and air guided if the air motion within the cylinder is used to direct the fuel towards the spark plug. If the injector is positioned in the top centre it is called spray guided charge, the fuel gets close to the spark plug regardless of the air motions in the cylinder. With wall guided fuel motion, the piston usually has a build-up that redirects the fuel towards the spark plug. Air and spray guided have relatively flat piston crowns.

A common design of a direct injection injector is shown in figure 2.4. The fuel inlet is at the top of the injector. Fuel is provided by a fuel-rail that nowadays usually is common for every injector. Increased volume of the fuel-rail gives less oscillations of the fuel pressure and thereby lowering cycle to cycle variations of the spray. The fuel flows down towards the nozzle. Electronically pulses from the engine control unit activates the coil which lifts the nozzle needle. Fuel will flow out of the opening(s) at the tip of the injector.

There are different ways to actuate the nozzle needle. Old injectors used the fuel pressure to open the needle. When the pressure becomes low, the needle is closed by the push back spring. The method has uncertainties in when injection takes place and how much fuel will be injected, solenoid activation is as mentioned before common today. It is possible to alter the injection timing electronically and it is also a relatively low priced injector. Instead of an electrical coil, a piezo electric stack could be used. Since the mechanical movement of the piezo-crystal is very small it will need many layers of active elements and a motion amplifier to open the needle enough. This option is more expensive but also the most accurate injector in terms of timing and the fuel mass being injected.

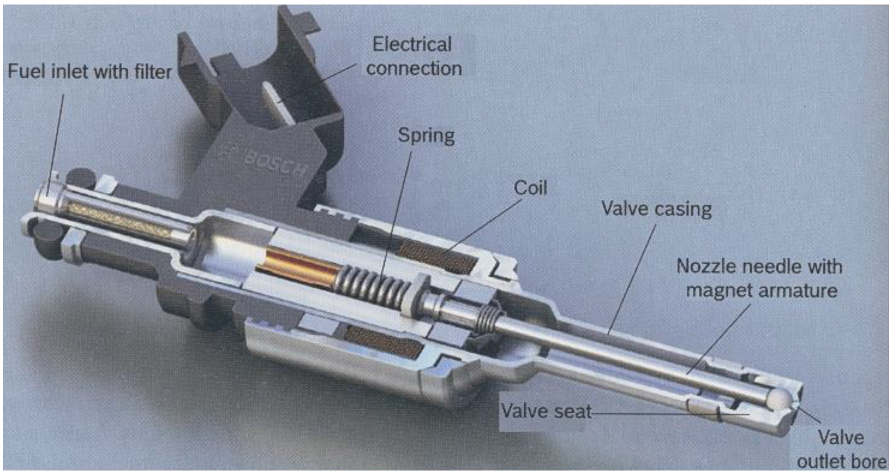


Figure 2.4: Cutaway picture of a electrical coil fuel injector [8].

Air entrainment and spray behaviour theory

The DISI injector requires a relatively high pressure to be able to achieve the rapid atomisation of the fuel droplets that is needed. This because the need for a sufficiently small Sauter mean diameter, SMD, of the fuel droplets. This is to get acceptable levels of unburned hydrocarbons, UHC. It is because smaller sized droplets evaporates easier and therefore less liquid fuel will be present. The evaporation is aided by the piston crown, if it is sufficiently warm. Increased injection pressure will give the flow more momentum leading to longer initial spray penetration.

The increased pressure will also lead to better atomisation of the spray [28]. The phenomenon that affect the early spray breakup are turbulence and cavitation. Afterwards the break up is because of air entrainment. Air entrainment is a phenomena where the air is entrained into the flow which leads to development of vortices in the area between the spray and the surrounding air. This promotes the evaporation of the fuel due to the decrease of fuel concentration. There is a connection between an increased injection pressure and higher entrainment of the air into the fuel spray, see [18]. The theory is based on diesel injector which can be used because of the similarities between a diesel and a gasoline multi hole injector, due to high relatively high injection pressures for DISI.

2.1.4 In-cylinder gasflow

In an internal combustion engine, ICE, there exist different types of gaseous bulk flows that make different conditions for the combustion. The other type of flow is turbulent flow which consists of unorganised small scale flow of fluids. There are three bulk flows that can occur inside the combustion chamber, these are: squish, swirl and tumble. Swirl is the gaseous motion in which the rotation is around the axis of the cylinder. The tumble is a rotational motion around an axis that is perpendicular to the swirl flow.

Variable valve timing, VVT

The camshaft controls when the valves will open and close and how much the valve lifts from its seat. The timing of the valve lift has a significant effect of the engine performance. If the exhaust and intake valves are open at the same time, so called valve overlap, there will be internal EGR at low engine speed. The relatively slow events at low engine speed allows the exhaust gases to flow from the exhaust port and back in to the combustion chamber. This will lead to less power due to less fresh air in the mixture. In order to overcome the higher cylinder pressure due to EGR there is need for more air. By open the throttle more the pumping losses will decrease and more air can enter the combustion chamber. At higher engine speeds, valve overlap is beneficial to some extent. The events within the cylinder are rapid and there is less time to such fresh air through the intake valves. With increased valve overlap the intake valve(s) opens earlier. It will give more time for the intake air to enter the cylinder, increasing the volumetric efficiency and potential power output. At high engine speeds there is a low pressure in the exhaust, leading to increased scavenging and increased suction for fresh air. With variable valve timing, VVT, it is possible to change the timing of the camshaft and affect the engine accordingly [24].

Manifold air pressure

The manifold air pressure is controlled by a throttle that sets how much air/air pressure there will be in the intake manifold. Less air means less fuel equals less power. Without the throttle, the engine would accelerate uncontrolled to self destruction. A throttle is basically a flat disc that opens or closes the intake manifold to regulate the amount of fresh air sucked into the engine. At high engine speeds, the disc is wide open and for low speeds it is almost closed. The air flow at low engine speeds is considerably lower than at high engine speed. Despite this is an unrestricted flow towards the intake valves preferable. The throttle does just the opposite, making it more difficult for the air to pass through. This will lead to higher pumping losses; it is more difficult to suck the air in to the cylinder than with wide open throttle. If the valve timing is set to give internal EGR, the engine will need more air to operate stoichiometric which induce less throttling leading to less pumping losses and increased overall efficiency. With too much internal EGR will the mixture not be ignitable. With higher levels of turbulent kinetic energy, TKE, an engine can operate with more internal EGR because of the improved mixing [24].

High engine speed gas motion

With high engine speeds, a new phenomenon appears a so called ram effect. The irregularity of the intake valves opening and closing combined with the piston movement creates pulses in the intake ports. These pulses increase the air flow through the valves as they are closing by contributing to a strong jet-flow over the valves. This effect can be used to increase the air mass entering the cylinder in the intake stroke by closing the valves later. The downside is a risk for a reverse flow because of the back pressure of a piston moving upwards. VVT decreases the problem since the intake valves can be closed early at low engine speed and be closed late at high engine speeds, thereby gain from the ram effect. The gain is not only from the increased amount of air, increased volumetric efficiency, it is also from the fact that the air jet will induce a tumble motion that will end up in turbulence around TDC [15].

Squish

Squish is the gas movement that occurs when the piston is moving close to the cylinder head and the gas flows towards the middle of the cylinder. Figure 2.5 displays the principle. When the piston leaves TDC the pressure will be lower and the combustion gases will flow back into the squish area. This flow occurs in most engines and the effect is dependent on cylinder head and piston top surface design [16].

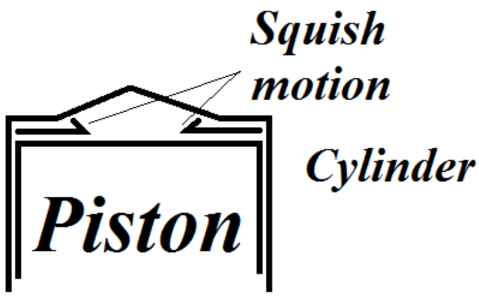


Figure 2.5: Principle sketch of squish motion.

Swirl

Swirl is the gas motion in the cylinder around the axis of the cylinder, see figure 2.6. This airflow is primarily used in CI engines to promote mixing. For a SI engine the flow can be used to get a more stabilised combustion at part load conditions. Swirl can be induced by directing the inlet port or changing the valve seat design. The trouble is that with a port that directs its flow a pressure drop is inevitable which demands additional pumping work during the intake stroke [16, 15].

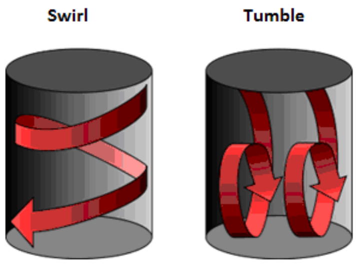


Figure 2.6: Principle sketch of swirl and tumble motion [4].

Tumble

Tumble is the third type of bulk flow that can exist inside a combustion engine. The flow rotates around an axis parallel to the crankshaft, a principle sketch can be seen in figure 2.6. Tumble flow is the most common type in SI engines. The flow will cause less pumping losses in the intake due to the way tumble is created, having straight and relatively small intake ports [16]. This will increase the flow velocity and thereby help the flow separation. The flow will enter the combustion chamber directed over the valve. The trouble with smaller intake ports is the lower mass flow. This means that the volumetric efficiency will decrease according to the definition of volumetric efficiency [16] see equation 2.2. The heightened level of tumble will lead to a higher level of turbulence which induce more combustion noise due to a faster combustion. The advantages of tumble motion includes a more stable combustion due to improved mixing and increased level of turbulence during combustion. The combustion is relatively faster which have the advantage of less time for heat transfer to the cylinder wall. Faster combustion will also mean that less of the expansion is used for combustion i.e. the relative expansion ratio is higher, leading to higher thermodynamic efficiency [16]. Since the combustion will not occur instantly and isochoric but in both compression and expansion stroke, less expansion can be used decreasing thermodynamic efficiency.

When the piston approaches TDCF the vortices becomes smaller and the tumble flow starts to break up. This is because the angular momentum is constantly forcing the vortex to rotate faster, which breaks up the tumble vortex into small scale turbulence. This will give an acceleration of the combustion and lower the heat losses. An optimised cylinder head design converts almost the whole tumble flow into turbulence. After the combustion the turbulence will begin to disappear, because the turbulence breaks up faster by heat than large flows[16]. Tumble allows for leaner mixture due to better mixing for the stratified charge allowing better fuel consumption. Tumble will also promote the mixing for the homogeneous charge.

The amount of present tumble is quantified by a tumble ratio. As mentioned before, tumble lowers the mass flow of air. This means that a high tumble ratio i.e. high level of tumble, will lead to a decrease in the mass flow of the charge air. The definition of tumble ratio is as equation 2.3 [3].

$$R_t = L_d \frac{\int_{\alpha_1}^{\alpha_2} C_f R d\alpha}{(\int_{\alpha_1}^{\alpha_2} C_f d\alpha)^2} \tag{2.3}$$

Where L_d represents a shape factor for the engine.

$$L_d = \frac{B * S}{n * D^2} \tag{2.4}$$

B is the bore of the engine, S the stroke, n the number of valves and D the valve seat diameter[3].

C_f is a flow coefficient.

$$C_f = \frac{4 * V}{D^2 * \pi * V_0 * 3.6 * 10^{-3}} \tag{2.5}$$

Where V equals the volume flow and V_0 a reference velocity[3].

R represents tumble for specific valve lift.

$$R = \frac{8 * (T_{left} + T_{right})}{m * B * V_0} * 3.6 * 10^{-3} \tag{2.6}$$

T_{left} and T_{right} is tumble torques measured in an tumble rig and m is the mass flow[3].

Tumble inducing methods

The least complicated technique of inducing tumble is by having a straight flat and relatively narrow intake port. The trouble with these types of ports is, as mentioned before, a lower mass flow. To compensate for the pump losses a turbocharger can be used. Another way to deal with the challenge is to use a system that closes one part of the inlet port, effectively only letting the charge air flow over one part of the valve. One example of this method is the Volkswagen 1.6 FSI engine [14].

The VW FSI uses a so called tumble flap and tumble plate, with the tumble plate acting as a divider. The tumble flap acts as the blocking mechanism that allows air to only pass through a part of the intake runner. The tumble inducing mechanism is then used in both stratified and homogeneous charge to make the charge less sensitive to external and internal EGR. The decreased sensitivity to EGR is because of the relatively high velocity of the charge air with activated tumble inducing mechanism. The behaviour of with improved stability of the combustion at part load was also seen in [3]. When the need for a high mass flow, especially during full load cases, the flap is open. This is a method for inducing a higher amount of tumble and the Volkswagen tumble flap is a very good example of the tumble flap method.

Turbulence

Since turbulence is important for gas flow in an ICE and especially in a DISI engine it is important to understand what turbulence is. Turbulence is small scale unstructured flow. In the combustion engine there will always be turbulent flows due to the flow velocity, temperature, pressures and geometries inside the engine. Turbulence is always three dimensional and therefore the velocity fluctuations need to be defined in three dimensions. By using steady turbulent flow conditions it is possible to evaluate the instantaneous flow velocity at a specific crank angle defined in equations 2.7 and 2.8 [24, 15]. To make the equations easier to interpret the nomenclature for velocity and turbulence intensity will be defined. U is the instantaneous flow velocity and $\overline{U}$ is the averaged flow velocity over a certain time. $u^{'}$ is the turbulence or a way to show the variation of the flow velocity. The first is the average of the instantaneous velocity equation 2.7 and then follows the definition for turbulent intensity $u^{'}$ equation 2.8. In equation 2.8 u^2 is the deviation of $U(t)$ as equation 2.9 describe.

$$\overline{U}(t) = \lim_{\tau \to \infty} \int_{t_0}^{t_0+\tau} U(t)dt \tag{2.7}$$

$$u^{'} = \lim_{\tau \to \infty} \left(\frac{1}{\tau} \int_{t_0}^{t_0+\tau} u^2(t)dt \right)^{\frac{1}{2}} \tag{2.8}$$

$$U(t) = \overline{U} + u(t) \tag{2.9}$$

Turbulence in an engine can not be explained to satisfaction with steady flow situations. Inside an ICE there will always exist cycle to cycle variations. One way of compensating is by using ensemble averaged velocity (for equations see reference [15]). To be able to capture the cycle to cycle variation of the turbulence with satisfaction, an individual cycle time averaged method can be used. This is an ensemble since the information for each cycle is not sufficient in many cases. The average velocity for an interval $\Delta\Theta$ becomes equation 2.10. Θ is the present crank angle degree where the speed is averaged, N is the number of cycles and i is the present cycle.

$$\overline{U}(\Theta, i) = \frac{1}{N} \sum_{\alpha=\Theta-\frac{\Theta}{2}}^{\alpha=\Theta+\frac{\Theta}{2}} U(\alpha, i) \tag{2.10}$$

For the turbulent intensity it becomes equation 2.11.

$$u^{'}(\Theta, i) = \left[\frac{1}{N} \sum_{\beta=\Theta-\frac{\Theta}{2}}^{\beta=\Theta+\frac{\Theta}{2}} U(\beta, i) - \overline{U}(\beta, i)^2 \right]^{\frac{1}{2}} \tag{2.11}$$

Flame propagation

The flame velocity is divided into two parts, laminar flame velocity and turbulent flame velocity. The definition of laminar flame velocity is the way unburnt gas is transported to the reaction zone inside a laminar flow field. There are different types of methods to explain the laminar flame speed [16]. Turbulent velocity depends on the turbulent intensity and the laminar velocity. With a completely laminar flow the flame propagation would not be fast enough to give enough power output. With turbulence the flame front will be wrinkled and a greater area of flame front which means that more gases can be combusted at the same time hence the flame propagation speeds up. Examples of flame propagations can be seen in the flame propagation pictures in section 4.2.

The flame propagation in a combustion engine can be different. When the mixture is rich, there is a shortage of air which makes the flame propagate with a diffusion flame. This flame will produce a relatively high amount of soot. The flame can also propagate with a premixed flame; here the mixture is stoichiometric or globally lean meaning either perfect air/fuel mixture or excess of air [16].

2.2 Optical engine

Optical engines have optical access to the combustion chamber. It is a method for observing gas flows, flame propagation and different combustion species inside the engine. The visualisation methods can be divided into passive / "non -laser methods" and laser methods.

2.2.1 Chemiluminescence

To explain what types of light different methods use, chemiluminescence has to be explained. In combustion study the chemiluminescence from the flame is used to investigate the flame propagation. Chemiluminescence is electromagnetic radiation emitted due to chemical reactions inside a material. Different species emit chemiluminescent light at different wave lengths. Therefore it is possible to filter out unwanted wavelengths,[7][1].

2.2.2 Mie scattering

For the spray study the spray has to be illuminated to make it visible. This will be done with a xenon light. Therefore the light scattering phenomenon for the illumination will be summarised. The type of scattering is called Mie scattering. This is characterised in a way that the studied particles are larger than the light that is emitted from the light source[2].

2.2.3 Optical access

To be able to study the in cylinder behaviours of an engine inside there is a need for optical access in to the combustion chamber. One way to gain optical access is shown in figure 2.7.

Fibre optic

One optical access method is fibre optics, were the cylinder head is constructed so a fibre optic can be mounted. A problem with the method is the that can be observed is limited. Therefore several optic fibres will be needed to observe the whole combustion chamber.

Quartz windows

This method uses a cylinder liner made from glass. Either it is the whole cylinder liner or small windows [16]. The method with the whole liner gives more line of sight. The shape of the liner introduce distortion of the observed area and needs to be compensated for[10] [24]. To give more access it is possible to use what is called a bowditch piston extender [5] [16]. This method uses a piston extender with a piston made partly in quartz and a mirror giving an optical access from below, figure 2.7 displays the bowditch design.

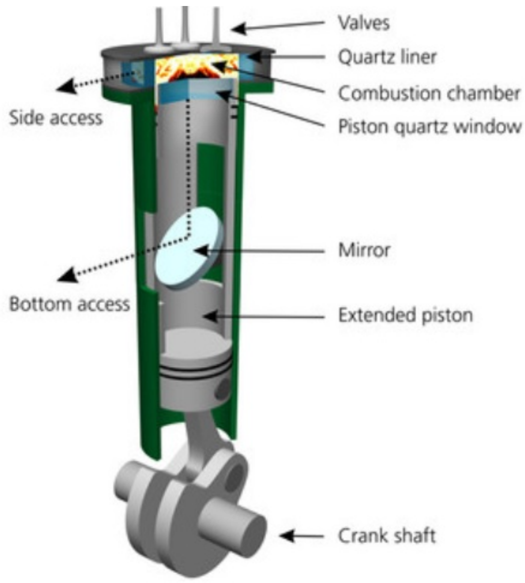


Figure 2.7: Example of a bowditch extender type of optical engine [25].

2.2.4 Passive and ”non-laser” diagnostic methods

The difference between optical methods is the area of usage for each method. It all depends on the scope of the study which method should be used. The different types involves methods for investigating the flame propagation, flow behaviours and combustion species. Some examples are high speed video for combustion or spray and different types of laser methods.

The passive methods are characterised by only using natural radiation from the combustion. This in terms of chemiluminescence from soot formation and other radicals inside the combustion flame [2].

In a motored case it is possible to use e.g. xenon or halogen lamp to illuminate the spray. A high speed camera is used to take the photos from the now illuminated chamber. Albeit not a passive method since the spray is affected by the lamp.

In [13] passive methods were used to evaluate atomised oil droplets in the crankcase of a diesel engine. An example of how it is used in combustion diagnostics is shown in [5]. In this study a bowditch piston extender and a halogen lighting unit were used. Then mirrors were used to get the observations out of the engine, which was then passed through filters and was taken up by a CMOS camera. The trouble with this type of method is that only the light that can be seen with the human eye can be used, other lights can be seen with filters. But it is not possible to investigate reaction zone species [2].

High speed camera

High speed cameras are used in optical engine studies to capture the processes in the combustion chamber. Most optical engines are run at relatively low engine speeds due to durability issues of the optical parts. To fully understand what happens inside the cylinder, high frame rate is necessary. Typically it is sufficient to record every crank angle. When background light is used, it is of great importance that the risk of reflections are taken into consideration. Filters can be used to amplify the signal of certain combustion species. For instance filters can be used to see the formation of soot.

CCD vs CMOS

CCD and CMOS are two types of image sensors used in digital cameras. Both types have advantages and disadvantages. The CCD sensor is more sensitive to light than the CMOS sensor. A CCD sensor is more sensitive to darkness than the human eye. A benefit regarding high speed cameras for the CMOS sensor is the possible frame rate, which is significantly higher than for CCD. The CMOS can be made smaller and it consumes less power than the CCD which makes the CMOS ideal for small applications such as phones and toys [27].

Infrared camera

Infrared cameras can be beneficial for studying the behaviour of fuel vapours and the air/fuel mixing process. In a study, a diesel engine was used to investigate infrared techniques[23]. Diesel fuel produces infrared energy emissions that an infrared camera can capture. According to [21] there are infrared techniques that can be applied for gasoline fuel as well. The fuel plume can still be seen with both ordinary and infrared cameras. Another benefit of using infrared technology is the possibility to measure the temperature. Disturbances from soot particles can give high infrared emissions which makes the recorded temperature a bit higher than the actual. [23].

2.2.5 Active and laser methods

For these methods lasers are used to excite either added particles or combustion species particles. To record the pictures CCD cameras are used.

LII

Laser induced incandescence, LII, is a method where a laser heats up soot particles. The increased temperature will shift the spectral colour of the spray to blue. Therefore there will be a difference in light emitted from the soot particles compared to the surroundings. The method is used for investigation of soot formation, concentration and particle sizes [2].

PIV

Particle image velocimetry, PIV, is a method where flow tracer particles are introduced, which are made visible with thin laser sheets. The pictures are used to determine the flow structure inside the combustion chamber. [12, 16].

LDV

Laser doppler velocimetry, LDV, is a method for local velocity measurements where two laser beams are crossed which give an intersection point. When a particle passes the point light will scatter and therefore the light intensity is going to oscillate. The oscillation makes it possible to calculate the velocity of the passing particle and to detect the oscillations a photomultiplier is used.[16].

LIF

Laser induced fluorescence, LIF, is a method to visualise certain components in the combustion chamber. The method uses a laser to excite electrons in molecules; when the electrons go back to their original state, light with a specific wavelength is emitted. The wavelength of the laser needed is different depending on the species tested. Therefore, only substances where the spectral properties are known can be studied. This method is used to study combustion species but also added substances. LIF can be used when analysing e.g. fuel concentration.. If the light from the species is especially temperature dependent, LIF can be used for temperature measurements. To give an introduction to what can be studied with this method a few variants of LIF will be explained. [20].

The method allows the study of different combustion species, as mentioned before. The types of species give different information about the combustion, which means, depending on the combustion species, certain filters and laser wavelength will be needed. Therefore there follows an explanation of how to do LIF on some combustion species. Planar LIF, PLIF, is a method where the laser creates a plane from where measurements can be taken [20].

OH-LIF

This method studies the OH radical, a specie that does not suffer from interference from other combustion species, making it a relatively easy species to detect according to [22]. The signal strength of the OH radical is also relatively high, giving advantageous properties for detection of the radical. The OH radical is present in high temperature areas, making it suitable for flame front detection and therefore flame structures [9]. The excitation of the OH radical requires a laser wavelength of 284 nm [22, 9].

CH-LIF

This species is found in low concentration inside the combustion chamber, present in the richer parts of the fuel air mix and for this reason it can be used to detect flame front [9]. The excitation that is needed for the CH to be detected requires a laser with wavelength of 378 nm or 431nm [22].

CO-LIF

CO is of interest because it is an important pollutant in combustion that exist in rich mixtures. To be able to detect the species there is a need for a laser with the wavelength of 230 nm and detection at 400-600 nm according to [22].

Fuel tracer LIF

Planar LIF is used to measure the fuel distribution inside the combustion chamber. In commercial fuels there exist species that can be used for fluorescence measurements. It is possible to add to add substances to the fuel to act as fuel tracers. The trouble is that different types of fuel tracer species will react differently and therefore affect the distribution measurements [22].

2.3 Litterture studies on tumble motion

Articles summarised in this chapter were found to be of special interest with subjects close to the purpose of this project.

2.3.1 SAE 2002-01-1645 Influence of an Adjustable Tumble-System on In-Cylinder Air Motion and Stratification in a Gasoline Direct Injection Engine

In [11] a study is made on the influence of variable tumble system on an air-guided DISI engine. The study was made on a single cylinder engine from a BMW motorcycle. The method that is used to create the tumble increase is a tumble flap that can be opened and closed depending on the load case. When the flap is activated one third of the intake port is used for air flow. The study investigated the effects of the tumble flap and also the possibility to redirect the flow towards a centrally mounted spark plug. The intake system has the runners separate to each other. The tumble flaps are adjustable in twisting, twisting away from each other. The tested angles were 0°, 30°, 60°, 90°and 135°of twist from horizontal start position. The tumble flap is tested in half open, closed or fully open. PIV measurements are used to study the bulk flow and the turbulence. The conclusions that where made from this study were that a twist angle of 60°decreased the cycle to cycle variations compared to the other angles. The 60°setting gave a more concentrated flow that gave a more homogeneous charge than for other settings, hence less cycle to cycle variations. It was also concluded that the flap should be completely closed.

2.3.2 SAE 2001-01-1306 - Tumble Generator Valve (TGV) control of in-cylinder bulk flow and its turbulence near spark plug in SI Engine

In [17] a valve is used to direct the flow in the intake port. When the valve is closes the lower part of the intake port and tumble is induced because of the higher velocity and different direction of the gas flow. The valve is like a flat wall perpendicular to the oncoming air, when closing the lower part of the intake port. When the valve is open, it is positioned in line with oncoming flow. The experiments were performed in a Subaru EJ25 engine. Less unburnt hydrocarbons were produced during cold start when tumble was induced, (closed TGV). With PIV measurements, it was shown that the increase of turbulence around the spark plug was not significant but the turbulence scales became smaller with 66% [17]. Smaller scale of turbulence means improved break up of the bulk flow.

2.3.3 SAE 2014-01-2885 The Investigation and Application of Variable Tumble Intake System on a GDI Engine

In [6] the air flow in a GDI engine was analysed. Four different cases were studied with computational fluid dynamics, CFD. The most interesting case is “case 1” where a tumble flap is used to close the lower part of the intake port. The tumble flap is positioned slightly above the centreline of the intake port. The intake port is designed for tumble and the piston uses a spherical piston crown. The second case, “case 2”, has the same intake port and piston but without the tumble flap. “Case 3” has a different intake port, the same piston design and no tumble flap. The fourth case, “case 4”, has the same intake port as case three without a tumble flap. Case 4 has a dent piston crown instead of the spherical piston crown used in the other cases. The simulations showed that case 1 has the highest velocity of the gases along the combustion walls. Tumble is induced in all cases but the maximum tumble ratio is about four times higher in case 1 than the others. The normal flow around the intake valve is that gases flow on both up and down-side of the valve. Thus, the flow at the lower side will disrupt or weaken the tumble motion started by the upside flow. In case 1, however, the flow from the intake port on the downside of the valve is almost zero because of the tumble flap. This enables some gas flow from the tumble motion in the combustion chamber to flow above the intake valve when it is open and hence not weaken the tumble motion. Also in case 1, the discharge coefficient of the valve is significantly lower than the other cases. The tumble ratio is increased for all valve lifts with case 1 [6].

2.3.4 SAE 2011-24-0054 The Effect of Tumble Flow on Efficiency for a Direct Injected Turbocharged Downsized Gasoline Engine

In [3] the effect of tumble inducing intakes was studied. The study was performed with a divider and partly closed intake ports. This study was conducted to see if tumble would affect mixing during part load and how much it could reduce the combustion duration. It would allow for improved combustion phasing since the level of tumble and therefore turbulence is high close to TDC. To get the information about the combustion process and how it was related to CFD simulations, PIV measurements were done. The conclusion was that the increased tumble due to the divider led to reduced knock sensitivity. Also internal EGR sensitivity was reduced at mid load which makes it possible to throttle more hence lower pumping losses. One goal was to see if it was possible to create too much tumble, leading to a reduction of efficiency. This behaviour did not occur. The shape of the tumble flow seemed to be of importance. The study focused on valve timing to optimise the flow.

3. Method

3.1 Experiment

To study the effect of increased tumble on a DISI engine an optical engine was used. The engine had an intake port with a variable tumble flap restriction. The influence of the tumble flap was studied by varying the operating parameters. These parameters were, levels of restriction, engine speed, manifold air pressure and valve timing and load point. An increased engine speed will increase intake air velocity giving more tumble bulk motion, resulting in possible improvements of the fuel air mixing. The valves timing will change the valve overlap, and therefore the Internal EGR level.

Fuel injection pressure was also varied. According to theory, a high fuel pressure will give higher momentum of the spray. Higher momentum should improve spray cone stability against the intake air. The spray may be dissipated more efficiently as mentioned in the "Air entrainment and spray behaviour" theory section and may lose their momentum faster. To better understand the injector and the fuel spray a study was made in a spray bomb. The injection pressure was changed to see differences in penetration and spray behaviour. The differences in the spray bomb compared to . At higher fuel pressure the injection time was will be shorter which had to be taken into consideration for the spray analysis.

3.2 Engine configuration

Important engine parameters for the optical engine can be seen in table 3.1. As table 3.1 shows the tumble flap had three different settings. These three settings were tested for reference operating

Table 3.1: The optical engines size and attached equipment

Displacement	492 cm^2
Bore	82 mm
Stroke	92.3 mm
Compression ratio	10.8:1
Fuel	Iso-octane
Injector	6 hole multi hole injector
Fuel pressure	100-350 bar
Tumble flap restriction	Non, Low, Medium, High

points to isolate the effect due to level of restriction. For other operating points only the tumble flap with high level of restriction was tested.

3.3 Operating points

The operating points were taken from earlier studies at Volvo Car Corporation. The mini map points were also motivated from the knowledge of the optical engine limits. The optical engine had an engine speed limit of 2400 rpm and an IMEP limit of 8 bar. Therefore an operating point at 1500 rpm and 3.62 bar IMEP was used for the combustion study. For the spray study an operating point at 1750 rpm was used as a reference point. The spray study reference point and the combustion study reference point were not the same due to that the spray had evaporated before any effect was seen on the spray, see the spray results. The method for illuminating the spray required another operating point where the injection was longer. According to theory, the increase of tumble should promote the mixing of the fuel. Therefore a stratified charge operating point was supposed to be studied.

A wide open throttle WOT full load point was also tested in the spray study, This was an operating point where a clear tumble motion was visible without the tumble flap which made it interesting to investigate how a tumble flap would affect a full load point. A so called "cat-heating" point was also. This was an operating point with two injections. The second fuel injection may cause diffusion flames due to possible piston and valve wetting. Therefore it was interesting to understand how the tumble flap would affect the spray and combustion. The operating point is called cat heating since the dual injection and late ignition is used to increase the temperature of the catalytic converter.

3.4 Optical access

The optical access was a Bowditch design piston extender see section 2.2.3. With a glass piston and 45 degree mirror it was possible to observe the flame propagation from underneath. The piston was made from quartz glass and cylinder liner was also made from quartz glass: The spray was studied form the side view and the flame propagation from underneath. The fuel air interaction is most visible in the plane where tumble occur. To observe the fuel spray a Karl Storz Xenon Nova 300 lamp with a 300 W xenon light bulb was used. The lamp was mounted inside the piston extender to illuminate the centre of the combustion chamber with little or no reflection.

3.5 Camera

The used camera was a Phantom v7.3 high speed video camera with a SR-CMOS photo sensor. The camera was used with colour settings to make it possible to observe the difference between diffusion and premixed flame. To extract the data from the pictures the cameras picture handling program was used. In the program brightness, gain and gamma settings were changed to enhance the faint premixed flame. The objective on the camera was a Carl-Zeiss 85 mm f 1.4 objective with a plus one macro lens. The macro lens was used to get a better close-up of the combustion chamber. To conserve the light from the combustion, the aperture was adjusted its most open setting. For the spray filming, the aperture was adjusted to a more closed setting to lower the reflections

For the spray study the picture focus was located on front edge of the rearmost intake valve seen from the camera. This since there are no reference points closer to the tumble plane than the valves. For the combustion the picture focus was placed at the base of the spark plug electrode to ensure focus on the early flame.

3.6 Experimental set-up

The test facility offered flexibility when positioning the different equipment needed for the experiments. A beam structure was used to mount the camera, illumination and other equipment for the experiments. The engine was placed on a vibration absorbing table. The beam structure was bolted to the floor. Therefore engine vibrations did not affect the camera. An overview of the test rig is presented in figure 3.1. Apart from some temperatures, all measured parameters were logged in a logging computer.

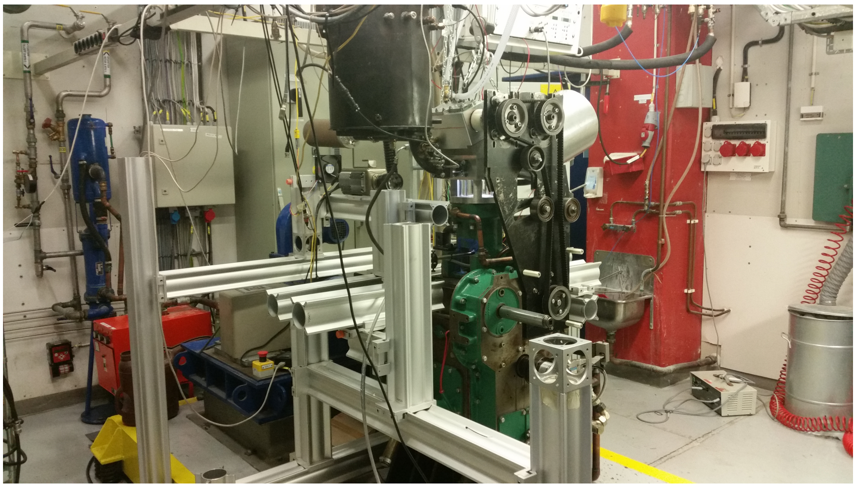


Figure 3.1: Overview picture of the test rig

3.6.1 Side view

To see the effect of increased tumble on the fuel spray, the camera was positioned at the side of the engine. From the camera perspective, intake valves to the left and exhaust valves to the right. A beam was positioned horizontally with the camera directed to the side of the engine, see figure 3.2. A fan was placed behind the camera to provide cooling to the camera. When filming the spray a xenon light was placed beneath the piston to spread the light evenly in the combustion chamber. Experiments with two light sources were performed, however it proved difficult to get less reflections than with one lamp.

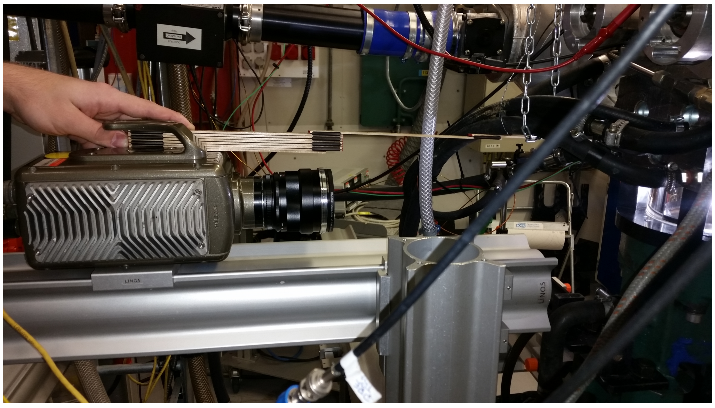


Figure 3.2: The pictures shows the camera position in side view mode.

3.6.2 Underneath view

For the combustion study, a 45 degree mirror was used to reflect the light from the combustion chamber onto the camera lens. The camera was positioned on the exhaust side of the engine due to the layout of the test rig. The mirror was mounted on the opposite side. When filming the combustion no illumination was used due to the faint flame. See figure 3.3 for details of the positioning.

Figure 3.3: The pictures shows the distance between the camera and the engine in under view mode.

3.6.3 Camshaft phase set-up

Volvo Cars Corporation has implemented VVT for several years. The optical engine had no VVT which limited the cam phasing adjustability. The camshafts could be split between the head and the mounting for the cam sprockets. The connecting transmission was an internal cogwheel that transferred the torque to the camshaft. The partition allowed the possibility to change the cam phasing. It was however only with relatively large steps that (around 18 degrees for each cog) the phasing could be changed. A more time consuming way was to loosen the cam belt and turn the engine without moving the cam sprockets. It was concluded that the differences between the "right" cam settings and the ones achieved should not affect the investigation significant. The largest errors would be apparent during combustion study where it was important to have the right amount of internal EGR for normal operation. The reference position was for an operating point with WOT and 2000 rpm.

3.6.4 Measured parameters

Several engine parameters were measured, some were logged in a logging computer and some were just instantaneously visible at displays. In the engine room there were four temperatures that were noted: intake, exhaust, coolant and oil temperature. In the control room the lambda value and the air mass flow were displayed on separate instruments. The lambda sensor was mounted on the exhaust manifold and the mass air flow meter was a hot wire sensor mounted upstream of the throttle. From the logging computer, pressure trace, cylinder pressure, IMEP, CA 50 % and coefficient of variation, CoV, of IMEP and crank angle degree 50% burnt, CA50%, could be obtained. On the logging computer pressure trace, ignition, injection and camera trigger signals were displayed. The signals showed start and duration of ignition, injection and camera trigger position. These values were used to ensure ignition and injection occurred at the correct CAD. These signals were set by a trigger control unit which sends the signals to the camera and engine equipment.

3.6.5 Spray-bomb

To study the effect of increased injection pressures, an investigation was made in a spray-bomb. The same injector was used in the spray and combustion study. A spray bomb is a vessel where it is possible to mount an injector and study the fuel spray. Compared to an engine there exist no back-pressure meaning that the results from the spray bomb is not directly applicable to the spray study.

3.7 Test plan

There are a number of operating points that will be studied to understand the effect of an increased TKE. These points must be representative to their intended application. These operating points are taken from earlier studies on a similar intake port system made by Volvo Car Corporation. The mini map points also represent common operating points within Volvo Car Corporation. A mini map is a set of operating point that will be tested in a study. These operating points are evaluated to see if they could be studied in the optical engine within its operational limit. Important operating points for a real engine was used to be able to make a conclusive and time efficient study.

3.7.1 Spray study

For the spray study, no combustion, it was possible to run most points under 2400 rpm which is the limit of the engine. The ignition was turned off during the spray study to avoid combustion and pressure rise due to combustion, allowing WOT points.

1500 rpm, 6 bar IMEP

This operating point was a reference point that had been used in earlier studies at Volvo Car Corporation. It is a commonly used operating point. It was used as the reference point in the study of the spray interaction. Early testing established that the operating point was not suitable for the spray study. The interaction between spray and the intake air could not be investigated which can be seen in section 4.1.2.

1750 rpm, 9 bar IMEP

This is an operating point that is representative for normal use at highway speeds. This operating point had a longer and later fuel injection compared to 1500 rpm, 6 bar IMEP. These settings were more suited to the spray study than compared to 1500 rpm 6 bar point. Therefore was this operating point chosen to test every level of restriction and fuel pressure.

2000 rpm, WOT, 1 bar relative air intake pressure

This point was representative to a full load operating point that had a relatively high intake air velocity, compared to 1500 rpm. Operating at full load condition, it is not usual to increase the intake air velocity and consequentially the TKE with a tumble flap. These WOT operating points usually have enough tumble motion to promote the mixing and turbulence giving stability of the combustion. However a tumble flap may improve the stability of the combustion even further. Therefore it could improve robustness against knocking for an operating point like this where knock is a limiting factor.

700 rpm, stratified start

This is a transient operating point during start up of the engine and it has a late injection, giving a stratified charge. This has the problem that the mixing of the fuel and air is poor, for this reason there will be relatively rich zones. This may cause particle emissions and unburned hydro carbons UHC. With the improved tumble movement for this type of operating point the mixing would be promoted, improving stability and flame propagation and allow for leaner mixtures to be used meaning less fuel needed for a comparable operating point. The spray interaction was checked for this operating point to see if increased tumble would affect the late injection.

1450 rpm, cat-heating

This was a special operating point where the fuel injection is divided into one early and one late injection. The method is used to heat the catalytic converter to its operating temperature. Therefore it was concluded to use this point in the spray study. It was an effort to see how these fuel sprays interacted with the intake airflow when changing the tumble flap.

3.7.2 Combustion study

The combustion study was supposed to be made for several operating points. The focus was to study the effect of tumble on the flame propagation. Since it is expected that the combustion would occur faster and give a higher load, the 1750 rpm 9 bar IMEP point was not tested. 9 bar IMEP is on the limit of safe operation for this optical engine. The operating points were taken from old investigations with a full metal single cylinder gasoline engine.

1500 rpm, 3.62 bar IMEP

This operating point was chosen since it had a low load for the optical engine and it was a point with low cycle to cycle variations. This was a homogeneous charge operating point. Therefore this operating point was used for the flame propagation study of the project. Also at this point the maximum brake torque, MBT, was investigated for different levels of restriction. The spark timing was changed in order to find the MBT-timing. This was because an increased level of tumble should increase the flame velocity. This may cause a combustion that is so fast that the CA50% would occur before TDC instead of after. CA50% should be after TDC to allow most of the energy to be released in expansion stroke. The flame propagation was studied in order to see how the different levels of turbulence near TDC affected the flame propagation.

700 rpm, stratified start

This operating point was investigated to see if the increased tumble would promote the mixing and make the combustion burn with less cycle to cycle variations. It would also be used to see if the combustion would have less variation for stratified charge conditions. Unpublished CFD calculations and theory states that the turbulent kinetic energy started to drop at CAD close to TDCF. Therefore it was interesting to see if the increased turbulence had an effect on the combustion in this operating point which was very late.

3.8 Post processing programs

After the pictures had been adjusted with regard to brightness and gamma which is the sensitivity to difference in light and dark areas, Matlab was used to further analyse the images. Algorithms were made to improve the understanding of observed phenomenon and quantify the results in terms of flame propagation. From the Phanton Camera program, images for each time step during the film sequence were saved in .tif format.

3.8.1 Information from XML-files

The camera program produces a xml-file containing pictures data, for instance when the pictures were taken. In the latest versions of Matlab there was a function called ”xml2struct” that converted the xml file to a code structure using the Matlab command ”xmlread”. By comparing the times for the different images, the number of cycles in the sequence was calculated. Cycles that are not complete (beginning or the end of the sequence of pictures) were not used in further analysis. The program returned the number of images in each cycle which could differ with one image from cycle to cycle depending on initial camera timing.

3.8.2 Cycle averaging

The algorithms performed averaged pictures of a certain CAD for a certain number of cycles. The mean value is calculated for the 50 last cycles in each test point. With the command "imread" in Matlab, the pictures are converted from images to matrices, spanning 608x600x3. The matrix stands for the number of pixels and the values for the three colours (red, green and blue). The colours were divided into separate matrices for easier handling and user friendliness. The program loops through every picture in every complete cycle in the test sequence.

3.8.3 Combustion analysis

The algorithms checked in which direction the combustion started and how the flame propagated. It also calculated the amount of pixels that had combustion in each image section. The image was read in the same way as used in the averaging algorithm. The image is categorised into eight wedge shaped sections. The corner of the pictures are not of interest since the combustion chamber is circular rather than square. Therefore a circle with 300 pixels radius is used as outer limit for the analysis. The spark might not be directed in the same direction as the combustion will propagate. A circular inner limit is created so that the spark was not mistaken for a propagating flame. A circle with a radius of 60 pixels was placed around the spark plug. The image was converted to greyscale which made it easier to determine the flame front by just using one intensity measure instead of three.

Combustion in a pixel was defined as an intensity value above around 12000 RGB-value (16 bits of resolution corresponds to a maximum intensity value (RGB value) of 65536). Due to cycle to cycle variation and different input parameters, the limit for combustion was changed until it suited the cycle. Since the light from the combustion was relatively weak in the very beginning it was not easy to determine in which direction the very early flame propagated. Therefore a limit of 100 pixels was introduced before combustion was indicated.

Another part of the algorithm checked the maximum distance from the combustion chamber centre to the edge of the flame. Since only one pixel in each direction is considered the edge the method is very sensitive to the threshold value. This part of the algorithm was meant to be used for flame speed evaluation. However it turned out to be a very sensitive method that did not show the expected results. Therefore the flame area speed was calculated instead. The algorithm calculates the flame area by finding the edge of the flame. The area is compared between to adjacent pictures to calculate the flame area change. The time difference between the pictures is used to get the flame area speed. The principle can be seen in figure 3.4. A1 represents the flame areain the previous picture and A2 in the present picture, The flame area was expressed in SI units as square meter and the flame area speed as square meter per second.

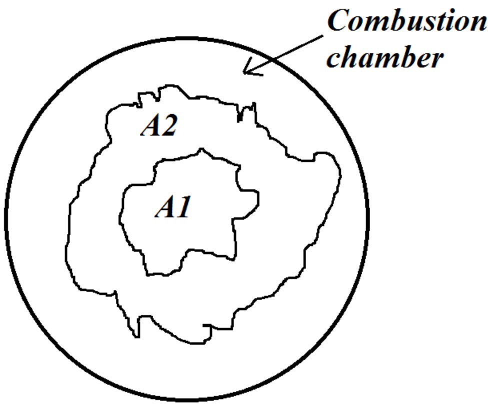


Figure 3.4: A1 is the flame area of the combustion at the first picture, A2 is the flame area of the combustion in the second picture. The time difference between the two pictures is used to get the flame area speed.

4. Results

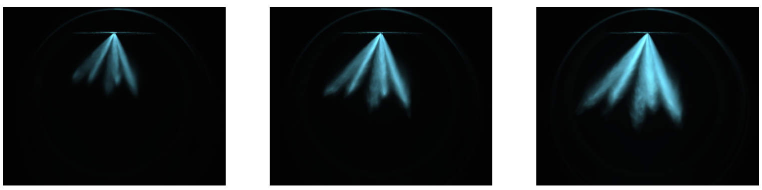
The result chapter will be divided into two parts. The first part will explain what was seen in the spray study. The next part will show the results form the combustion study.

4.1 Spray study

This part of the results shows what was seen in the study of the fuel spray interaction with the intake air. First the results of the spray bomb study will be presented since this was used to understand the changes in spray behaviour due to increased fuel pressure. It was so the effect was not mistaken for tumble induced fuel/air interaction. In the spray pictures from the engine, the intake valve side is on the left side of the pictures. The cylinder liner is placed in the right and left edge of the pictures.

4.1.1 Spray-bomb

The behaviour at 52 bar injection pressure 1.2 ms after start of injection, SOI, can be seen in figure 4.1(a). Comparing this picture with 100 bar pressure at the same time after SOI in figure 4.1(b) and 168 bar in figure 4.1(c) the following could be seen. The differences between low and high fuel pressures were an increased initial spray penetration, at high fuel pressure. At higher pressures the total fuel mass will be injected faster. With higher pressure it was possible to observe larger vortices in between air and fuel. Less distinctive individual sprays suggest there also was a finer break up of the spray. These phenomenons cant be confirmed with the used methods.



(a) Instant picture of spray with 52 bar injection pressure 1.2 ms after SOI

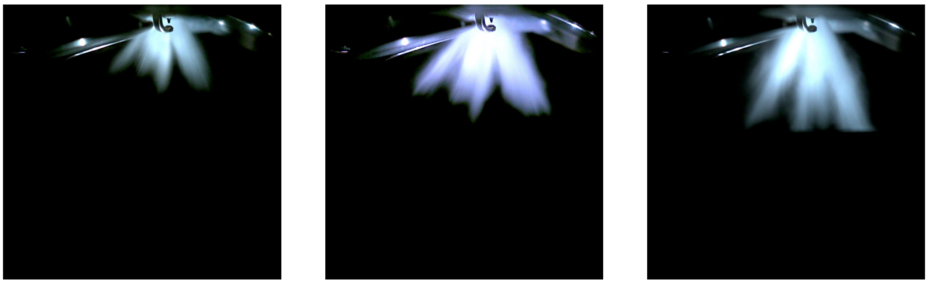
(b) Instantaneous picture of spray with 100 bar injection pressure 1.2 ms after SOI

(c) Instantaneous picture of spray with 168 bar injection pressure 1.2 ms after SOI

Figure 4.1: Instantaneous picture of injections from the injector used in the study at different injection pressures.

4.1.2 1500 rpm, 6 bar IMEP

The effect of the increased fuel pressure acted in the same way as for the spray-bomb. With 350 bar injection pressure the individual sprays were less distinctive and had an increased initial spray penetration. At 200 bar and 100 bar injection pressure the initial penetration were less and the individual spray plumes were more distinctive. This can be seen in figure 4.2(a), 4.2(b) and 4.2(c) which display the spray at different injection pressures in the optical engine.



(a) Instantaneous picture of spray with 100 bar injection pressure 310.5 CAD before TDCF.

(b) Instantaneous picture of spray with 200 bar injection pressure 310.5 CAD before TDCF.

(c) Instantaneous picture of spray with 350 bar injection pressure 310.5 CAD before TDCF.

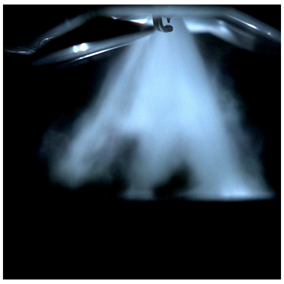
Figure 4.2: Instantaneous pictures of the fuel spray at 1500 rpm with SOI 315 CAD before TDCF.

With an unrestricted port, there was no visible tumble motion for the spray. For the medium level of restriction there was no visible tumble flow before the spray had evaporated. This was also the case for the high level of restriction. The fuel pressure had no effect on the spray cone stability for this operating point with either medium or high restriction. The difference in the spray bomb experiments and the optical engine is the back pressure that does not exist in the spray bomb.

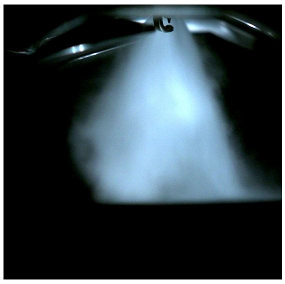
4.1.3 1750 rpm, 9 bar IMEP

For this operating point with the unrestricted intake port it was still not possible to observe tumble bulk movement. The increased fuel pressure gave longer initial penetration and an increased risk of piston wetting. The 350 bar injection pressure reached the piston faster compared to 200 bar. The same behavioural differences were seen between 200 and 100 bar. The comparison can be seen in figure 4.3 where the behaviour for different pressures are displayed at different CAD.

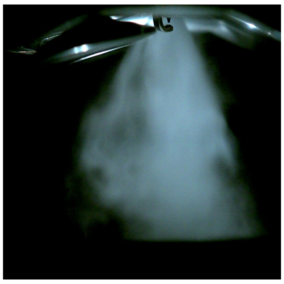
(a) Instantaneous picture of spray with 100 bar injection pressure 287.3 CAD before TDCF.



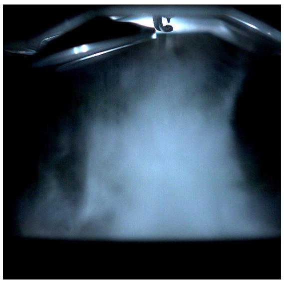
(b) Instantaneous picture of spray with 200 bar injection pressure 287.3 CAD before TDCF.



(c) Instantaneous picture of spray with 350 bar injection pressure 287.3 CAD before TDCF.

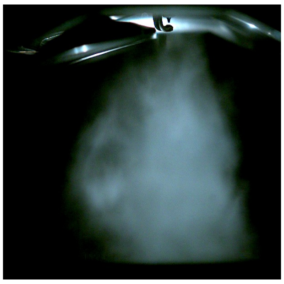


(d) Instantaneous picture of spray with 100 bar injection pressure 276.1 CAD before TDCF.



(e) Instantaneous picture of spray with 200 bar injection pressure 276.1 CAD before TDCF.

(f) Instantaneous picture of spray with 350 bar injection pressure 276.1 CAD before TDCF.



(g) Instantaneous picture of spray with 100 bar injection pressure 269.7 CAD before TDCF.

(h) Instantaneous picture of spray with 200 bar injection pressure 269.7 CAD before TDCF.

(i) Instantaneous picture of spray with 350 bar injection pressure 269.7 CAD before TDCF.

(j) Instantaneous picture of spray with 100 bar injection pressure 266.5 CAD before TDCF.

(k) Instantaneous picture of spray with 200 bar injection pressure 266.5 CAD before TDCF.

(l) Instantaneous picture of spray with 350 bar injection pressure 266.5 CAD before TDCF.

Figure 4.3: Instantaneous pictures of the fuel spray at 1750 rpm with SOI 300 CAD before TDCF with deactivated tumble flap.

When the intake port had low level of restriction it was possible to detect a tumble motion for 100 bar injection pressure. For 200 bar and 350 bar injection pressure this could not be observed. The effect of the increased air velocity on the spray was increased during the injection. It was most visible late in the injection and caused wall wetting on the intake side of the cylinder liner. The behaviours of the spray for different pressures at low level of restriction can be seen in figure 4.4.

Medium level of restriction gave no visible differences compared to low level of restriction. With 350 bar injection pressure, the spray cone was relatively robust against the intake air. The only visible effect was seen when the spray had been slowed down by the in cylinder conditions. The tumble motion was centred beneath the intake valves in the combustion chamber and this moved further towards the intake side late in the intake stroke. This can be seen for every level of restriction.

For the high level of restriction of the intake port it was possible to observe a stronger tumble motion. The effect of the tumble motion became visible for 350 bar injection pressure, although the spray cone still retained its stability. The affect was only visible after the end of injection, EOI. These behaviours of the spray can be seen in figure 4.5, where the different pressures are displayed against each other at four different CAD.

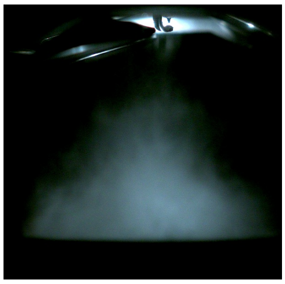
(a) Instantaneous picture of spray with 100 bar injection pressure 287.3 CAD before TDCF.

(b) Instantaneous picture of spray with 200 bar injection pressure 287.3 CAD before TDCF.

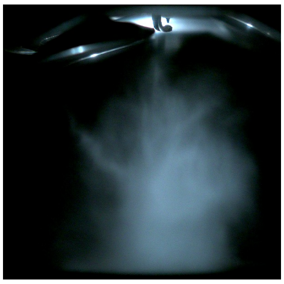
(c) Instantaneous picture of spray with 350 bar injection pressure 287.3 CAD before TDCF.

(d) Instantaneous picture of spray with 100 bar injection pressure 276.1 CAD before TDCF.

(e) Instantaneous picture of spray with 200 bar injection pressure 276.1 CAD before TDCF.

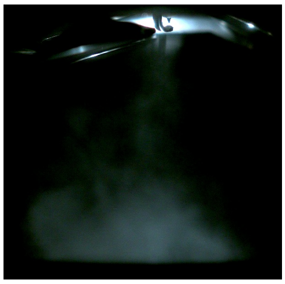


(f) Instantaneous picture of spray with 350 bar injection pressure 276.1 CAD before TDCF.



(g) Instantaneous picture of spray with 100 bar injection pressure 269.7 CAD before TDCF.

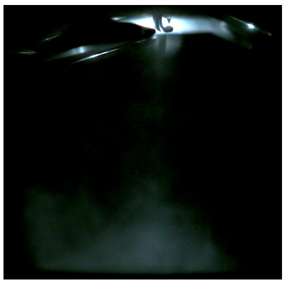
(h) Instantaneous picture of spray with 200 bar injection pressure 269.7 CAD before TDCF.



(i) Instantaneous picture of spray with 350 bar injection pressure 269.7 CAD before TDCF.

(j) Instantaneous picture of spray with 100 bar injection pressure 266.5 CAD before TDCF.

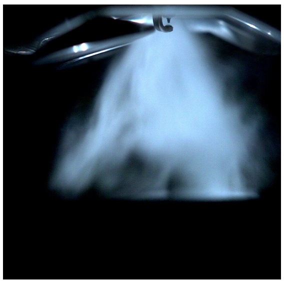
(k) Instantaneous picture of spray with 200 bar injection pressure 266.5 CAD before TDCF.



(l) Instantaneous picture of spray with 350 bar injection pressure 266.5 CAD before TDCF.

Figure 4.4: Instantaneous pictures of the fuel spray at 1750 rpm with SOI 300 CAD before TDCF at low level of restriction.

(a) Instantaneous picture of spray with 100 bar injection pressure 287.3 CAD before TDCF.



(b) Instantaneous picture of spray with 200 bar injection pressure 287.3 CAD before TDCF.

(c) Instantaneous picture of spray with 350 bar injection pressure 287.3 CAD before TDCF.

(d) Instantaneous picture of spray with 100 bar injection pressure 276.1 CAD before TDCF.

(e) Instantaneous picture of spray with 200 bar injection pressure 276.1 CAD before TDCF.

(f) Instantaneous picture of spray with 350 bar injection pressure 276.1 CAD before TDCF.

(g) Instantaneous picture of spray with 100 bar injection pressure 269.7 CAD before TDCF.

(h) Instantaneous picture of spray with 200 bar injection pressure 269.7 CAD before TDCF.

(i) Instantaneous picture of spray with 350 bar injection pressure 269.7 CAD before TDCF.

(j) Instantaneous picture of spray with 100 bar injection pressure 266.5 CAD before TDCF.

(k) Instantaneous picture of spray with 200 bar injection pressure 266.5 CAD before TDCF.

(l) Instantaneous picture of spray with 350 bar injection pressure 266.5 CAD before TDCF.

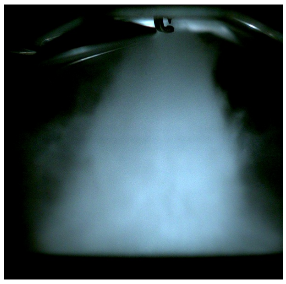
Figure 4.5: Instantaneous pictures of the fuel spray at 1750 rpm with SOI 300 CAD before TDCF at high level of restriction.

4.1.4 2000 rpm, WOT, 1 bar relative intake pressure

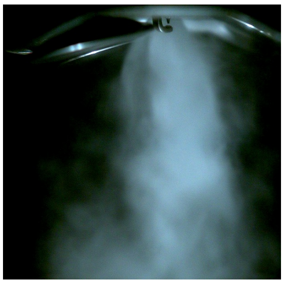
This point is a full load operating where normally a tumble flap would not be utilised. With the unrestricted intake port it was possible to observe a tumble bulk movement for this operating point at 100 bar injection pressure. The high level of restriction was tested to investigate the most extreme condition. The behaviours for this operating point with the unrestricted intake port can be seen in figure 4.6.

(a) Instantaneous picture of spray with 100 bar injection pressure 280.2 CAD before TDCF.

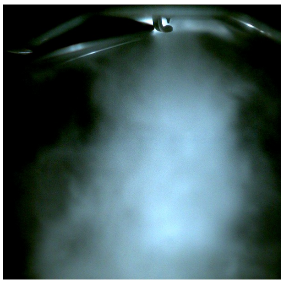
(b) Instantaneous picture of spray with 200 bar injection pressure 280.2 CAD before TDCF.



(c) Instantaneous picture of spray with 350 bar injection pressure 280.2 CAD before TDCF.



(d) Instantaneous picture of spray with 100 bar injection pressure 264.2 CAD before TDCF.



(e) Instantaneous picture of spray with 200 bar injection pressure 264.2 CAD before TDCF.

(f) Instantaneous picture of spray with 350 bar injection pressure 264.2 CAD before TDCF.

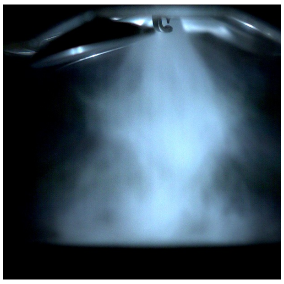
(g) Instantaneous picture of spray with 100 bar injection pressure 248.3 CAD before TDCF.

(h) Instantaneous picture of spray with 200 bar injection pressure 248.3 CAD before TDCF.

(i) Instantaneous picture of spray with 350 bar injection pressure 248.3 CAD before TDCF.

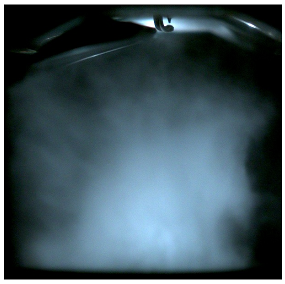
Figure 4.6: Instantaneous pictures of the fuel spray at 2000 rpm with SOI 320 CAD before TDCF with the unrestricted intake port.

The tumble bulk motion is stronger with high level of restriction than seen for throttled operating points at lower engine speeds. The tumble motion has a tendency to cause the spray to hit both the exhaust and intake sides of the cylinder liner. This can bee seen in figure 4.7 but is clearer when studying a film of the spray. The intake air affects the spray late in the intake stroke which also could be seen at the 1750 rpm, 9 bar IMEP operating point. The spray is affected at all fuel pressures.



(a) Instantaneous picture of spray with 100 bar injection pressure 280.2 CAD before TDCF.

(b) Instantaneous picture of spray with 200 bar injection pressure 280.2 CAD before TDCF.



(c) Instantaneous picture of spray with 350 bar injection pressure 280.2 CAD before TDCF.

(d) Instantaneous picture of spray with 100 bar injection pressure 264.2 CAD before TDCF.

(e) Instantaneous picture of spray with 200 bar injection pressure 264.2 CAD before TDCF.

(f) Instantaneous picture of spray with 350 bar injection pressure 264.2 CAD before TDCF.

(g) Instantaneous picture of spray with 100 bar injection pressure 248.3 CAD before TDCF.

(h) Instantaneous picture of spray with 200 bar injection pressure 248.3 CAD before TDCF.

(i) Instantaneous picture of spray with 350 bar injection pressure 248.3 CAD before TDCF.

Figure 4.7: Instantaneous pictures of the fuel spray at 2000 rpm with SOI 320 CAD before TDCF with high level of restriction.

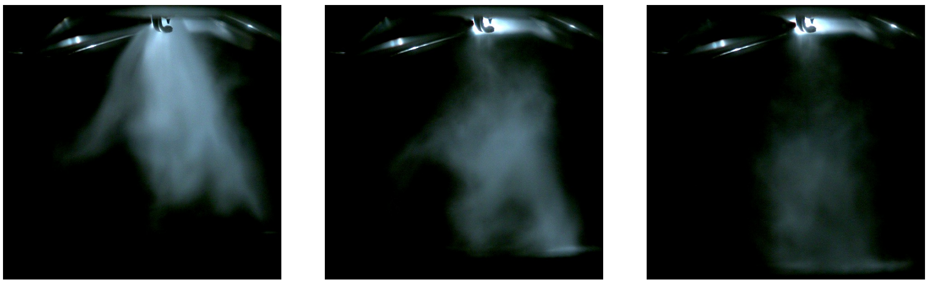
Higher injection pressure gave an increased robustness against the higher intake air velocity. For all injection pressures there was a risk of wall wetting on the intake valve side of the combustion chamber. The tumble motion carried fuel droplets back towards the intake valves and the spark plug before the fuel spray was completely evaporated.

4.1.5 700 rpm, stratified start

The effects of intake port restrictions on the spray could not be seen. The break-up of spray was more efficient for both 200 and 350 bar injection pressure compared to 100 bar injection pressure as seen for other operating points.

4.1.6 1450 rpm, cat-heating

When this point was studied, the effect on the two injections had to be evaluated. By studying the main injection with high speed images it was possible to see a wider spray cone. A less distinctive individual spray suggesting finer break up and further initial liquid spray penetration was also seen with increased injection pressure. The unrestricted intake port had no visible effect on the spray. The behaviour of the spray at 100 bar can be seen in figure 4.8.



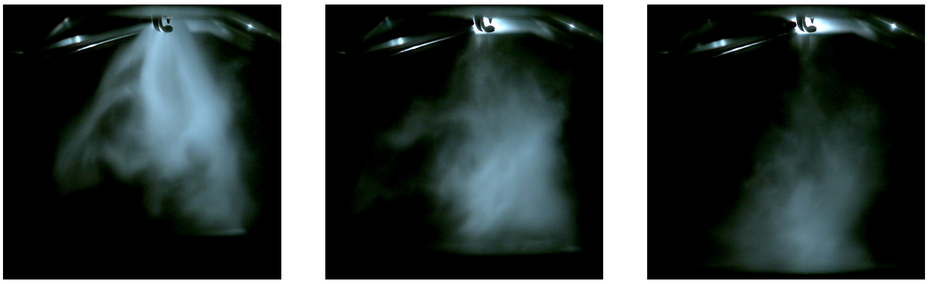
(a) Instantaneous picture of spray with 100 bar injection pressure 272.7 CAD before TDCF

(b) Instantaneous picture of spray with 100 bar injection pressure 266.3 CAD before TDCF

(c) Instantaneous picture of spray with 100 bar injection pressure 261.5 CAD before TDCF

Figure 4.8: Instantaneous pictures of the early main spray for the cat-heating operating point with SOI at 295 CAD before TDCF with deactivated tumble flap.

With a high level of restriction the spray was effected by the intake air and that developed a tumble motion. The spray was more robust when the injection pressure was increased as were seen at other operating points. The tumble motion still had its rotational centre beneath the intake valves, with the risk of wall wetting. The behaviour at 100 bar can be seen in figure 4.9.



(a) Instantaneous picture of spray with 100 bar injection pressure 272.7 CAD before TDCF

(b) Instantaneous picture of spray with 100 bar injection pressure 266.3 CAD before TDCF

(c) Instantaneous picture of spray with 100 bar injection pressure 261.5 CAD before TDCF

Figure 4.9: Instantaneous pictures of the early main spray for the cat-heating operating point with SOI at 295 CAD before TDCF with high level of restriction.

The late post injection was too short to observe any interaction with the intake air flow. Which made it impossible to say anything about the spray interaction of the late injection.

4.2 Combustion study results

This section will display the results obtained from the combustion study. The results were for an operating point which correspond to 1500 rpm and 3.62 bar IMEP in a metal single cylinder engine. In the optical engine 1500 rpm was kept but 3.62 bar IMEP was not achieved with the suggested settings. To keep risks at a minimum it was decided to use the suggested air flow and have a slightly lower IMEP. The operating point was stable with combustion in every cycle. In the pictures the intake valves are placed as in figure 4.10.

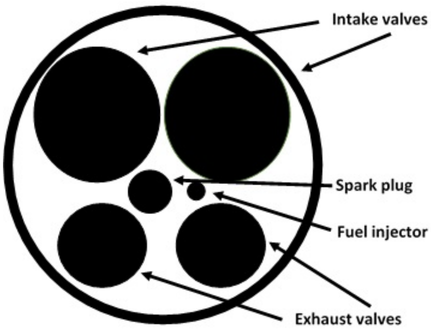
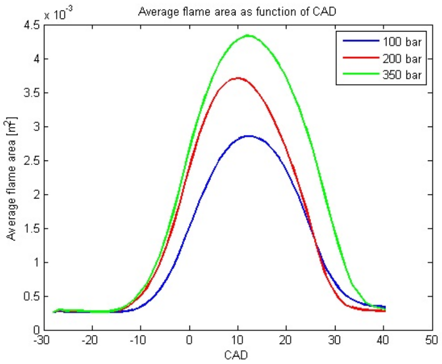


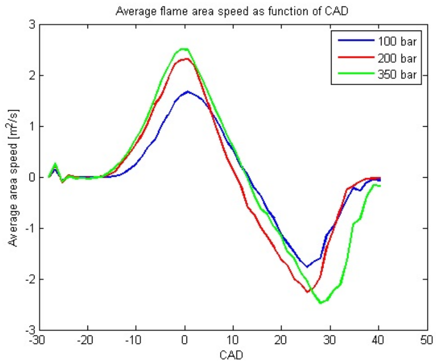
Figure 4.10: Figure showing the principle layout of the roof of the combustion chamber with orientation as in the combustion pictures

4.2.1 Unrestricted intake port

With an unrestricted port the flame propagation changed with increased injection pressure. Figure 4.11(b) shows how the average flame area speed developed with increasing CAD. A negative area velocity means that the flame kernel is decreasing in area. The first entry on the x-axis represents one CAD before ignition timing which was 27 degrees before TDC. The first entry corresponds to the recording trigger. All settings except for the fuel pressure were kept constant. With 100 bar injection pressure, the injection duration was 1.17 ms, for 200 and 350 bar It was 0.83 and 0.65 ms. For further details about the settings with unrestricted port, see appendix 8 for test point 2015-05-05-TF48, 2015-05-05-TF50 and 2015-05-05-TF52. The difference in average flame area and flame area speed can be seen in figure 4.11(a) and 4.11(b). When evaluating the test point at 100 bar, the flame propagated in different directions at different times. This meant that the combustion had finished in different direction at different times, giving lower total flame area. With 200 and 350 bar injection pressure the flame propagated faster and more spherical, according to the combustion pictures and figure 4.11(b). The flame front reached the cylinder walls almost at the same time and thereby had a larger flame area. The ignition delay appeared to be longer at 100 bar compared with higher fuel pressures. Figure 4.12 shows a typical behaviour of the combustion at 1500 rpm, 100 bar fuel injection pressure and unrestricted intake port. As intended, the early flame propagated towards the exhaust valves in an effort to increase the knock limit.

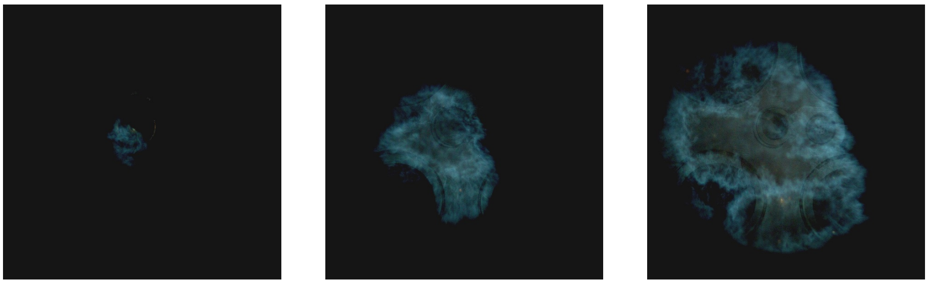


(a) Average flame area as a function of crank angle degree from TDCF for approximately 50 cycles



(b) Average flame area speed as a function of crank angle degree from TDCF for approximately 50 cycles

Figure 4.11: Average flame area and average flame area speed as functions of CAD with open intake port



(a) Combustion with 100 bar injection pressure with open intake port, 15.7 CAD before TDCF and 11.3 CAD after ignition.

(b) Combustion with 100 bar injection pressure with open intake port, 3.4 CAD before TDCF and 23.6 CAD after ignition.

(c) Combustion with 100 bar injection pressure with open intake port, 6.1 CAD after TDCF and 33.1 CAD after ignition.

Figure 4.12: Instantaneous pictures of the combustion at 1500 rpm with 0.58 bar intake pressure.

4.2.2 Low level of restriction

Figure 4.13 shows the flame area speed as a function of CAD for the port with the lowest level of restriction. The behaviour for the flame area speed was similar for all fuel pressures. A problem with diffusion flame on valves and at the spark plug was seen at 100 bar, however it decreased at higher fuel pressures. The flame area speed was significantly higher with the low level of restriction, figure 4.13, compared to an unrestricted port, figure 4.11(b). For details of the test points, see Appendix 8 and test 2015-05-05-TF49, 2015-05-05-TF51 and 2015-05-05-TF53. Figure 4.14 shows the flame propagation with low level of restriction. Figure 4.14 shows the flame at the same CAD as in figure 4.12, with the same spark timing. The bright orange or yellow areas in figure 4.14(c) was thought to be diffusion flames.

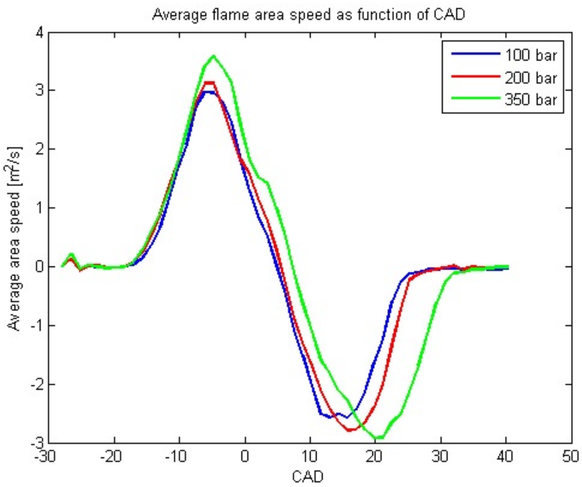
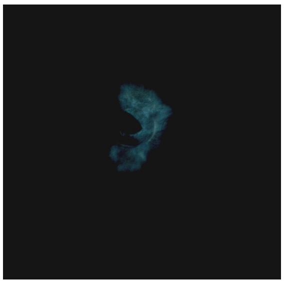
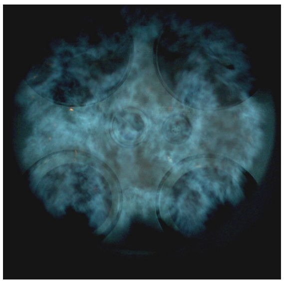


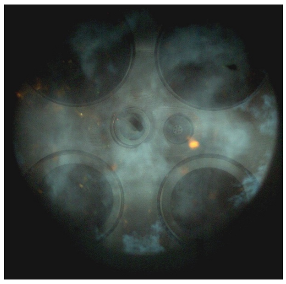
Figure 4.13: Average flame area as a function of crank angle degree from TDCF for approximately 50 cycles with low tumble level



(a) Combustion with 100 bar injection pressure and low level of restriction, 15.7 CAD before TDCF and 11.3 CAD after ignition.



(b) Combustion with 100 bar injection pressure and low level of restriction, 3.4 CAD before TDCF and 23.6 CAD after ignition.



(c) Combustion with 100 bar injection pressure and low level of restriction, 6.1 CAD after TDCF and 33.1 CAD after ignition.

Figure 4.14: Instantaneous pictures of the combustion at 1500 rpm with 0.58 bar intake pressure and low restriction level of restriction with 100 bar injection pressure.

4.2.3 Medium level of restriction

At medium level of restriction the flame area speed increased compared to low levels of restriction, as can be seen comparing figure 4.15 and figure 4.13. With the increased level of tumble, the early combustion direction was not as consistent as with an unrestricted port. The flame propagated more often towards the intake side instead of towards the exhaust valves. The flame propagation can be seen in figure 4.16. Another phenomenon that the could be observed in certain cycles, was that the flame propagation was significantly slower between the exhaust valves which can be seen in figure 4.16(c).

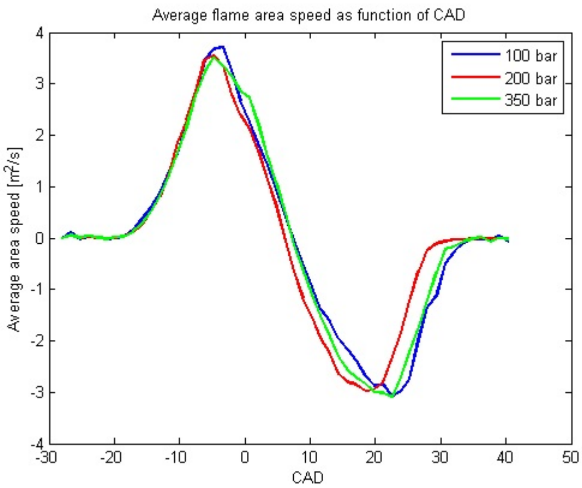
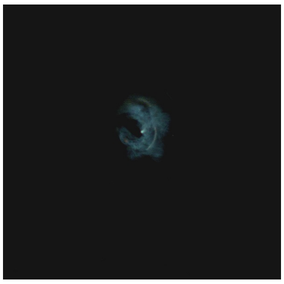
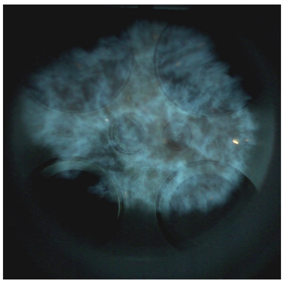


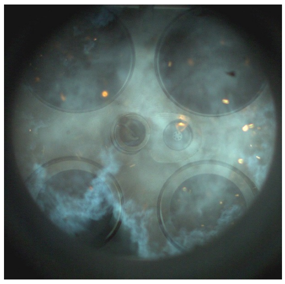
Figure 4.15: Average flame area as a function of crank angle degree from TDCF for approximately 50 cycles with with medium tumble level



(a) Combustion with 100 bar injection pressure and medium level of restriction, 15.7 CAD before TDCF and 11.3 CAD after ignition.



(b) Combustion with 100 bar injection pressure and medium level of restriction, 3.4 CAD before TDCF and 23.6 CAD after ignition.



(c) Combustion with 100 bar injection pressure and medium level of restriction, 6.1 CAD after TDCF and 33.1 CAD after ignition.

Figure 4.16: Instantaneous pictures of the combustion at 1500 rpm with 0.58 bar intake pressure and medium restriction level of intake port.

4.2.4 High level of restriction

At the highest level of restriction the flame area speed appeared to be faster. Figure 4.17 shows the average flame area speed as a function of CAD. The combustion illuminated the combustion chamber ahead of the flame front. This caused a problem for the flame area algorithm that searches for pixels above a certain intensity, further discussion is carried out in chapter 5.3. The early flame propagated towards the intake side more frequently with high level of restriction compared to unrestricted as can be seen in figure 4.18. There was a risk for diffusion flames which also could be seen. The flame propagation appeared to be slower between the exhaust valves as seen in figure 4.18(c).

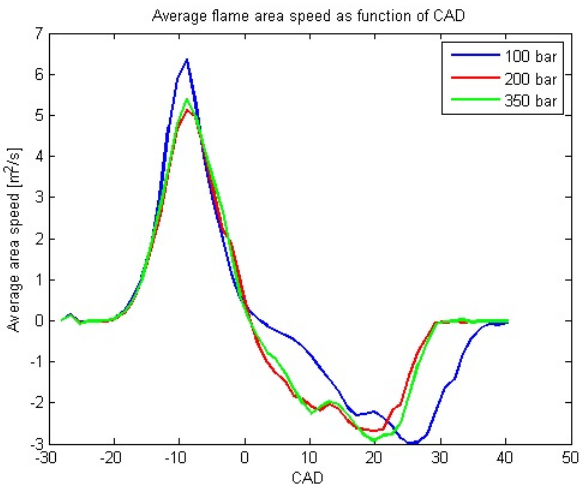
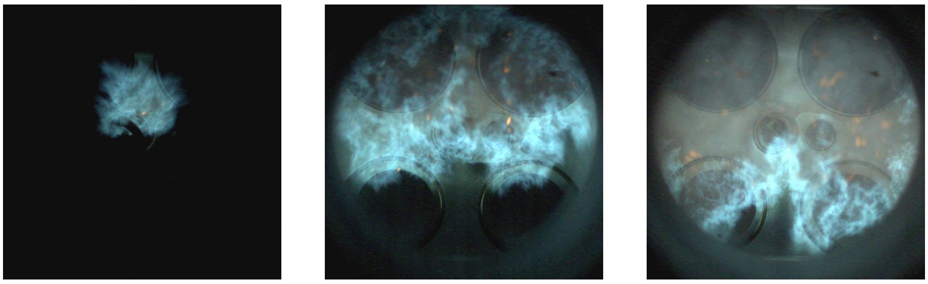


Figure 4.17: Average flame area as a function of crank angle degree from TDCF for approximately 50 cycles with high tumble level



(a) Combustion with 100 bar injection pressure and high level of restriction, 15.7 CAD before TDCF and 11.3 CAD after ignition.

(b) Combustion with 100 bar injection pressure and high level of restriction, 3.4 CAD before TDCF and 23.6 CAD after ignition.

(c) Combustion with 100 bar injection pressure and high level of restriction, 6.1 CAD after TDCF and 33.1 CAD after ignition.

Figure 4.18: Instantaneous pictures of the combustion at 1500 rpm with 0.58 bar intake pressure and high restriction level of intake port.

4.2.5 Search for MBT-timing

With the increased level of restriction, the combustion was faster which moved CA 50 % to a few degrees before TDC. The CA 50 % was taken from the logging computer. It should be around 10 CAD after TDC for maximum torque [15]. With the settings used throughout the experiments with the unrestricted port, the CA 50 % were at around 11 to 13 CAD after TDC. Figure 4.19 shows how the value of CA 50 % changed with the spark timing for the different levels of restriction. For low level of restriction, CA 50 % reached the target value with less adjustment of the spark timing than for medium and high level of restriction. This test sequence was performed at 200 bar injection pressure. Figure 4.20 shows the average flame area speed for all levels of restriction and with spark timing at MBT. The cases were not perfectly comparable since the CA50 % value differed with a few of degrees. The spark timings used were however 12, 12, and 14 degree before TDCF. With the unrestricted port, the MBT is approximately 28 degrees before TDCF.

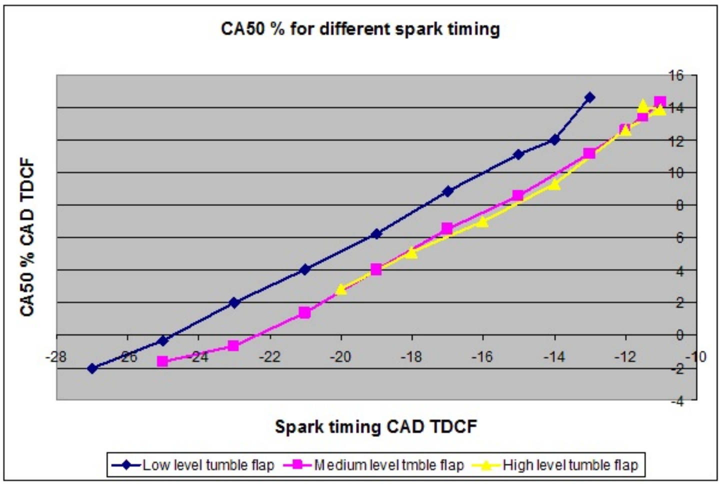


Figure 4.19: CA50 % as a function of ignition timing

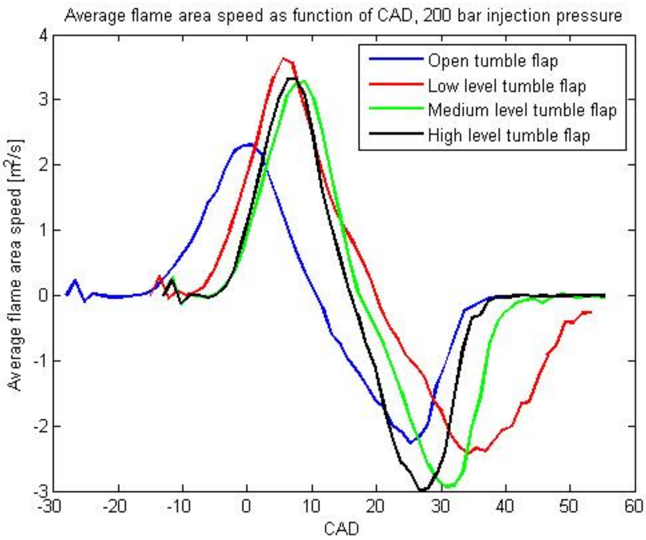


Figure 4.20: Average flame area speed as a function of crank angle degree from TDCF for approximately 50 cycles with different levels of tumble flap at 200 bar injection pressure

5. Discussion

The following sections will mention uncertainties, discuss possible causes to problems and discuss the results from the study. It will make it possible to get an explanation to seen behaviours of both spray and combustion.

5.1 Theory discussion

Theory and literature show that a tumble flap decreases the volumetric efficiency due to lower air mass flow. But since the usage of the tumble flap is at low loads and low engine speed, it will instead promote a stable combustion. High volumetric efficiency and high levels of tumble are both beneficial for the combustion because more fuel can be burnt, hence more power. It is theoretically explained and shown, especially in [3], that increased tumble may decrease the cycle to cycle variations due to a faster and more stable combustion. Increased tumble is directly linked to increase in-cylinder turbulence, which is beneficial for the flame propagation. Another aspect that has been studied in the shown reports is the possibility of having half open tumble flaps. The benefits were however minor compared to completely deactivated flap.

5.2 Discussion of the spray study

The results from the spray study showed increased tumble motions with increased level of restriction. However the injection timing and duration are of great importance when filming the spray. The tumble motion need time before it is strong enough to affect the spray. Since the target is to have realistic injection timing and duration, the fuel might be fully evaporated before any extra tumble motion can be seen.

5.2.1 Spray bomb

Increasing the fuel pressure seemed to enhance the air entrainment due to an observed increase in the size of the vortices at the spray periphery. To confirm this PIV measurements must be made, with HSV it is only possible to see tendencies that suggest this behaviour. According to fuel jet theory an increased fuel pressure leads to higher momentum of the fuel spray. An increased spray penetration can also be seen due to higher momentum. There seems to be more efficient spray breakup at higher injection pressures. This can not be confirmed without optical methods that specifically observes the sizes. The increased fuel pressure caused increased turbulence and cavitation in the nozzle exit. This results in enhanced breakup and a "brushier" appearance. When the spray has reached a certain distance into the spray bomb, the high pressure spray tends to stop faster. This is probably because of the more efficient breakup leading to smaller particles that are more affected by drag phenomena. This would mean that the risk for wall wetting is higher at low pressures due to reduced air entrainment and larger particles.

5.2.2 1500 rpm, 6 bar IMEP

The effect on the spray did not seem to be affected by the tumble flow. This is not completely true, since the spray particles have evaporated before the increase in TKE affects the spray. Since it is not visible the spray can not be followed with the optical method used. This operating point has very early injection which means there is not enough time to build up a tumble motion before the spray evaporates. This means that the risk for wall wetting of fuel droplets may be smaller due to that fuel particles have evaporated before the tumble movement is strong enough. This is the reason why the 9 bar IMEP point is used in the spray study to have a spray that exists long enough to get an understanding about the spray interaction with the tumble flow.

5.2.3 1750 rpm, 9 bar IMEP

For this point the injection was long and late enough to study the spray effects caused by the increased TKE, due to the restricted port. The TKE had a slow build up before it effected the fuel spray, this was especially easy to see for the 100 bar injection pressure. The tumble motion momentum probably needs time before it is strong enough to effect the high momentum of the fuel spray. The behaviour of the lower injection pressure, 100 bar, suggests that the spray has less momentum than at higher injection pressures. Despite that the larger particles have larger mass and therefore could achieve high momentum. Therefore increased fuel pressure affects the spray by giving the spray higher velocity and increase the resulting momentum of the spray. This gives the behaviour of less spray robustness at lower fuel pressures.

Lost robustness in the spray at lower injection pressures may cause wall wetting i.e. fuel hitting the cylinder liner. With the larger fuel spray droplets, it may give increased risk of flame quenching when the flame approaches the wall and can also cause diffusion flame on the cylinder wall. When the spray particles are deposited on the cylinder liner, a heat transfer between the fuel droplets and the cylinder liner occurs to evaporate the fuel. This will lower the wall temperature causing wall quenching of the flame and also due to high concentration of fuel. The effect is there will be an increased amount of UHC i.e. more emissions. It is partly because UHC may come from crevices where fuel has been deposited. Also because the fuel entering crevices when the piston passes the fuel film caused by the intake air. The lower fuel pressure also shows a heightened risk of liquid on the valves and spark plug due to the lessened robustness. This had the risk of causing locally rich zones that might burn with a diffusion flame. It would in turn mean that particle emissions would increase. Increased fuel pressure gave an improved robustness of the spray cone, the effect of tumble was less visible on the not yet evaporated spray. This does not mean anything bad since the spray is almost completely evaporated before the tumble starts to give a visible effect. This behaviour is favourable since it is important that the spray is not affected so much by the intake air. As mentioned earlier, if the spray is strongly affected by the intake air it may cause wall wetting on cylinder liner which is not wanted.

The injected spray seems to be kept inside an air cushion that makes the spray cone more stable on the exhaust side of the combustion chamber. This can help prevent wall wetting on the exhaust side. This is probably because of the tumble motion that builds up before the SOI which acts as the air cushion that was observed. This can be seen especially for 100 bar injection pressure where the spray interacts strongly with the intake air and the effect explained above is visible. This possibly has a positive effect as it will promote mixing with the increased air motion and also prevent wall wetting on the exhaust side. Still there is a risk of wall wetting on the intake side which may cause wall quenching or diffusion flames. This is because the rotational centre moves towards the intake side of the cylinder liner as mentioned in the results.

The difference that can be seen in the affect from increasing restriction is as predicted by theory a promoted intake air velocity and mixing. This can be seen when the low level of restriction is used, the effect on the spray is smaller, not giving the clear vortex. This should in turn give less TKE and therefore the change in the turbulent flame velocity will be smaller. Therefore it will lead to longer combustion duration and therefore more time for wall heat transfer which is not wanted. The effect of high level of restriction gives more TKE and therefore the mixing should be promoted more and the combustion should be even faster.

5.2.4 2000 rpm, 1 bar relative intake pressure

For this point, as mentioned in the result section, the effect on the spray will cause it to hit both sides of the cylinder liner with high level of restriction. This is due to the increased velocity of the intake air due to increased engine speed. It will cause an even more aggressive flow into the cylinder causing the very strong movement of the spray, which is observed. This operating point is a full load and for this type of operating points the intake air has such a high velocity that there already is a strong tumble motion in this intake port. The ports themselves have such a design that they promote tumble. Therefore the additional increase in the intake air velocity may not be needed to promote the mixing. The risk with using the system in a full load operating point is that unwanted fuel is being deposited on valves and the cylinder wall. It will imply a heightened risk of wall quenching, leading to more UHC and locally rich zones. As mentioned before, a diffusion flame gives more soot emissions.

When the 350 bar injection pressure was tested, the risk of wall wetting on the exhaust side was lessened compared to lower fuel pressures. This means that it is possible to use a tumble flap for a relatively high engine speed. It is possible to promote the mixing even for this point and possible to get turbulence levels that are comparable to higher engine speeds. This has the possibility to achieve a more stable combustion and faster combustion although the possibilities for improvement are lower due to the character of the operating point.

5.2.5 700 rpm, stratified start

For this operating point there is a late and relatively short injection which makes it impossible to see a visible effect on the spray. The wanted effect is that the stratified charge would have improved mixing and therefore burn with a premixed flame. This would make the operating point more stable and therefore make it possible to run with leaner mixtures, while still having stable combustion i.e. low coefficient of variation in IMEP. It would be globally lean with a combustible mixture close to the spark plug.

5.2.6 1450 rpm, Cat-heating

For this operating point it was impossible to see an effect on the late spray and this is expected since it is after TDC. Here the turbulence should have died out and therefore are diffusion flames still a problem.

For the early main injection there occurred, as said before, a clear tumble motion. This would suggest that at the 1500 rpm point there exists an increased level of tumble, but it is not seen due to the very early injection. The reason was an early evaporation due to the short injection. This shows that it should be a tumble flow that will promote the mixing for 1500 rpm which shows the importance of deciding the injection strategy. At the early injection of the 1500 rpm, 6 bar operating point, the spray has evaporated before the tumble motion is built up enough to be visible. The spray study is mostly used to see how the intake air affects the spray as such. It is complicated to use the spray as an indication of the tumble level since the spray evaporates and becomes invisible with the illumination method that was used in the study.

5.3 Discussion of the combustion study

The combustion part of the project was only carried out with one operating point. It is therefore difficult to predict how the behaviour seen in this point may translate to other load cases. A comparison between 3.62 and 6 bar IMEP at the same engine speed could display possible trends. It would give an indication how the tumble flap can be used at a wider range of operating points. Due to shortage of time other load cases could not be tested.

To minimise the differences between a metal and an optical single cylinder engine a similar engine should be used. It can also be possible to isolate the combustion chamber in the optical engine and very strictly control its temperature in order to mimic a metal engine. The main cause for the differences is due to different engine temperatures. Even though the coolant and oil temperatures are approximately the same, the engine structure might not have reached the right temperature which affects the measurements. Using more temperature sensors and an improved controller for the temperature of the fluids may lower the variations. In the optical engine the piston rings are much further below the piston crown compared to a metal engine which leads to larger crevices. The glass has different heat conductivity than the metal which makes the heat transfer through the cylinder walls different. An important difference between the metal engine and the optical engine is the cam phasing which is not as variable in the optical engine due to no VVT. This makes differences in gas flow and the amount of throttling that will be needed for the same load case.

5.3.1 Levels of restriction and injection pressure

With an unrestricted intake port the combustion was irregular, i.e. CoV is higher at 100 bar injection pressure. The improved mixing that comes with higher injection pressure appeared to improve flame propagation and with regard to cycle to cycle variations. However, there might be a margin of error in the method used to approximate the average flame area speed. With the CA 50 % and the pressure trace, there seemed to be a correlation between faster flame area speed and faster combustion. The method uses all cycles in a test sequence to calculate the mean value which might decreases its accuracy. It is possible that extreme cycles will change the average flame area speed so it is not representative. It is possible that a strange or extreme mean flame area speed could indicate a problem that might need further investigation. As seen in some cases, the combustion chamber is illuminated ahead of the flame front which may triggers the criteria for combustion. It will make the flame propagation appear faster than it is.

With increased level of restriction the flame area speed became higher. Due to low engine speed, 1500 rpm, the natural tumble motion in the engine is weak. At higher injections pressures the fuel spray was more efficiently dissipated and the mixing throughout out the combustion chamber was improved. Increased restriction of the intake port increased the turbulence, seen due to faster flame propagation compared to an unrestricted port. At all levels of restriction, the difference in flame propagation between the injection pressures was less obvious. Low injection pressures gave a higher risk for diffusion flame on valves, spark plug and pool fires with a restricted port. Higher injection pressures reduced the occurrence of diffusion flames on the mentioned parts. The study was made in an optical engine that is different from a metal engine, where the combustion may behave differently. Liquid fuel deposited on valves etc. may not be a problem with correct cam phasing. Strong tumble motion brought fuel to the roof of the combustion chamber and deposited fuel on valve seats or in the spark plug etc. Higher injection pressure creates smaller droplets that evaporate faster and therefore less fuel will be deposited in crevices. In a metal engine are the temperatures higher which will improve the evaporation of the fuel spray and lessened the fuel deposition.

5.3.2 MBT-Timing

It is said that 50 % of the energy has been released should occur seven CAD after TDCF. With the ignition timing used for an unrestricted port, CA50 % occurred 12 to 14 degree after TDCF. With restricted port with the same spark timing, CA50 % was close to or even before TDC. The ignition timing was retarded until the same CA50 % was achieved. The ignition had to be retarded 16 degrees when using the high level of restriction. For low and medium level of port restriction, the ignition was retarded 14 and 16 degrees respectively. With the ignition adjusted to the CA50 % for an unrestricted port approximately the same IMEP was achieved. The prediction would be a higher IMEP due to the faster combustion. The optical engine should not be used to make predictions about engine torque output since it behaves differently than a metal engine. The cam phasing might also be less suitable for the restricted port compared to the unrestricted port.

5.4 Discussion of possible sources of error

To understand some of the differences in the operating points the effects caused by errors in measurements or the post processing of the pictures had to be analysed. This is because it is important that the methods are stable so that the results are representable

A problem in the study was a MAP that returned incorrect values. This because it had not been calibrated before the study and its placement in the intake port. For this reason the intake air mass flow differed between operating points which had the same input parameters. Therefore some input parameters had to be adjusted to achieve the same air mass fow. Lambda, λ, is especially important for the combustion study since running at other mass flow would cause a difference in λ and may give a low λ if the amount of fuel injected is not adjusted. The manifold air pressure deviated between 1 and 6 kPa from the actual value at WOT giving a deviation of 1 to 6 %. The deviation was allowed since a MAF sensor was used to check the air mass flow into the engine. Therefore the MAP values are not exact enough in the test plan due to that it was not used to get the correct air mass flow. One source of error was the adjustability of the intake and exhaust cams, it deviated at most 5 degrees from the suggested value. It could result in more rest gases being trapped inside the combustion chamber, which makes it more difficult to combust the mixture. The calculated fuel mass deviated for this reason from the actual value for stoichiometry. The cam phasing also lead to different air mass flow of the intake air compared to the suggested value. One possible error was that the pressure trace maximum, which could be observed in the logging computer, was not placed just before TDC. This was checked and it was concluded that it was placed at the reference position

For some operating points there was a difference of the coolant temperature of 15 °C. This was due to a tight schedule and meant that sometimes the engine was not at the reference temperature. Different conditions in the test cell and what type of measurements that were done also affected the temperature due to an inaccurate temperature controller. This could result in a difference in the mass airflow into the engine and the evaporation of the spray. This might make the spray evaporate faster for those test points close to 88 °C compared to the test points carried out at 75 °C, see 8. This was not considered a problem for the spray study where spray interaction trends were of interest. The temperature differences were taken into consideration, for the spray study. For the combustion study the temperature of the coolant was kept around 85 °C. The unfortunate differences in the operating temperature for three tests points were the result of an attempt to control the water temperature more exact, the air mass flow difference was compensated different throttling. The average operating temperature of the coolant was approximately 83 °C. One of the test points at lower coolant temperature was retested at the average temperature but no difference in flame propagation and flame behaviour could be seen. There was always a reference point test made in the beginning of every day to see that operating parameters had not changed.

5.4.1 Post processing errors

The threshold algorithm for the flame propagation checked, as mentioned before, the differences in intensity for a set number of degrees around the middle of the combustion chamber. There are other methods that can be used, as edge functions in Matlab that follow edges in a picture. It was not used since it would require that nearly every picture in every cycle had to be analysed manually in case the flame should die out in the middle. The Matlab code would register this as an edge and therefore the flame area would not be representative. For this reason the algorithm written for this study was used.

To investigate if the flame area speed algorithm returns valid results, the standard deviation of the area speed for an operating point was checked. The operating point for the investigation the combustion study operating point with an unrestricted intake port, for details see test point "2015-05-05-TF 50" in appendix 8. The results of the investigation returned a standard deviation of 100 % for the part of the combustion event that was investigated, this can be seen in figures 5.1 and 5.2.

The early flame will react differently between cycles, it will move differently and have different shapes. The early flame will also start at slightly different crank angles in different cycles. In figure 5.1 the peak is at -15 CAD due to the difference of flame behaviour between the cycles at that CAD. The peak at around 10 CAD in figure 5.1 is when the flame has reached the cylinder wall and starts to diminish. The combustion takes place between these peaks and the standard

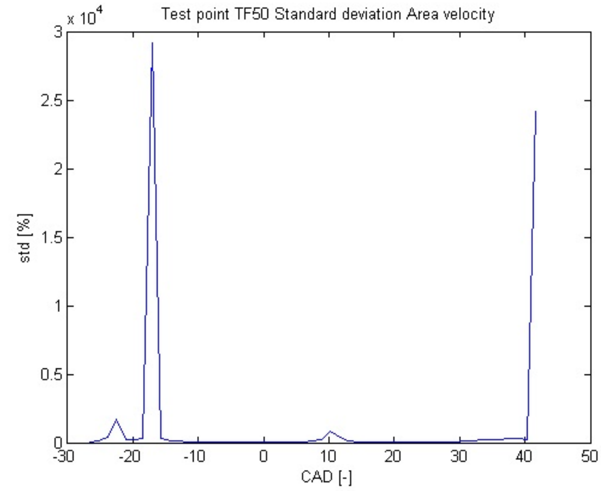


Figure 5.1: The figure shows the standard deviation of the flame area velocity over 50 cycles

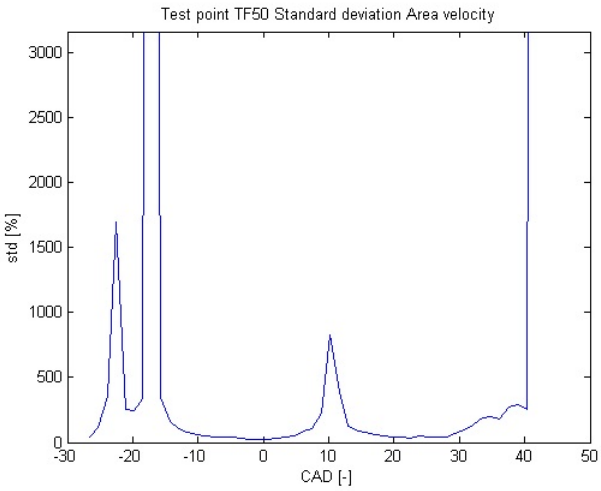


Figure 5.2: The figure shows a zoomed in picture of figure 5.1. Enhances the common standard deviation of the flame area velocity over 50 cycles

deviation is lower than 100 % as can be seen in figure 5.2. Because of the high standard deviation, the flame area speed plots can only be seen as a trend in the rate of increase in flame area and not as absolute values. The standard deviation of the IMEP for the combustion at this point was approximately 0.11 bar and for a restricted port below 0.1 bar. This means that the combustion was stable and the high standard deviation suggest the flame area speed algorithm was probably not suitable for the study. This shows one of the limitation of the high speed video method. It is not relatively photosensitive compared to other optical methods. Since the filming of the combustion was made from underneath, the standard deviation does not take into account the three-dimensional nature of the flame.

5.4.2 Sources of error within the spray study

Only the trends was interesting for the spray study, therefore no statistical evaluation was made. The lamp that was used to illuminate the combustion chamber gave a concentrated light. The reflections were dominant and the spray was not visible close to the injector. This was not considered a problem since the importance was to see where the tumble motion centre moved in relation to the combustion chamber.

5.5 Overall discussion

The combustion part of this optical study suggested that the combustion would be faster i.e. the combustion duration would be shorter with a restricted port. This may give potential improvements in the indicated efficiency since there will be less time for heat transfer to the wall. It can give an improvement in the fuel consumption of an engine. The decreased combustion duration open up the possibility for ignition retardation. That would mean a larger fraction of the power is produced in the expansion stroke, thus improving IMEP. It will also mean that the relative expansion ratio is increased. This was seen in the optical engine as explained in the discussion of the combustion study where a lower IMEP was received with the reference ignition timing. With restricted port the engine had a lower air mass flow of the intake air meaning the engine does not need to be throttled as much as with an unrestricted port. It will decrease the pump work that the engine has to do. Also the faster combustion increases the efficiency due to more heat release in the beginning of the expansion stroke. The observed improvements may correspond to a decreased fuel consumption in a production engine. This is linked to a decrease in CO_2 emissions, which can be essential to meet emissions standards like the EURO 6 and coming EURO 7.

One operating point that was not tested with combustion was a stratified operating point. According to theory, this operating point would have improved combustion due to better mixing. It may give less cycle to cycle variation and therefore a more stable operating point. Therefore it could be possible to run with leaner mixtures and improve fuel efficiency.

For the spray and combustion study a problem with restricted ports was the risk for wall wetting and liquid fuel deposits. It was confirmed in the combustion study that diffusion flame combustion occurred on valves and spark plug. The effect decreased at higher injection pressures which can be linked to the spray behaviour. Increased fuel pressure meant more efficient break up and enhanced air entrainment which aids the evaporation of the spray. At lower fuel pressures the breakup results in larger fuel particles. They are also more affected by the intake air due to lower initial momentum and it could be the reason why they hit the cylinder liner and crevices before they were evaporated. This will lead to locally rich areas, meaning relatively high particle emissions due to the diffusion flame combustion. When comparing the results from the studies mentioned in the theory section with the results and observations from this study it is possible to see a correlation. The other studies have concluded that the CoV decreased with increased restriction of the intake port which was also seen for this study. This would suggest that combustion results from this study are valid.

As said before retardation of the ignition can be possible with a restricted intake port. The problem that can occur according to theory is that the turbulent kinetic energy will decrease close to TDC and the flame may propagate slow between the exhaust valves which may give time for knock. It is possible that no time was given to create conditions for knock. However the faster combustion mean a quicker pressure rise and could cause knock.

5.6 The future of tumble flap

The next step in the evaluation of the tumble flap system should be to perform further experiments in the optical engine. It would be beneficial to test on a higher load than 3.6 bar IMEP at 1500 rpm, this will give information of how the tumble flap affects the combustion for different load cases. Possibly PIV studies should be made to better understand the flow conditions late in the intake and compression stroke. To further understand how the spray interacts a spray study should be conducted from underneath. This to get a three dimensional view of the spray interactions, so it is possible to design the combustion system correctly. A combustion study using the side view would be of interest to investigate where the diffusion flames originate from. Running the engine with a stratified charge operating point would be useful since it is dependent of the in cylinder turbulence for mixing and flame propagation. Since there was no time to perform these experiments in this study it should be studied both in the optical engine and later in a full metal engine. The cat heating operating point should be studied in the optical engine to see if the tumble flap affects the combustion in terms of diffusion flames.

In a single cylinder metal engine, the work made in the optical engine can be verified and later tried at higher load cases. Since it is possible to operate a metal engine at continuous condition, it is possible to get more stable combustion. This will eliminate the uncertainties in stability and load caused by a skip fire scheme. In the metal engine it is important to try to find the limit of usefulness with the tumble flap. In a metal engine with correct camshafts and VVT it would be possible to adjust them to optimum setting for each tumble flap restriction. It is needed to do emission studies to either confirm or dismiss the troubles with diffusion flames seen in the study.

Fuel tracer LIF would be a good complement, it would display how the fuel is distributed in the combustion chamber. Unpublished CFD calculations combined with some of the films of the combustion indicates that there is relatively low level of turbulence between the intake valves and between the exhaust valves. There would be a need to study the fuel distribution in two planes, one that can observe the fuel distribution in the tumble plane and one plane between the intake ports.

5.6.1 Recommendations

It has been shown that a tumble flap is beneficial for the flame propagation. It should always be tried to use the high level of restriction, although it may cause an increase in pumping losses. The combustion duration is much lower with the high level of tumble flap which means that the ignition can be retarded up to 16 degrees. Also it is recommended to use a high fuel pressure to minimise the risk for diffusion flame, since it occurred more frequently with the tumble flap restriction and low fuel pressures. The risk of diffusion flame due to wall wetting might be lessened with an adjusted spray targeting. It is recommended to use 350 bar injection pressures with tumble flap restriction due to high risk of wall wetting with low fuel pressures.

6. Conclusions

With tumble flap restriction there is an increased risk of wall wetting on the intake side.

With increased injection pressure the spray cone is less affected by the increase of intake air velocity caused by the tumble flap restriction. This is due to higher momentum of the fuel droplets.

With tumble flap restriction the combustion becomes more stable i.e. the CoV for IMEP was lower.

With tumble flap restriction it is more likely to have unwanted combustions from fuel in crevices due to the tumble motion depositing more fuel in crevices like valve seat and spark plug.

A tendency of less fuel deposits in crevices is noticed if the fuel pressure is higher. It is because of the more efficient break up of fuel at higher fuel pressures.

The combustion duration is significantly shorter with a tumble flap restriction. This is due to increased level of turbulence in the combustion chamber.

The faster flame propagation may improve the knock resistance due to less time for the conditions for knock to build up.

In the optical engine, the ignition could be retarded up to 16 degrees with high level of tumble flap restriction in order to have the same CA50 %. It means the reduced combustion duration will allow more power to be taken out in the expansion stroke and a larger expansion ratio i.e. higher thermodynamic efficiency.

7. References

[1] Combustion: From Basics to Applications. *Maximilian Lackner,Árpád Palotás,Franz Winter.* Wiley-VCH Verlag, 2013.

[2] Ulf Aronsson. "Processes in Optical Diesel Engines -Emissions Formation and Heat Release". PhD thesis. Lund univerity of technology, 2011.

[3] Andreas W. Berntsson et al. The Effect of Tumble Flow on Efficiency for a Direct Injected Turbocharged Downsized Gasoline Engine. **4** (Sept. 2011), 2298–2311. DOI: 10.4271/2011-24-0054. URL: http://dx.doi.org/10.4271/2011-24-0054.

[4] Martin Blell et al. "Modellering av motor till Chalmers EcoMarathon". PhD thesis. Chalmers univerity of technology, 2013.

[5] Francesco Catapano, Paolo Sementa, and Bianca Maria Vaglieco. Optical characterization of bio-ethanol injection and combustion in a small {DISI} engine for two wheels vehicles. **106** (2013), 651–666. ISSN: 0016-2361. DOI: http://dx.doi.org/10.1016/j.fuel.2012.11.064. URL: http://www.sciencedirect.com/science/article/pii/S0016236112009970.

[6] He Changming and Xu Sichuan. The Investigation and Application of Variable Tumble Intake System on a GDI Engine. **7** (Oct. 2014), 2022–2034. DOI: 10.4271/2014-01-2885. URL: http://dx.doi.org/10.4271/2014-01-2885.

[7] Jr Charles E. Baukal. *ndustrial Combustion Testing.* CRC Press, 2010.

[8] Petter Dahlander. *GDI Combustionsystems part2: An introduction to sprays, for internal combustion engines.* Published on course MTF225 webpage. Lecture notes. 2014.

[9] Sven Linow Egon P Hassel. Laser diagnostics for studies of turbulent combustion (1999).

[10] Tiegang Fang, Yuan-Chung Lin, and Tien Mun. Biodiesel combustion in an optical {HSDI} diesel engine under low load premixed combustion conditions. **88**.11 (2009), 2154–2162. ISSN: 0016-2361. DOI: "http://dx.doi.org/10.1016/j.fuel.2009.02.033. URL: http://www.sciencedirect.com/science/article/pii/S0016236109000994.

[11] J. Fischer et al. "Influence of an Adjustable Tumble-System on In-Cylinder Air Motion and Stratification in a Gasoline Direct Injection Engine". *SAE Technical Paper.* SAE International, May 2002. DOI: 10.4271/2002-01-1645. URL: http://dx.doi.org/10.4271/2002-01-1645.

[12] Torbjörn M. Green. "PIV in Practice". PhD thesis. Luleå University of Technology, 2011.

[13] Benjamin T. Johnson Loughborough Univ. Graham K. Hargrave, Benjamin A. Reid Loughborough Univ Vivian J. Page, and Stuart wagstaff Perkins Engines Co Ltd. *Optical Analysis and Measurement of Crankcase Lubricant Oil Atomisation. 2012-01-0882.* Tech. rep. SAE International, 2012.

[14] W. Hentschel et al. "Optical Diagnostics and CFD-Simulations to Support the Combustion Process Development of the Volkswagen FSI® Direct-Injection Gasoline Engine". *SAE Technical Paper*. SAE International, Sept. 2001. DOI: 10.4271/2001-01-3648. URL: http://dx.doi.org/10.4271/2001-01-3648.

[15] John B. Heywood. *Internal combustion engine fundamentals*. Ed. by Anne Duffy and John M. Morriss. McGrew-Hill, 1988.

[16] Bengt Johansson. *Förbränningsmotorer*. Department for Combustion Engines Lunds Institute of Technology, 2006.

[17] Makoto Kaneko, Yuji Ikeda, and Tsuyoshi Nakajima. "Tumble Generator Valve (TGV) Control of In-Cylinder Bulk Flow and Its Turbulence Near Spark Plug in SI Engine". *SAE Technical Paper*. SAE International, Mar. 2001. DOI: 10.4271/2001-01-1306. URL: http://dx.doi.org/10.4271/2001-01-1306.

[18] Günter P Merker, Christian Schwarz, and Rüdiger Teichmann. *Introduction to internal combustion engine*. Springer-Verlag Berlin Heidelberg, 2012.

[19] Richard Pekelney. *C.F. 'O' CLASS SUBMARINES*. 2011. URL: http://archive.hnsa.org/doc/oberon/diesel/index.htm#toc (visited on 01/29/2015).

[20] Christof Schulz and Volker Sick. Tracer-LIF diagnostics: quantitative measurement of fuel concentration, temperature and fuel/air ratio in practical combustion systems. 31 (2005), 75–121.

[21] Florian Schulz et al. Gasoline Wall Films and Spray/Wall Interaction Analyzed by Infrared Thermography. **7** (Apr. 2014), 1165–1177. DOI: 10.4271/2014-01-1446. URL: http://dx.doi.org/10.4271/2014-01-1446.

[22] Johan Sjöholm. "High repetition rate laser diagnostics for combustion applications". PhD thesis. Lund univerity of technology, 2012.

[23] Cody William Squibb et al. "A Demonstration of Simultaneous Infrared and Visible Imaging Techniques with Pressure Data in an Optically Accessible Diesel Engine Operating at Part Load with High EGR". *SAE Technical Paper*. SAE International, Apr. 2011. DOI: 10.4271/2011-01-1395. URL: http://dx.doi.org/10.4271/2011-01-1395.

[24] Richard Stone. *Introduction to internal combustion engine*. Fourth edition. Palgrave Macmillan, 2012.

[25] Martin Tuner. *Advanced Combustion Engine Concepts, lecture 4, Measurement techniques*. Published on course MVKN55 webpage. Lecture notes. 2013.

[26] Martin Tuner. *Advanced Combustion Engine Concepts, lecture Advanced Otto*. Published on course MVKN55 webpage. Lecture notes. 2013.

[27] *What is CMOS camera. What is it different from CCD camera*. 2006. URL: http://www.mintron.com/htm/q&a/Htm/CMOS%20VS%20CCD%20camera.htm (visited on 02/05/2015).

[28] F. Zhao, M.-C. Laia, and D.L. Harringtonb. *Automotive spark-ignited direct-injection gasoline engines*. Elsevier science Ltd, 1999.

8. Appendix

Appendix A, Test plan

Punkt #	Varvtal [rpm]	Insugstryck [Bar]	Tumble tunga	Tumble Flap	Injektion stryck (faktisk) [Bar]	Injektions start [CAD]	Memgate [CAD]	Kamera filmtid [ms]	Injektiostid (Matlab) [ms]	Injektionstid (faktisk) [ms]
2015-04-16-TF1	1500	0.86	Medium	Aktiverad	100	[redacted]	-316	7	1.37	1.37
2015-04-16-TF2	1500	0.86	Medium	Aktiverad	200	[redacted]	-316	7	0.94	0.94
2015-04-16-TF3	1500	0.86	Medium	Deaktiverad	200	[redacted]	-316	7	0.94	0.94
2015-04-16-TF4	1500	0.86	Medium	Deaktiverad	100	[redacted]	-316	7	1.37	1.37
2015-04-17 TF5	1750	0.98	Medium	Deaktiverad	100	[redacted]	-301	8	2.1	2.1
2015-04-17 TF6	1750	0.98	Medium	Aktiverad	100	[redacted]	-301	8	2.1	2.1
2015-04-17 TF7	1750	0.98	Medium	Deaktiverad	199	[redacted]	-301	8	1.43	1.43
2015-04-17 TF8	1750	0.98	Medium	Aktiverad	199	[redacted]	-301	8	1.43	1.43
2015-04-17 TF9	1750	0.98	High	Aktiverad	200	[redacted]	-301	8	1.43	1.43
2015-04-17 TF10	1750	0.98	High	Aktiverad	100	[redacted]	-301	8	2.1	2.1
2015-04-20 TF11	1500	0.86	High	Aktiverad	100	[redacted]	-316	7	1.37	1.37
2015-04-20 TF12	1500	0.86	High	Aktiverad	200	[redacted]	-316	7	0.94	0.94
2015-04-20-TF13	2000	1.94	High	Deaktiverad	100	[redacted]	-321	12	6.5515	6.5
2015-04-20-TF14	2000	1.94	High	Deaktiverad	200	[redacted]	-321	12	4.6315	4.6
2015-04-20-TF15	2000	1.94	High	Aktiverad	200	[redacted]	-321	12	4.6315	4.6
2015-04-20-TF16	2000	1.94	High	Aktiverad	100	[redacted]	-321	12	6.5515	6.5
2015-04-23-TF17	1750	0.98	High	Deaktiverad	350	[redacted]	-301	8	1.08	1.1
2015-04-23-TF18	1750	0.98	High	Aktiverad	350	[redacted]	-301	8	1.08	1.1
2015-04-24-TF19	1500	0.86	High	Deaktiverad	350	[redacted]	-316	8	0.71	0.71
2015-04-24-TF20	1500	0.86	High	Aktiverad	350	[redacted]	-316	8	0.71	0.71
2015-04-24-TF21	2000	2	High	Aktiverad	350	[redacted]	-321	12	3.47	3.45
2015-04-27-TF22	2000	2	High	Deaktiverad	350	[redacted]	-321	12	3.47	3.45
2015-04-27-TF23	700	0.9	High	Deaktiverad	350	[redacted]	-30	12	0.99	0.99
2015-04-27-TF24	700	0.9	High	Aktiverad	350	[redacted]	-30	12	0.99	0.99
2015-04-27-TF25	700	0.9	High	Deaktiverad	200	[redacted]	-30	12	1.315	1.32
2015-04-27-TF26	700	0.9	High	Aktiverad	200	[redacted]	-30	12	1.315	1.32
2015-04-27-TF27	700	0.9	High	Deaktiverad	100	[redacted]	-30	12	1.8619	1.86
2015-04-27-TF28	700	0.9	High	Aktiverad	100	[redacted]	-30	12	1.8619	1.86
2015-04-27-TF29	1450	0.85	High	Deaktiverad	350	[redacted]	-296 + 14	12+4		0,73 + 0,30
2015-04-27-TF30	1450	0.85	High	Aktiverad	350	[redacted]	-296 + 14	10+4		0,73 + 0,30
2015-04-28-TF31	1450	0.85	High	Deaktiverad	200	[redacted]	-296 + 14	10+4	1.0677	1.07 + 0,25
2015-04-28-TF32	1450	0.85	High	Aktiverad	200	[redacted]	-296 + 14	10+4	1.0677	1.07 + 0,25
2015-04-28-TF33	1450	0.85	High	Aktiverad	100	[redacted]	-296 + 14	10+4	1.0677	1.6279 + 0,21

2015-04-28-TF34	1450	0.85	Low	Deaktiverad	100	[redacted]	-296 + 14	10+4	1.0677	1.6279 + 0,21
2015-04-28-TF35	1750	0.98	Low	Aktiverad	350	[redacted]	-301	8	1.08	1.1
2015-04-28-TF36	1750	0.98	Low	Aktiverad	200	[redacted]	-301	8	1.43	1.43
2015-04-28-TF37	1750	0.98	Low	Aktiverad	100	[redacted]	-301	8	2.1	2.1
						[redacted]				
2015-04-29-TF38	1500	0.86	High	Deaktiverad	100	[redacted]	-26	8	1.37	1.35
2015-04-30-TF39	1500	0.86	High	Deaktiverad	100	[redacted]	-26	8	1.37	1.35
2015-04-30-TF40	1500	0.86	High	Deaktiverad	100	[redacted]	-26	8	1.37	1.6
2015-04-30-TF41	1500	0.52	High	Deaktiverad	100	[redacted]	-26	8	0.81	0.8
2015-04-30-TF42	1500	0.5	High	Deaktiverad	100	[redacted]	-26	8	0.7273	0.73
Ej filmad 2015-04-30-TF43	1500	0.42	High	Deaktiverad	100	[redacted]	-28	8	0.7273	0.73
2015-05-04-TF43	1500	0.52	High	Deaktiverad	100	[redacted]	-26	8	0.7273	1.05
2015-05-04-TF44	1500	0.52	Low	Deaktiverad	100	[redacted]	-28	8	0.7273	1.05
2015-05-04-TF45	1500	0.52	Low	Deaktiverad	100	[redacted]	-28	8	0.7273	1.05
2015-05-04-TF46	1500	0.52	Low	Deaktiverad	100	[redacted]	-28	8	0.7273	1.07
2015-05-04-TF47	1500	0.58	Low	Deaktiverad	100	[redacted]	-28	8	0.7273	1.17
2015-05-05-TF48	1500	0.58	Low	Deaktiverad	100	[redacted]	-28	8	0.7273	1.17
2015-05-05-TF49	1500	0.58	Low	Aktiverad	100	[redacted]	-28	8	0.7273	1.17
2015-05-05-TF50	1500	0.58	Low	Deaktiverad	200	[redacted]	-28	8	0.83	0.83
2015-05-05-TF51	1500	0.58	Low	Aktiverad	200	[redacted]	-28	8	0.83	0.83
2015-05-05-TF52	1500	0.58	Low	Deaktiverad	350	[redacted]	-28	8	0.62	0.65
2015-05-05-TF53	1500	0.58	Low	Aktiverad	350	[redacted]	-28	8	0.62	0.65
2015-05-06-TF54	1500	0.58	Medium	Deaktiverad	350	[redacted]	-28	8	0.62	0.65
2015-05-06-TF55	1500	0.57	Medium	Aktiverad	350	[redacted]	-28	8	0.62	0.65
2015-05-06-TF56	1500	0.58	Medium	Aktiverad	200	[redacted]	-28	8	0.83	0.83
2015-05-06-TF57	1500	0.58	Medium	Aktiverad	100	[redacted]	-28	8	0.7273	1.17
2015-05-06-TF58	1500	0.6	High	Deaktiverad	100	[redacted]	-28	8	0.7273	1.17
2015-05-06-TF59	1500	0.59	High	Aktiverad	200	[redacted]	-28	8	0.83	0.83
2015-05-06-TF60	1500	0.59	High	Aktiverad	350	[redacted]	-28	8	0.62	0.65
2015-05-07-TF61	1500	0.59	High	Aktiverad	200	[redacted]	-13	8	0.83	0.83
2015-05-08-TF62	1500	0.59	Medium	Aktiverad	200	[redacted]	-13	8	0.83	0.83
Ej filmad 2015-05-08-TF64	1500	0.59	Medium	Aktiverad	200	[redacted]	-13	8	0.83	0.83
2015-05-08-TF65	1500	0.6	High	Aktiverad	100	[redacted]	-28	8	0.7273	1.17
2015-05-08-TF66	1500	0.6	Low	Aktiverad	200	[redacted]	-15	8	0.83	0.83
2015-05-08-TF67	1500	0.6	Low	Aktiverad	200	[redacted]	-15	8	0.83	0.83

Insugstemperatur [Celsius]	Oljetemperatur [Celsius]	Kylvattentemperatur [Celsius]	Luftmassflöde [kg/h]	Tändning föreslagen [CAD]	Tändning faktisk [CAD]	Uppladdningstid tändspole [ms]	Antal arbetscykler	Antal vilande cykler	Bränslemängd (Förslag Bränslemängd)/ cykel [mg]	EVO	EVC	IVO	IVC
20	20.3	86.5	18.8	[redacted]	[redacted]	-	60	200	19.4	159	12	344	212
21.3	24	76.4	18.8	[redacted]	[redacted]	-	60	200	19.4	159	12	344	212
22.1	25.2	74.5	19.3	[redacted]	[redacted]	-	60	200	19.4	159	12	344	212
23.8	22.4	75.9	19.3	[redacted]	[redacted]	-	60	200	19.4	159	12	344	212
19.1	24.8	89	27.9	[redacted]	[redacted]	-	60	200	28.4	159	12	344	212
19	25.1	86.6	39.5	[redacted]	[redacted]	-	60	200	28.4	159	12	344	212
21	23.3	77.3	34	[redacted]	[redacted]	-	60	200	28.4	159	12	344	212
20.4	23.5	84.1	39.5	[redacted]	[redacted]	-	60	200	28.4	159	12	344	212
20.3	24.1	81.9	32	[redacted]	[redacted]	-	60	200	28.4	159	12	344	212
20.3	25.1	88.7	33	[redacted]	[redacted]	-	60	200	28.4	159	12	344	212
21.8	21.9	81.3	18.6	[redacted]	[redacted]	-	60	200	19.4	159	12	344	212
22	22.4	86.6	18.6	[redacted]	[redacted]	-	60	200	19.4	159	12	344	212
23.4	23.3	85.9	Övertryck	[redacted]	[redacted]	-	60	200		159	12	322	192
23.5	24	88.1	Övertryck	[redacted]	[redacted]	-	60	200		159	12	322	192
24.5	24	78	Övertryck	[redacted]	[redacted]	-	60	200		159	12	322	192
24.3	24	82.5	Övertryck	[redacted]	[redacted]	-	60	200		159	12	322	192
23.6	51.8	77.6		[redacted]	[redacted]	-	60	200	28.4	159	12	344	212
23.4	58.5	85.3		[redacted]	[redacted]	-	60	200	28.4	159	12	344	212
24.7	53.1	81.1		[redacted]	[redacted]	-	60	200	19.4	159	12	344	212
27.7	57.3	85.5		[redacted]	[redacted]	-	60	200	19.4	159	12	344	212
21.4	60	85.6	Övertryck	[redacted]	[redacted]	-	60	200		159	12	322	192
21	55.9	84.3	Övertryck	[redacted]	[redacted]	-	60	200		159	12	322	192
22	59.3	86	8.1	[redacted]	[redacted]	-	60	200	25.8	159	12	*	*
21.9	61.9	89.5	8.1	[redacted]	[redacted]	-	60	200	25.8	159	12	*	*
			8.1	[redacted]	[redacted]	-	60	200	25.8	159	12	*	*
22.6	58.8	84.7	8.1	[redacted]	[redacted]	-	60	200	25.8	159	12	*	*
22	62.2	87.5	8.1	[redacted]	[redacted]	-	60	200	25.8	159	12	*	*
21.9	54.2	79.4	8.1	[redacted]	[redacted]	-	60	200	25.8	159	12	*	*
23.5	66.6	88.8	18.3	[redacted]	[redacted]	-	60	200		159	12	*	*
23.5	67.6	87.7	18.3	[redacted]	[redacted]	-	60	200		159	12	*	*
23.9	53	80	18.3	[redacted]	[redacted]	-	60	200		159	12	*	*
22.3	51	79	18.3	[redacted]	[redacted]	-	60	200		159	12	*	*
22	60.5	89.3	17	[redacted]	[redacted]	-	60	200		159	12	*	*

21.5	55.7	81.9	16.5	[redacted]	-		60	200		159	12	*	*
20.6	62.2	88.7	43	[redacted]	-		60	200	28.4	159	12	344	212
22.2	52.2	76.6	35	[redacted]	-		60	200	28.4	159	12	344	212
21.7	57.4	83.1	30	[redacted]	-		60	200	28.4	159	12	344	212
				[redacted]									
23.8	47.3	70.6	18.6	[redacted]		5	50	200	19.4	159	12	344	212
19.3	57.8	86	19.5	[redacted]		5	50	200	19.4	141	-6	322	192
21.1	61.8	85.6	19	[redacted]		5	50	200	19.4	141	-6	322	192
21.1	61.8	85.6	8.3	[redacted]		5	50	200	19.4	141	-6	322	192
24.9	62.3	89.2	7.8	[redacted]		5	50	200	19.4	141	-6	322	192
24.9	62.3	89.2	8.2	[redacted]		5	50	200	19.4	141	-6	344	212
18.8	58.8	88.4	8,2 (7,8)	[redacted]		5	50	200	19.4	141	-6	344	212
24	57.9	83.3	8.1	[redacted]		5	50	200	19.4	141	-6	344	212
25.4	53.8	77.3	8.1	[redacted]		5	50	200	19.4	141	-6	344	212
25.2	51.9	75.3	8.1	[redacted]		5	50	200	19.4	141	-6	344	212
25.2	51.9	75.3	9.5	[redacted]		5	100	300	19.4	141	-6	344	212
22.2	53.3	80.8	9.5	[redacted]		5	100	300	19.4	141	-6	344	212
22.5	60	88.4	9.8	[redacted]		5	100	300	19.4	141	-6	344	212
23.1	59.1	86.3	9.7	[redacted]		5	100	300	19.4	141	-6	344	212
24.5	57.9	83	9.7	[redacted]		5	100	300	19.4	141	-6	344	212
22.9	53.9	75.9	10	[redacted]		5	100	300	19.4	141	-6	344	212
23.1	56.9	77.8	9.8	[redacted]		5	100	300	19.4	141	-6	344	212
22.1	54.5	80	10.4	[redacted]		5	100	300	19.4	141	-6	344	212
22.4	62.1	88	9.7	[redacted]		5	100	300	19.4	141	-6	344	212
21.6	64.6	88.2	9.6	[redacted]		5	100	300	19.4	141	-6	344	212
24	57.9	82.6	9.7	[redacted]		5	100	300	19.4	141	-6	344	212
23.7	58.5	83.5	9.7	[redacted]		5	100	300	19.4	141	-6	344	212
25	62	85.6	9.6	[redacted]		5	100	300	19.4	141	-6	344	212
23.8	65.1	87.8	9.5	[redacted]		5	100	300	19.4	141	-6	344	212
22.3	62	85.4	9.6	[redacted]		5	100	300	19.4	141	-6	344	212
22.9	58	86.8	9.7	[redacted]		5	100	300	19.4	141	-6	344	212
21.8	60.9	88.1	9.5	[redacted]		5	100	300	19.4	141	-6	344	212
23.4	59.8	86	9.7	[redacted]		5	100	300	19.4	141	-6	344	212
23.5	61	85.3	9.7	[redacted]		5	100	300	19.4	141	-6	344	212
23.1	62.7	84.8	9.7	[redacted]		5	100	300	19.4	141	-6	344	212