Experimental Investigation of Impinging Diesel Sprays for HCCI Combustion

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Doctoral thesis
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Academic thesis, which with the approval of Kungliga Tekniska Högskolan, will be presented for public review in fulfilment of the requirements for a Doctorate of Engineering in Machine Design. The public review is held at Kungliga Tekniska Högskolan, Lindstedtsvägen 26 in room F3 at 14.00 on the 21st of February 2007.
ABSTRACT

Engine research and development is to a large extent driven by the quest of lowering exhaust emissions and fuel consumption. The combination of low emissions and low fuel consumption is not the simultaneous characteristic of the world’s primary engine concepts, the diesel and the spark-ignited (SI) engine. However, such a concept do exist, it is commonly called Homogeneous Charge Compression Ignition (HCCI).

The HCCI combustion concept is when a premixed air and fuel mixture is ignited by the heat of compression. The operation is unthrottled, like the diesel engine, which is advantageous for its efficiency. The premixed air / fuel mixture preclude soot formation, like the SI engine. An exclusive feature of HCCI combustion is extremely low NOX production due to low-temperature combustion.

The mixture preparation of the typical gasoline HCCI engine is similar to the SI engine, via port-injection, which results in a well homogenized mixture. Port injection of diesel fuel is however very difficult since the environment is too cold for the fuel to vaporise. A better alternative is therefore direct-injection. However, injection must occur in a way where a homogeneous mixture is formed, while contact of the liquid fuel with cold walls is avoided.

There are many approaches to direct-injected mixture formation. This thesis focuses on exploring the concept of impinging sprays; its characteristics and its impact on combustion and emissions. The work comprises unique information regarding impinging sprays, as well as results regarding engine performance. It is concluded that impinging sprays are well suited for early direct-injection.
ACKNOWLEDGEMENTS

I acknowledge my dear wife Helena and our kids Anya and Joel for all the love and joy. Thanks to my mother, father, brothers and family, I love you all. Thanks to all my friends, who make my life interesting and fun. Thanks to Hans-Erik Ångström for providing the possibility to do this work, and for all the powerful lab support. Ernst Winklhofer for your ability to put things into perspective and keep the focus. Andreas Cronhjort for invaluable numerical and experimental contributions. Lars Dahlén for the enthusiasm and energy. Greger Juhlin for the support and for employing me. Per Strålin as my buddy and for the special spray-humor. Per Risberg for the conversations and friendship. Eric Lycke, Tommy Tillman and Henrik Nilsson for their outstanding work in the lab. Uffe, Kurre and Lelle for their job in the work shop. Big thanks to all the people at Scania and KTH who has been a support to me.
LIST OF PAPERS

The work is expressed in papers which were written by the author and co-authors. The papers are appended in the end of the thesis. A summary of the papers and the contribution of each author are found in the section “Work progress”.

“Fuel Sprays for Premixed Compression Ignited Combustion - Characteristics of Impinging Sprays”, Fredrik Wåhlin and Andreas Cronhjort

“Segmentation Algorithm for Diesel Spray Image Analysis”, Andreas Cronhjort and Fredrik Wåhlin.


Paper IV. (Accepted for publication at Atomization and Sprays, 2007).
“Impinging Diesel Sprays”, Fredrik Wåhlin and Andreas Cronhjort

Paper V. To be submitted.
“HCCI Combustion with Impinging Diesel Sprays”, Fredrik Wåhlin.
“No success in your career can compensate for a failure in your private life”

Stephen R. Covey

“We must become the change we seek in the world”

Gandhi

“Self-Knowledge is best learned, not by contemplation, but by action. Strive to do your duty and you will soon discover of what stuff you are made.”

Johann Goethe

“You have lots of answers. But what is the question?”

Ernst Winklhofer
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BACKGROUND

The following section covers some background on why it is important to control emissions from vehicles.

The global energy situation

The total global energy production increased by 40% during 1980–2002. In 2002, petroleum (crude oil and natural gas plant liquids) continued to be the world’s most important primary energy source, accounting for 37.9% of world primary energy production [1]. Between 1992 and 2002, petroleum production increased by 13.3%. Coal ranked second as a primary energy source in 2002, accounting for 24.1% (with 6.5% increase since 1992), followed by dry natural gas at 23.5% (with a 23.3% increase since 1992).

Figure 1. World energy primary production trends [1].

Nuclear, hydro, and other (geothermal, solar, wind, wood and waste) electric power generation ranked fourth, fifth, and sixth, as primary energy sources in 2002, accounting for 6.63, 6.56, and 0.8% of world primary energy production, respectively. The global energy production is by over 90% satisfied by limited
natural resources, and the increasing demand is mainly covered with fossil fuels. This scenario raises the question of how long the supplies will last. Whatever answer one chooses to believe in, most will agree that finding new energy sources and economizing the ones we have is very important in the long perspective.

**The transport sector’s energy demand**

The transport sector accounts for about 25% of the total commercial energy consumed worldwide, and consume approximately one-half of total oil produced, and its proportion is increasing. The forecast is a considerable increasing demand of transport services, as economic growth occurs in developing countries, incomes rise, the trend toward urbanization continues and as the process of globalization moves forward with expected increases in world trade. Up until 2020, demand is predicted to grow by 3.6% / year in developing countries and by 1.5% / year in industrialized countries [2]. *The transport sector’s current trend demands improvement of its energy efficiency and development of new means and technologies that will reduce the oil dependency* [3].

**Emissions from ground transportation**

The motorized transportation of today emits greenhouse gases, particulate matter, nitrogen oxides, sulphur oxides and volatile organic compounds all of which have negative impacts at local and often at regional levels [2]. However, exactly what kinds of pollutants that are produced in what proportions varies significantly based on a number of factors, including engine type, fuel used and driving conditions.

To deal with pollution from ground transportation, the legislated emission levels have tightened over the years, and will so be continued. Table 1 shows the existing and forthcoming legislated emission levels for heavy-duty trucks and buses in Europe (Euro I –V) and the U.S. (EPA 07 - 10). The Euro VI is not set yet, but is expected to be a significant reduction in oxides of nitrogen (NO\textsubscript{X}) and particulates compared to Euro V.
Table 1. Current and forthcoming legislated emission levels for Heavy-Duty Truck and Bus Engines [9].

<table>
<thead>
<tr>
<th>Legislation</th>
<th>Year</th>
<th>NOx [g/kWh]</th>
<th>Particulates [g/kWh]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Euro I</td>
<td>1992</td>
<td>8</td>
<td>0.25</td>
</tr>
<tr>
<td>Euro II</td>
<td>1996</td>
<td>7</td>
<td>0.15</td>
</tr>
<tr>
<td>Euro III</td>
<td>2000</td>
<td>5</td>
<td>0.1</td>
</tr>
<tr>
<td>Euro IV</td>
<td>2005</td>
<td>3.5</td>
<td>0.02</td>
</tr>
<tr>
<td>Euro V</td>
<td>2008</td>
<td>2</td>
<td>0.02</td>
</tr>
<tr>
<td>EPA 07-10, US</td>
<td>2007-2010</td>
<td>0.27</td>
<td>0.013</td>
</tr>
</tbody>
</table>

Greenhouse gases

Statements are made that the global warming effect observed the latest 50 years only can be explained if human emissions of greenhouse gases are taken into account [4]. If no proceedings are taken to limit the emissions of greenhouse gases, the earth’s climate is expected to change with an unseen velocity, accompanying serious consequences:

- The CO₂ concentration in the atmosphere will increase by 50 % – 160 % until 2100, depending on what measures are taken.
- The number of extreme weather phenomena will increase; rainstorms, flooding, dry periods, fires etc.
- Lack of fresh water will increase in certain regions.
- Extinction of species and precious ecosystems: coral reefs, glaciers, mangrove forests, polar and alpine regions.

Greenhouse gases, unlike many local air pollutants, are considered stock pollutants. A stock air pollutant is one that has a long lifetime in the atmosphere, and therefore can accumulate over time. Stock air pollutants are also generally well mixed in the atmosphere. As a consequence of this mixing, the impact a greenhouse gas has on the atmosphere is mostly independent of where it was emitted. These characteristics of greenhouse gases imply that they should be addressed on a global (i.e., international) scale. Globally, transport accounts for about 21 % of CO₂ emissions. To lower this number, more fuel efficient vehicles and alternative fuels are required.
Particulate matter (PM)
Particulate matter is perhaps the most critical transport-sector pollutant for developing countries in the early part of the twenty-first century. Its effects on human health are significant while technical mechanisms to control particulate matter are costly. Current regulations of PM are by mass, although there is increasing evidence that smaller particles cause more damage to human health than large particles [5]. Also, there is little attention paid to the actual chemical composition (including the proportion of soot, sulphates and polycyclic aromatic hydrocarbons [PAH]) of PM. However, speculations are that the extent of health side effects depends on this composition.

Nitrogen Oxides (NO\textsubscript{X})
NO\textsubscript{X} are of concern both because of their direct effects on human health, and because they react in the atmosphere with volatile organic compounds (VOCs) to produce photochemical smog. Photochemical smog comprises mixtures of PM and noxious gases and is formed when NO\textsubscript{X} and VOCs react in the presence of heat and sunlight [6]. Nitric oxide (NO) and nitrogen dioxide (NO\textsubscript{2}) are released in combustion because the molecular nitrogen (N\textsubscript{2}) in the air/fuel mixture splits and is oxidized. The higher the flame temperature or longer the residence time, the more NO\textsubscript{X} production; consequently, the same technical interventions in engine calibration that might reduce VOCs will increase NO\textsubscript{X}. NO\textsubscript{X} is also a contributor to acid rain and the global warming. In addition, it is toxic and can cause impairing respiratory function and damage to lung tissue [7].

Volatile organic compounds (VOCs)
The term volatile organic compounds refer to a range of non-methane hydrocarbons (NMHCs) which evaporate at normal ambient temperatures. NMHCs are released during combustion due to incomplete combustion of the fuel. VOCs are usually regulated as a class because of their contribution to ground-level ozone formation through a photochemical smog process. Ozone is hazardous to human health as well as to plants [8]. It seems to impair respiratory function as a short-run response to exposure, but the longer-term effects are less clear. The production of ground-level ozone occurs through reactions in sunlight of VOCs and NO\textsubscript{X}. VOCs also contribute to particulate formation by coagulating onto soot. In addition, some VOCs are themselves hazardous to human health; they include benzene, polycyclic aromatic
hydrocarbons, 1,3-butadiene, aldehydes and, through groundwater seepage, methyl tertiary butyl ether (MTBE).

**Carbon monoxide (CO)**
CO emissions are often highly correlated with hydrocarbon (HC) emissions. In the human body, CO can cause oxygen deprivation (hypoxia), displacing oxygen in bonding with hemoglobin, causing cardiovascular and coronary problems, increasing risk of stroke, and impairing learning ability, dexterity and sleep [2]. CO is mostly hazardous in relatively confined areas such as tunnels under bridges and overpasses, and in dense urban settings. In unconfined areas or away from population centres, it will stabilize into CO₂ before damage to human health is likely.

**Sulphur Oxides (SOₓ)**
Sulphur present in fuel will be released either as sulphates (SO₄²⁻) (which can be an important component of PM) or sulphur dioxide (SO₂). SO₂ is a major health concern because of its effects on bronchial function, but in metropolitan regions with high concentrations of ambient SO₂, the contribution of the transport sector tends to be secondary to that of manufacturing and/or electricity production. For this reason, concern about sulphur in transport fuels tends to be driven more out of concern about particulates rather than SO₂. Fuels for ground transportation will be regulated to very low sulphur concentrations; to protect aftertreatment equipment and lower the particulate emissions.
INTRODUCTION

This chapter provides some introduction to diesel, SI and HCCI combustion concepts and their advantages and disadvantages.

Power plants for ground transportation

The two engine types that dominate today’s motorized ground transportation are the diesel engine and the SI engine. The diesel engine offers superior fuel consumption compared to the SI engine. The diesel engine ranges in sizes from powering the largest ocean-crossing super tanker to the smallest private car, and is the predominant power plant for heavy-duty transportation where fuel efficiency is a high priority. Unfortunately, the diesel engine emits high levels of nitric oxides (NOX) and particulate matter (PM) compared to the catalyst-equipped SI engine, see Figure 2. To cope with future emissions legislation, exhaust aftertreatment devices such as diesel particulate filters (DPF), lean NOX traps (LNT) and urea-based selective catalyst reduction (SCR), will most probably be required.

Figure 2. Severe smoke emissions from a diesel engine. In this case the smoke emissions are unacceptable; smoke emissions from a modern diesel engine are invisible.

The catalyst-equipped SI engine is popular for private cars and has very low emissions, but it is not problem-free. Its emissions are severe under cold-start
conditions, and its low efficiency (high fuel consumption) makes it a large producer of CO₂.

Engine manufacturers are forced to mobilize considerable effort and resources to find a solution that copes with the increasingly stringent emission levels. There are several possible technical paths this may take. However, if an engine manufacturer were able to achieve the legislated emission levels without exhaust aftertreatment, then they may have an advantage due to the following [10]:

- lower cost system
- improved engine and component durability
- reduced packaging concerns
- enhanced image as a technology leader
- product distinction in a crowded marketplace

A combustion concept with the potential to fulfil on this is HCCI. HCCI combustion could roughly be described as a hybrid with the best features from the diesel engine and the SI engine. Its NOₓ and smoke emissions are very low, while its efficiency is high, comparable with the diesel engine.

**Engine combustion and emissions formation**

This section consists of a brief description of combustion and emissions formation of SI, diesel and HCCI. The SI and diesel sections originate from ref. [11] if not otherwise stated.

**The SI engine**

The SI engine relies on a stoichiometric fuel/air mixture and a sparkplug for its ignition. Typically, the compression ratio lies around 10:1. Modern SI engines are usually port-injected; the fuel is injected through a low pressure system into the intake manifold. Its low boiling range (25 - 210 °C) [12] makes most of the gasoline to evaporate and mix with the air. The fuel/air mixture is then inhaled via the intake port into the cylinder during the intake stroke and compressed during the compression stroke. The fuel and air continues to mix throughout this process and at the end of the compression stroke, the mixture is ignited by a spark. A close-to-stoichiometric condition is required for two reasons: To enable for spark ignition and for the performance of the three-way catalyst. A flame kernel is initiated by the spark well before top dead centre (TDC). The mixture motion and composition
around the spark plug at time of spark discharge is decisive for the flame
development and subsequent flame propagation. The small flame kernel initiated by
the spark grows as the turbulent flame front propagates through the combustion
chamber, see Figure 3.

![Figure 3. Spark ignited combustion images [13]. Each flame image captured from consecutive cycle. The flame can be described as premixed turbulent.](image)

The temperature in the reaction front is well above the NO\textsubscript{X} formation limit. The
burned gases in the centre of the combustion chamber are compressed to the highest
temperature attained in the cylinder as the rest of the charge burns, causing
considerable NO\textsubscript{X} formation. The absence of luminous yellow flames indicates a
well pre-mixed mixture and hence no soot formation. The unburned mixture ahead
of the propagating flame front is also compressed, and some part of it enters crevices
around the top-land region and leaks past the piston ring into the crank case as blow-
by gases. As the flame front is extinguished at the cold combustion chamber walls,
the mixture in the crevices remains unburned. Some of the mixture in the crevices
flows back into the combustion chamber as the pressure drops in the expansion
stroke. The lower temperature in the expansion stroke will not oxidize all the fuel
and some of it will remain as HC and CO emissions. The unhealthy mixture of CO\textsubscript{2},
NO\textsubscript{X}, HC and CO in the cylinder is now dispatched into the exhaust system where
most of it is converted to N\textsubscript{2}, CO\textsubscript{2} and H\textsubscript{2}O by the three-way catalyst. However,
during cold start conditions the catalyst temperature is lower than its light-off
temperature (~300 °C), and the pollutants are released into the atmosphere. The catalyst temperature needs to be ~400 °C or more for a high conversion rate [12].

The efficiency of the SI engine is limited mainly for two reasons. Firstly, since it requires a stoichiometric charge, it is necessary to throttle the intake for lowering the load, which means losses and deteriorating efficiency. Secondly, to avoid knocking (which is actually HCCI combustion), a low compression ratio is required which lowers the overall efficiency. The reason to avoid HCCI in this case is because the reaction rate is much too fast at stoichiometric conditions. The rapid heat release causes violent oscillations that can destroy the combustion chamber. Knocking combustion also stirs up the thermal boundary layer and exposes the piston to higher temperatures, which can result in piston melting.

The diesel engine

In the modern direct-injected diesel engine, fuel is injected into compressed gas at TDC via a high pressure injection system. The compression ratio in the diesel engine is high, usually around 18:1, resulting in a high in-cylinder gas temperature and density at TDC, approximately in the range of 1000 K and 30 kg/m³ for a turbocharged engine at high load. The highly pressurized fuel is introduced into the combustion chamber via 5-8 fuel sprays, depending on the size of the cylinder. The fuel sprays break up, atomize, evaporate and mix with the high density high-temperature gas, which creates a combustible mixture. After an ignition delay of a few crank angle degrees (CAD), fuel which has mixed with air ignites, creating a rapid heat release. The combustion after the pre-mixed part is mixing controlled; the rate of heat release is controlled by how fast the fuel will atomize, evaporate and mix with air thus creating a combustible mixture. This mixing-controlled combustion can be described as a turbulent unsteady diffusion flame. The yellow luminosity of the flames originates from hot radiating soot particles, see Figure 4. Since the combustion is mixing controlled there is no knocking limit, hence the higher compression ratio and subsequent higher efficiency. To ensure a complete combustion, the diesel engine always operates in a globally lean condition. This enables unthrottled operation which is also good for the engine efficiency. However using a three-way catalyst is impossible. In-cylinder air swirl helps the transport of oxygen to the combustion zone and speeds up the heat release rate. After end of injection, the air swirl is mainly responsible for the continuing mixing between fuel and oxygen, thus promoting a complete combustion. Various techniques can be used
to lower NOX and PM emissions from the diesel engine. Unfortunately, lowering one usually results in raising the other.

Figure 4. Diesel combustion images on courtesy of Andreas Cronhjort. A crankangle-resolved sequence of 8 burning diesel sprays. The yellow luminosity of the flames originates from hot radiating soot particles.

Figure 5 shows a schematic of the conceptual model of direct-injected (DI) diesel combustion by Dec [14], during the mixing-controlled burn, prior to end of injection.

Figure 5. A schematic showing the conceptual model by Dec of DI diesel combustion during the mixing-controlled burn [14].
A short distance downstream of the injector tip, the fuel vapour and entrained air has formed a relatively uniform mixture. Then soot appears as small particles across the entire cross-section of the jet. The soot formation and particle growth continue as the soot moves down to the head vortex. The soot particles accumulate in the recirculating head vortex where they have time to grow to a larger size. Some of the soot particles reach the diffusion flame at the periphery of the jet where they can be oxidized by OH radicals. When the exhaust port opens, most of the particles are already oxidized.

The initial soot formation is most likely formed in a standing fuel-rich premixed flame where local air/fuel ratio typically is around 0.25. Adiabatic flame temperatures are typically ~ 1600 K which is below the thermal NO\textsubscript{X} formation limit. The presence of this standing premixed flame throughout the mixing controlled burn would mean that all the fuel first undergoes fuel-rich premixed combustion and later diffusion-flame combustion, rather than being a more classical pure fuel/air diffusion flame. The diffusion-flame combustion occurs at the jet periphery. The locally stoichiometric condition results in a high flame temperature, while excessive oxygen is available. This is a favourable condition for thermal NO formation. Soot oxidation occurs via OH radicals attack at the diffusion flame. OH radical attack is thought to be the primary method of soot oxidation [14].

**The HCCI engine**

It is said that a loved child has many names, and the HCCI-infant is no exception. Johansson [15] has summarized many of the invented acronyms for this type of combustion and associated author, institute and year, see Table 2.
Table 2. HCCI Acronyms (from [15]).

<table>
<thead>
<tr>
<th>Acronym</th>
<th>Meaning</th>
<th>Author</th>
<th>Location</th>
<th>Year</th>
</tr>
</thead>
<tbody>
<tr>
<td>ATAC</td>
<td>Active Thermo-Athmosphere Combustion</td>
<td>Onishi</td>
<td>Nippon clean engine research institute</td>
<td>1979</td>
</tr>
<tr>
<td>TS</td>
<td>Toyota-Soken combustion</td>
<td>Nogushi</td>
<td>Toyota/Soken</td>
<td>1979</td>
</tr>
<tr>
<td>CIHC</td>
<td>Compression-Ignited Homogeneous Charge</td>
<td>Najt</td>
<td>Univ. Wisconsin-Madison</td>
<td>1983</td>
</tr>
<tr>
<td>HCCI</td>
<td>Homogeneous Charge Compression Ignition</td>
<td>Thring</td>
<td>SwRI</td>
<td>1989</td>
</tr>
<tr>
<td></td>
<td>Selbstzündning</td>
<td>Stockinger</td>
<td>Univ. Hamburg</td>
<td>1992</td>
</tr>
<tr>
<td>AR, ARC</td>
<td>Active Radical Combustion</td>
<td>Ishibashi</td>
<td>Honda</td>
<td>1996</td>
</tr>
<tr>
<td>FDCCP</td>
<td>Fluid Dynamically Controlled Combustion Process</td>
<td>Duret</td>
<td>IFP Institute Français du Pétrole</td>
<td>1996</td>
</tr>
<tr>
<td>PCCI</td>
<td>Premixed Charge Compression Ignition</td>
<td>Aoyama</td>
<td>Toyota</td>
<td>1996</td>
</tr>
<tr>
<td>MK, M-fire</td>
<td>Modulated Kinetics</td>
<td>Kawashima</td>
<td>Nissan</td>
<td>1997</td>
</tr>
<tr>
<td>PREDIC</td>
<td>Premixed Diesel Combustion</td>
<td>Tsujimura et al.</td>
<td>New ACE</td>
<td>1996</td>
</tr>
<tr>
<td>MULDIC</td>
<td>Multiple Stage Diesel Combustion</td>
<td>Tsujimura et al.</td>
<td>New ACE</td>
<td>1998</td>
</tr>
<tr>
<td>HIMICS</td>
<td>Homogeneous Charge intelligent Multiple Injection Combustion System</td>
<td>Yokota et al.</td>
<td>Hino Motors</td>
<td>1997</td>
</tr>
<tr>
<td>HCDC</td>
<td>Homogeneous Charge Diesel Combustion</td>
<td>Suziki et al.</td>
<td>Traffic Safety and Nuisance Research Institute</td>
<td>1997</td>
</tr>
<tr>
<td>UNIBUS</td>
<td>Uniform Bulky Combustion System</td>
<td>Yanagihara</td>
<td>Toyota</td>
<td>1997</td>
</tr>
<tr>
<td>HC</td>
<td>Homogeneous Combustion</td>
<td>Willand</td>
<td>Daimler-Benz</td>
<td>1998</td>
</tr>
<tr>
<td>SPAC</td>
<td>Space Combustion</td>
<td>?</td>
<td>Daimler-Benz</td>
<td>1998</td>
</tr>
<tr>
<td>RZV</td>
<td>Raumverzündung</td>
<td>?</td>
<td>Daimler-Benz</td>
<td>1998</td>
</tr>
<tr>
<td>CAI</td>
<td>Controlled Autoignition</td>
<td>?</td>
<td>IFP</td>
<td>1998</td>
</tr>
<tr>
<td>PCI</td>
<td>Premixed Compression-Ignited combustion</td>
<td>Iwabushi</td>
<td>Mitsubishi</td>
<td>1999</td>
</tr>
</tbody>
</table>

Although it exist differences between the concepts listed in Table 2, what they all have in common is that they rely on a pre-mixed charge, which is ignited with the heat of compression. A diluted mixture (usually recirculated exhaust gases (EGR) and/or air) is used in the HCCI engine to suppress the otherwise too rapid heat release (knocking) and to decrease or eliminate NOX formation. Since the HCCI engine operates in diluted mode it enables for unthrottled operation (as the diesel engine), which eliminates the gas-exchange losses associated with gasoline engine. Moreover, a high compression ratio, low heat transfer rate and rapid heat release provides the HCCI engine with high diesel-like efficiency. Major problems and disadvantages with the HCCI engine are:

- Controlling the combustion phasing.
- High peak pressures and rapid heat releases limits the practical power density of the engine.
High amounts of HC and CO emissions due to “cold” combustion.

In addition, HCCI with diesel fuel has two important problems: the low resistance to autoignition and the preparation of the in-cylinder air/fuel mixture. These problems are discussed under the section **HCCI Fuel Injection Concepts**.

**The HCCI combustion process**

As its name indicates, Homogeneous Charge Compression Ignition relies on the compression ignited combustion of a “homogeneous” charge. The term “homogeneous” is however somewhat misleading since there are always heterogeneities in the mixture, especially in direct injected HCCI but also when using a customary port injection system. Measurements using fuel planar laser induced fluorescence (PLIF) reveals inhomogeneities with a scale of roughly 4 to 6 mm using port-injection, however there is no significant difference on the combustion process to a fully homogeneous mixture [16].

The HCCI combustion process is fundamentally different from the SI and diesel combustion process. The ignition occurs at seemingly arbitrary points simultaneously in the combustion chamber [17], [18], see Figure 6. The heat release in the HCCI engine occurs throughout the bulk of the charge, as can be seen in a conceptual model proposed by Onishi [19] in 1979, see Figure 7.

*Figure 6. HCCI combustion images [13]. Ignition occurs at several points simultaneously. (On courtesy of E. Winkhofer).*
A conceptual model of HCCI combustion was proposed by Hultqvist [20] to support their experimental observations. If a parcel of gas was traced in the combustion chamber, an integral function over time is defined, that reaches a critical value when ignition occurs. The function accounts for histories of temperature, pressure, air/fuel equivalence ratio ($\lambda$), internal and external recycled exhaust gases (EGR), and reactive residuals. However, for port-injected HCCI: Pressure, air/fuel equivalence ratio, internal and external EGR are close to homogeneous and reactive radicals are not likely to survive the gas exchange process. What remains is the temperature distribution that is inhomogeneous due to heat transfer through piston and cylinder walls. In the model, ignition occurs locally where the critical temperature is reached. This critical temperature can be reached at multiple points simultaneously, as indicated by the model and also by the flame images in Figure 6.
However for direct-injected HCCI combustion, the $\lambda$ distribution is not considered homogeneous. In this case, zones with a closer to stoichiometric mixture will contribute to an earlier ignition. This mechanism is directly related to the combustion control possibilities associated with variable injection timing, since the injection timing defines the $\lambda$ stratification at start of combustion (SOC).

**HCCI commercialized**

There are three commercial vehicle engines that use some form of premixed combustion during a portion of their operating range. A light truck engine by Nissan that uses MK-combustion with diesel fuel at light load, a 2-stroke motorcycle gasoline engine by Honda partially running in HCCI-mode, and a diesel engine by Toyota which is working with a combination of early and late injection timings at low load and speed.

**Nissan MK-combustion**

The Nissan MK-combustion system (also called M-fire) [21] is based on a standard diesel engine. The engine was brought into production in 1998 for the Japanese market. The engine runs in MK mode at low load and switches to regular diesel operation at high loads. A high swirl ratio is employed when operating in MK mode, as well as a high EGR level and retarded injection timing to prolong the ignition delay. With this approach it is possible to achieve a longer ignition delay than the
injection duration, hence providing time for the fuel to mix with air before combustion. The pre-mixed combustion results in very low NOX and particulate matter emissions.

**Honda AR Motorcycle Engine**

The Honda Active Radical (AR) engine [22] is a 2-stroke single-cylinder engine that operates at spark-ignition mode at high loads, idle, and for cold-starts. Transition to HCCI combustion occurs at low load. The engine has a low (6.1:1) trapped compression ratio, and HCCI operation is obtained by throttling the exhaust. With exhaust throttling, the engine operates with a high fraction of hot residual gases, which is enough to obtain HCCI combustion, even at this very low compression ratio. Exhaust throttling is decreased as the load increases, until finally the residual fraction is too low to keep the engine operating in HCCI mode, where the engine switches to spark-ignition mode. The performance map has a "transition region" where the engine can operate in both HCCI mode and SI mode. The AR engine has demonstrated considerable advantages in fuel economy; at 50 km/h cruising speed the fuel economy improved 57%, with a simultaneous 65% reduction in HC emission.

**Toyota UNIBUS**

The Toyota UNIBUS comprises a combination of early and late injection timing. The operation regime is up to half load and speed. The concept is described by Hasegawa and Yanagihara [23]. The early injected fuel undergoes a low-temperature reaction while the late injection serves as an ignition trigger and sets off the high temperature reaction. The first injection quantity was small, 5 to 15 mm³. The concept is known as UNIBUS (Uniform bulky combustion) and was applied to the production engine (1KD-FTV, 3-liter 4-cylinder) in August 2000 on the Japanese market.
HCCI FUEL INJECTION CONCEPTS

The two main fuel injection concepts for HCCI are port injection (PI) and direct injection (DI). Port injection works with volatile fuels (like gasoline) while direct injection is more suitable for heavy fuels (like diesel).

**Port injected HCCI**

A common way to premix fuel with air in an engine is to use a port injection system, which is simple, cheap and creates a well-mixed mixture [16]. This approach is appropriate when running HCCI combustion on gasoline and numerous studies of HCCI have been conducted using this configuration. A limitation for port injected gasoline HCCI is on very low load, when the combustion temperature falls under a critical value (1400-1500 K for speeds typical for heavy-duty truck engines) and large amounts of HC and CO are formed, thus depleting the combustion efficiency [27]. Another drawback is the wall film in the intake system that is rather slow reacting to engine transients and could possibly obstruct the combustion control when using variable valve timing (VVT). Port injection with diesel fuel has additionally difficulties since its higher boiling range (200 – 300°C for Swedish MK1) results in poor evaporation. The non-evaporated fuel will adhere in the intake system and on the combustion chamber walls, which dramatically increases the smoke and HC emission and dilutes the lubricating oil [16], [30], [31]. Raising the intake temperature minimizes the fuel wall-film in the intake system and reduces the HC emission, even if this still is on a high level [16], [30], [32]. Furthermore, raised intake temperature reduces power density and advances the combustion timing of the engine. This is a drawback since the HCCI engine suffers from low power density. The diesel-HCCI engine needs a very low compression ratio to delay the ignition timing, which is bad for the efficiency. Therefore it is not a good idea to further advance the ignition timing with a raised intake temperature.

**Gasoline direct injected HCCI**

Although port injection works for gasoline-HCCI, several investigations demonstrate the advantage of using a gasoline direct injection (GDI) system [24]-[29]. The possibility of stratifying the air/fuel charge with late direct-injection will increase the local combustion temperature. This results in decreasing HC and CO emissions, hence increasing combustion efficiency at lower loads. It also offers a
limited possibility of phasing the combustion timing with the injection timing. A drawback with a GDI system is higher cost.

**Direct injected diesel-HCCI**

Direct injection eliminates the need for a raised intake temperature, since injection can occur at elevated temperatures in the compressed in-cylinder gases. The task of the injection process is to create a sufficiently premixed mixture before ignition occurs. A sufficiently premixed mixture can be defined as one which creates low NOX and smoke. Also, it is desirable to keep CO and HC emissions as low as possible, even though these emissions are a natural consequence of low-temperature premixed combustion. An equally important task for the injection strategy is avoiding fuel adhering or condensing to the cylinder walls. A key parameter is to control the fuel penetration length, i.e. avoid any contact of liquid fuel with the cylinder and combustion chamber walls. Direct-injected diesel HCCI can be divided into two areas, early injection (early under the compression stroke) and late injection (usually after TDC).

**Early-injected diesel HCCI**

A certain time is always needed for the process of fuel injection and mixture homogenization. The consequence is advanced injection timings for early-injection diesel HCCI combustion. Early injection timings mean lower gas temperature and density in the cylinder, which will reduce fuel vaporization rate and enhance the liquid penetration length. On the other hand, “later” early injection timing means higher gas densities and eventually vaporizing conditions, which will lower the liquid penetration length and reduce the risk for wall wetting. Also, the rate of entraining gas mass into the spray is higher at late injection timings, which creates a leaner spray that mixes faster to globally lean conditions [33]. This situation is illustrated in Figure 9. A potential of reduced HC and CO emissions is also accompanied with late injection timings due to the potential of less fuel in the squish region [34]. Shorter residence time for the fuel in the combustion chamber before combustion reduces the risk of fuel condensation on cold surfaces. However, drawbacks with later injection timings are shorter mixing time before combustion which causes a need of a more advanced high-pressure injection system which can provide the high-momentum sprays which are needed for a high mixing rate.
When injection timing is advanced, the ignition delay becomes longer and the mixture becomes more homogeneous before combustion. Emissions formation from “late” early-injection combustion was studied by Musculus [35]. The operating conditions were low-temperature combustion, where injection was completed just before main heat-release. The results indicate that NO formation occurs throughout the jet, rather than being formed near a thin diffusion flame on the jet periphery, as in conventional diesel combustion. Liquid fuel penetration lengths were in this case found to be approximately twice as long as for conventional diesel combustion. Soot is formed in the fuel-rich head vortex that is formed after impingement, where the mixing rate is slow. This is illustrated in Figure 10.
Main goals of DI-diesel HCCI research are reduced cylinder wall wetting and increased evaporation and mixing of the spray. Charge stratification should be achieved to avoid low temperature combustion near walls and to enhance load stability [36]. Different strategies have been tested for early-injection mixture formation. These can roughly be divided in hollow-cone sprays, impinging sprays and ordinary diesel sprays – often in combination with a narrow included angle (definition in Figure 17), and/or multiple injections.

**Impinging Sprays**

Takeda et al. [37] tested a system for early injection consisting of two side-mounted injectors, where the sprays impinged in the centre of the combustion chamber. Very low NO$_X$ emissions were found for this premixed combustion type, however HC and CO were high. A small effect of swirl ratio on emissions was found. The load could be increased by adding a late injection to the early injection (Hashizume et al. [38]), but this was paid for with increased NO$_X$ and smoke. Akagawa et al. [39] found that the mixing process of the side-mounted impinging injectors resulted in significant wall-impingement. Split-injection could reduce the HC and CO emissions, however paying with an increase of NO$_X$ (Nishijima et al. [40]).

Iwabuchi et al. [41] improved the mixing characteristics using a novel impinging-spray nozzle, where the fuel from two injector holes impinged at their orifice exit. The sprays from this nozzle were found to have large cone angles and low penetration rates. Combustion studies with this nozzle resulted in a significant reduction in fuel consumption and emissions compared to the conventional single hole nozzle.
Impinging-spray nozzles and nozzles with narrow included angles were tested by Nordgren et al. [42]. Low NO\textsubscript{X} was achieved with all nozzles, however lowest HC emissions were obtained with the impinging-spray nozzles.

**Hollow-cone sprays**

Harada et al. [43] tried fully premixed combustion using swirling-flow pintle-nozzles. One of the pintle-nozzles provided a more uniform mixture than the others. HC and CO emissions were improved compared to a micro-hole nozzle. Akagawa et al. [39] concluded that major sources to HC and CO were adhesion of fuel to the cylinder line and flame quenching by the cylinder liner or combustion chamber walls. Adhesion of fuel could be reduced using a pintle-nozzle with swirl grooves. Reducing the topland crevice volume significantly reduced HC.

A hollow-cone spray in conjunction with a deep-bowl combustion chamber was tested by Ishima et al. [44]. The operation regime was low load, up to 0.35 MPa indicated mean effective pressure (IMEP). The fuel consumption was generally high, presumably due to bad combustion efficiency.

A low penetrating hollow cone spray was tested for premixed diesel combustion by Ra et al. [45]. Injection timing had to be very early to avoid NO\textsubscript{X} formation. High CO emissions were observed for all injection timings. The main cause was identified as wall wetting.

**Standard diesel sprays**

Takeda et al. [37] tested three different types of nozzles for early injection. One standard six-hole nozzle, one sixteen-hole nozzle with reduced hole diameter and one 30-hole nozzle with reduced hole diameter and three different included angles. The nozzle that had best fuel consumption in combination with low NO\textsubscript{X} was the 30-hole nozzle; however, HC and CO emissions were still high. Small variations in emissions with injection pressure were found. The same 30-hole nozzle was later tested by Harada et al. [43], and it was found to significantly over-penetrate.

The problem of fuel wall impingement was recognized by Iwabuchi et al. [41]. Nozzles of different included angles were tested to minimize the wall wetting. A minimum of wall-impingement was found with 80° included angle.

The approach of a small included spray angle was also tested by Gatellier et al. [46]. In combination with a deep bowl combustion chamber, it could operate at high load using conventional diesel combustion, and at lower load using HCCI combustion. HCCI fuel consumption was high in some cases due to high HC and
CO emissions. Good fuel consumption with HCCI combustion could be achieved up to 0.9 MPa IMEP.

Shimazaki et al. [47] performed premixed diesel combustion using a narrow included-angle nozzle. The injection timing was late in the compression stroke near TDC. The strategy was based on high-pressure late injection into high temperature – which promotes the turbulent mixing rate and reduces the risk of wall wetting. CFD-simulations indicated that the low HC emissions observed at the late injection timings are due to reduced fuel in the squish area. The smoke emission appeared related to the duration between end of injection (EOI) and ignition. The authors suggest a dual-mode operation with conventional diesel combustion at high load. The examined premixed diesel operation regime was at very low load.

A combination of early and late injection timings was investigated by Hasegawa and Yanagihara [23]. The early injected fuel undergoes a low-temperature reaction while the late injection serves as an ignition trigger and sets off the high temperature reaction. The first injected quantity was small, 5 to 15 mm³.

A very narrow (60°) included-angle nozzle in combination with multiple injections was tested by Helmantel and Denbratt [48]. 0.9 MPa IMEP was achieved with a low compression ratio and high EGR levels. Combustion efficiency based on HC and CO ranged from 86 % at low load to 94 % at high load. However, ordinary diesel operation with the narrow included angle resulted in high fuel consumption and smoke emissions, despite modifications to the combustion chamber (Helmantel et al. [49]). The 60° nozzle was therefore replaced with a 140° nozzle, which improved combustion efficiency. HCCI operation with the 140° nozzle was successful using multiple injections; however the tested load in this case was only 0.3 MPa IMEP.

A study on injection parameters such as included angle, injection timing, number of injections and injection pressure was made by Buchwald et al. [50]. Low fuel consumption was found with the combination of either:

- Large included spray angle, low injection pressure, many injections, late injection timing.
- Small included spray angle, high injection pressure, few injections, early injection timing.

The possibility to operate with a large included spray angle is advantageous for conventional diesel operation at high load. However; the investigation was
performed at low load, IMEP was 0.33 MPa. Expanding the load range with premixed diesel combustion using a large included angle may be difficult due to wall wetting.

An optical study of premixed diesel combustion was made by Kanda et al. [51]. The first experimental setup had a narrow included angle nozzle in combination with a shallow bowl combustion chamber. The other setup was large included angle nozzle with a re-entrant bowl. Fuel impingement onto the piston at early injection timings was identified as a source of HC, CO and soot.

Premixed-charge compression-ignition (PCCI) diesel combustion was investigated by Hardy and Reitz [52], using high EGR levels. A narrow included angle nozzle was used. Equivalence ratios of up to 0.94 were tested. High HC emission was observed at this high equivalence ratio. CO emissions were high at low load. Piston impingement was thought to be the reason for high HC and fuel consumption.

Different nozzle included angles were tested by Lee and Reitz [53]. An attempt to optimize the spray targeting in the bowl was made. Testing conditions were low load and high EGR levels. It seemed be a correlation between low soot emissions and spray targeting to the piston bowl edge near the squish region, regardless of included angle. However, all the details of the mixing process are not clear from the results.

Late-injection diesel HCCI

Late-injection diesel HCCI is commonly referred to as MK-combustion, after the combustion concept by Nissan [54] - [56]. A huge advantage with the concept is that the combustion timing can be controlled by the injection timing, unlike the early-injection or port-injection concepts. The use of standard diesel fuel injection equipment also allows for running with ordinary diesel combustion at high load. The concept relies on prolonging the ignition delay to allow for sufficient mixing between fuel and gas. A longer ignition delay is obtained with late injection timing. A high swirl level reduces HC and smoke through enhanced mixing. In opposite to what might be expected, a higher swirl ratio also seems to reduce heat losses in this combustion mode. The overall reduced cooling losses counteract the deteriorating fuel consumption due to late combustion timing. The MK-combustion mode is operative at part load; however the load-range can be increased by increasing the ignition delay through a lower compression ratio and cooled EGR. Applying a large-diameter combustion chamber bowl reduces HC emissions during cold-start, presumably due to less wall-impingement. A higher injection pressure was effective in reducing NOX and particulate matter under MK combustion conditions.
This late-injection strategy was examined by Miles et al. [57]. The study indicates that although longer ignition delay promotes a formation of a more uniform mixture, it is still far from homogeneous. It is suggested that the initial part of combustion is dominated by chemical kinetics and that a significant portion of the mixture is fuel-rich. The subsequent part is mixing-controlled combustion – similar to normal diesel operation. The reduced heat losses as a function of swirl ratio was explained as a larger formation of combustible mixture near the piston bowl walls at lower swirl ratio.

Comments
The late-injection MK-combustion concept seems to be the simplest and most useful concept. It offers the ability to control the combustion timing with the injection timing and does therefore not require any extra equipment to control the effective compression ratio. It can also easily switch between premixed combustion at low load to diesel combustion at higher load. High load HCCI is limited by the rapid heat release, so a dual-mode system is probably a good idea. Regarding early-injection concepts; hollow-cone sprays require very advanced injection timing, and seem to suffer from problems with wall interaction. Standard sprays with narrow included angles seem to have potential, as well as the concept of the impinging-spray nozzle.
FUEL INJECTION AND SPRAYS

A key to diesel-HCCI is the in-cylinder air / fuel mixing process. One of the most important, if not the most important, parameter for in-cylinder air / fuel mixing is the injection process and spray formation. The purpose of this chapter is to provide an overall background to this topic.

Spray structure and breakup

The fuel injection pressure is transformed to velocity over the nozzle holes. The fuel velocity together with the mass flow makes up the fuel momentum, which is transferred to surrounding gas in the combustion chamber, causing mixing. The liquid jet rapidly disintegrates into drops, which tend to maintain the general direction of motion of the original jet [58]. The corresponding propagating mixture between fuel and gas is recognized as a spray, see Figure 11.

Atomization is the process where bulk liquid transforms to droplets. As the fuel exits the nozzle hole and penetrates the surrounding gas, the liquid breaks up and mixes with the surrounding gas. The breakup process is very different depending on the conditions and characteristics of the liquid and surrounding gas. Breakup regimes are generally defined depending on Reynolds number and Weber number [58]. At lower Weber and Reynolds number lie the Rayleigh, First- and Second-wind breakup regimes (See for example [58]). These are not relevant for diesel injection conditions and are not further dealt with here. The diesel injection process operates in the atomization regime, which is located at higher Weber and Reynolds numbers.
Historically, diesel atomization was thought to progress by primary breakup forming droplets through stripping from boundary layers of an intact liquid core surface which extended far downstream of the nozzle, followed by a secondary breakup of ligaments and large drops [59]. The intact liquid core was thought to extend beyond more than 100 nozzle diameters downstream, see Figure 12 left image. Measurements of tip penetration showed an initially linear development with time, thereafter a sharp transition to square root of time. The transition was interpreted to be that from an intact liquid core to an atomized spray [60]. More recently, however, it has been shown that the transition is rather smooth, and the alternative interpretation is that the spray evolves from primarily liquid to primarily gas [61]. The current concept is that the diesel spray structure under normal operating conditions is completely atomized at, or close to the orifice exit, thereafter there is no trace of a prevailing liquid core, see Figure 12 right image.

According to Smallwood and Gülder [59], primary breakup mechanisms are cavitation, turbulence-driven instability and possibly buckling during the very early phase of injection. Aerodynamic shear appear much less important than previously thought, but may be responsible for secondary breakup. Cavitation influences breakup through collapsing and bursting vapour bubbles, which contributes to the disintegration of the jet, and also increases the turbulence and instability of the jet. Soteriou [62] observed that when plug cavitation reached the end of the hole, there was a sudden additional increase in the spray angle. Cavitation causes significantly increased turbulence, and was therefore proposed to be a major factor for atomization. Kim et al. [63] observed high turbulence in the sac chamber generated by the high velocity of the needle seat flow when the needle lift was small. This
caused a large spread angle of the spray plume. The importance of cavitation on spray breakup was also investigated by Hiroyasu [64], where cavitation-induced turbulence in the nozzle hole appeared to be the primary source leading to spray disintegration.

**Nozzle Flow**

The disintegration process of a liquid jet from a diesel nozzle seems primarily controlled by internal nozzle-flow rather than interfacial forces between jet and surrounding gas. The details of nozzle flow are however complicated, and have become a field of study on its own. An early investigation was made by Bergwerk [65] of spray holes between 0.2 and 2.5 mm. He concluded that as cavitation number $CN = (P_f - P_a)/(P_a - P_v)$ increased, cavitation bubbles formed at the sharp entrance to the nozzle hole, where velocity was locally high and thus pressure was low. At a certain pressure ratio, the cavity extended the full length of the hole. This situation is illustrated in the study by Winklhofer et al. [66], see Figure 13.

Bergwerk [65] concluded that a more rounded inlet prevents the high velocity and thus delays cavitation. A higher nozzle hole length / diameter (l/d) required a higher $CN$ for bubbles to reach the orifice exit. At a critical injection pressure the jet leaves the wall of the hole so that only the upstream corner of the hole has any effect and the jet emerged straight and smooth. This is known as hydraulic flip and is an undesirable condition in an engine, since the spray breakup, and thus mixing with gas, is inhibited. In practice, this never occurs in diesel engines since there is
Experimental Investigation of Impinging Diesel Sprays for HCCI Combustion

sufficient turbulence in the flow, partly due to the nozzle geometry but also due to imperfections in the material from manufacturing. Soteriou et al. [67] concluded that the high turbulence in the sac would prevent hydraulic flip, but identified partial hydraulic flip, a situation where one side of the nozzle hole is filled with gas. This causes an asymmetric spray, and is more likely to occur in a diesel nozzle. The Reynolds number was concluded unimportant for nozzle discharge while CN dominated. Schmidt et al. [68] observed separation at the orifice exit to be a function of Re for low CN, and a function of CN for high Re numbers. Chaves et al. [69] made discharge and flow velocity measurements of real-size transparent nozzles. They found that if the pressure at the vena contracta reaches the vapour pressure, any further increase in injection pressure will make the cavitation reach the nozzle exit (supercavitation), which causes a large increase in the spray angle. The C_D and the spray angle levelled off at a value almost independent of any further increase in injection pressure, and flow became independent of back pressure. Chaves et al. also found that long nozzles have a similar C_D as short nozzles; once supercavitation is reached, any losses in the nozzles are compensated by a change in the effective area of the nozzle. The true exit flow velocity is much higher than the geometric mean velocity; velocity calculated on discharge measurements and geometric hole area are not a measure of the true exit velocity. The measured velocity was very close to the velocity given by the Bernoulli equation (Bernoulli velocity).

Collicott and Li [70] describe the flow in diesel fuel injector orifices as an unsteady non-equilibrium two-phase flow with substantial roughness-induced cavitation. They conclude that modelling of either the internal flow or the spray formation process outside the hole are exceedingly difficult tasks.

Spray macro-scale characteristics

A lot of work has been done studying diesel sprays. The spray can be characterized with geometrical properties like cone angle, penetration and volume, see example in Figure 14.
Spray penetration was investigated by Hiroyasu et al. [60] for non-evaporating diesel sprays. They developed a correlation for spray penetration, however injection pressures were lower than 40 MPa in their experiments.

A study by Naber and Siebers [61] involved injection pressures between 75 – 160 MPa and gas densities between 3-61 kg/m$^3$ for vaporizing conditions, and 3 – 200 kg/m$^3$ for non-vaporizing environments. A spray model based on conservation of mass and momentum was derived using integral control surface techniques. The model, as well as derivation of the orifice coefficients, is presented in the Appendix.

**Evaporating sprays**

It was found by Naber and Siebers [61] that vaporization decreased penetration and dispersion by as much as 20% relative to non-vaporizing conditions, with decreasing effect of vaporization with increasing gas density. They hypothesized that the gas density increases locally as it is cooled by the evaporating spray. The higher density mixture slows newly injected fuel more rapidly, thus slowing the tip penetration. The increase in density also leads to a contraction of the spray that explains the reduced spray dispersion angle compared to non-evaporating sprays.

Evaporating diesel sprays were further investigated by Siebers [71] where the liquid length was examined with respect to variations in injection pressure, orifice diameter and aspect ratio, ambient gas density and temperature, and fuel volatility and temperature. The results were:
Decreasing liquid length with decreasing orifice diameter.
- No significant impact of injection pressure on liquid length.
- Decreasing liquid length with increasing ambient gas density or temperature, but with a declining sensitivity to each one as they increase.
- Decreasing fuel volatility or fuel temperature increases the liquid length.
  
  At 1300 K gas temperature, three fuels with different boiling regimes had nearly the same liquid length, while at 700 K, the difference was as much as 70%.
- The liquid length of a multi-component fuel is controlled by its lower volatility fractions.
- Orifice aspect ratio (hole length over hole diameter) had a small and inconsistent impact on liquid length.

In further studies by Siebers [99], a scaling law for the maximum liquid penetration length of a vaporizing spray was derived. It was suggested that vaporization is controlled by the mixing process in the spray, which includes air-entrainment and overall transport and mixing of fuel and air throughout the spray cross-section. The results imply that the atomization and the ensuing interphase transport of mass and energy at droplet surfaces are not limiting steps with respect to fuel vaporization in DI diesel sprays. Vaporization was examined using jet theory:

\[
\dot{m}_f \propto \rho_f \cdot d^2 \cdot U_f \tag{1}
\]

\[
\dot{m}_a \propto \sqrt{\rho_a \cdot \rho_f \cdot d \cdot x \cdot U_f \cdot \tan(\theta/2)} \tag{2}
\]

where \( \dot{m}_f \) is the injected fuel mass flow rate and \( \dot{m}_a \) is the total entrained gas mass flow rate up to any axial location in a spray. The terms in the equations are the ambient gas density \( \rho_a \), the fuel density \( \rho_f \), the orifice diameter \( d \), the injected fuel velocity \( U_f \), the axial distance from the orifice \( x \), and the spray cone angle \( \theta \). Thus, doubling the orifice diameter quadruples the fuel injected but only doubles the entrained gas, which results in twice as long liquid penetration length before the fuel is vaporized. Since both equations are linearly dependent of injection pressure, there is no impact of injection pressure on liquid length.

Interestingly also, the liquid length does not shorten significantly after ignition occurred, vaporization occurs largely upstream of the combustion zone formed after ignition.
The schlieren images of evaporating sprays in Figure 15 range injection pressures from 30 – 110 MPa. The liquid-phase fuel penetration appears constant regardless of injection pressure, as was predicted by eq. (1) and (2). However, it can be seen that the gas-phase penetration is significantly impacted by the injection pressure. The gas-phase penetration will impact the air-fuel distribution in the cylinder and thus influence the combustion process and emissions formation.

![Figure 15. Average of 30 schlieren images of evaporating sprays, on courtesy of E. Winklhofer. The surrounding gas is air with temperature 900 K. Nozzle hole diameter is 0.11 mm. The liquid penetration lengths (black core) appear constant regardless of injection pressure. The vapour penetration is significantly higher with higher injection pressure, which will have a large impact on the lambda distribution in the engine.](image)

**Surrounding Flow Field - Air Entrainment**

Investigating the surrounding flow field of a spray can give information of the air entrainment into the spray. Such a study was performed by Sasaki et al. [72]. They investigated the surrounding air field by using a particle imaging velocimetry (PIV) system. They found that rather small amounts of air was entrained near the nozzle tip, more air entrainment occurred in the spray mid-section and tip. Furthermore, they found that increasing the fuel velocity imposed an almost proportional increase in the surrounding air velocity in vicinity of the nozzle. Further downstream, however, the influence was much smaller. A smaller hole diameter reduced the
surrounding air velocity, which is expected since the momentum transferred to the surrounding air is lower with lower injection rate.

Ishikawa and Zhang [73] studied air-entrainment by using the air density difference as a tracer of the moving air. They found that $\frac{\partial m_{\text{air}}}{\partial m_{\text{fuel}}}$ did not change with injection velocity, which is in agreement with Sasaki et al. [72], and also the jet theory used by Siebers, eq. (1) and (2).

Rajalingam and Farrell [74] studied air-entrainment using PIV. They found little differences in air entrainment with injection pressure on the first two-thirds of the spray plume, while on the last third there was a major difference.

Further studies on non-evaporating diesel sprays by Rhim and Farrell [75] suggest that a significant part of the overall gas entrained in a spray plume is entrained from the spray tip. This stands in contrast to the common perspective that most of the gas is entrained through the lateral sides of the sprays, and that the gas near the spray tip is just pushed aside by the spray tip. However, the velocity plots agree with the conventional belief about the gas flow pattern and resulting gas entrainment along the sides of the spray plume. A schematic of the results is shown in Figure 16.

Figure 16. General image of gas motion relative to the spray boundary for non-evaporating transient sprays [75].

For a six-hole injector, the positive normal velocities relative to the spray boundary were lower than for a spray from a single-hole nozzle (Rhim and Farrell [76]). A difference of ~ 20% more mass was measured for the single spray. An increase in velocities tangential to the spray boundary was found for the six-hole injector. With
regard to evaporating sprays; both tangential and normal velocities seems to be larger for vaporizing sprays (Rhim and Farrell [77]). The air-entrainment with respect to burning sprays appear similar as for vaporizing sprays, (Rhim and Farrell [78])

**Impinging sprays**

The mutual impingement of two or more jets (or sprays) have been investigated by many researchers under different conditions. The motivations for investigation are varying. Impinging jets are a common application in biliquid propellant rocket engines, where the jet-impingement results in simultaneous atomization and mixing of the liquid propellants, see for example [79] - [86]. Understanding this process is important for maximizing the performance of the engine.

Impinging jets can also be used for reaction injection moulding, a manufacturing process where two liquid resins are rapidly metered, mixed, and delivered to a mould where they react and cure to form a solid polymeric part [87]. Another technique is to use impinging jets for industrial precipitation where the kinetics is rapid, and a fast mixing is required [88].

In direct-injected reciprocating engines, impinging sprays are interesting due to their different spray characteristics, such as cone angle and penetration. The spray characteristics are decisive for the in-cylinder air-fuel mixture preparation. A number of investigations have been made on impinging sprays with higher velocities [89] - [94].

**High-velocity impinging sprays**

The location of impingement can be inside the nozzle, at the nozzle exit or downstream of the nozzle exit. Both sac and valve-covered orifice (VCO) nozzles can be used. Examples of impinging spray nozzle designs are shown in Figure 17.
The general characteristics of the impinging spray are a larger cone angle and a shorter penetration than the non-impinging spray. This is illustrated in Figure 18.

Impinging high-velocity sprays can be assumed to be very different from impinging liquid jets; an impingement of a more or less fully atomized, highly turbulent cavitating flow. If cavitation-induced turbulence has a big impact on breakup, then one plausible effect of the impingement is to contribute to spray disintegration through collision-induced turbulence. However, measured droplet diameters as a function of impingement angle [90] and impingement distance [91] reveal a fairly
small impact on droplet diameters. The reason for this might be that the spray already is more or less fully atomized, at least at the location for measurement.

An investigation of impinging flow nozzles were made by Yamamoto and Niimura [89]. Nozzles with the impingement point located internally, externally and at the orifice exit were tested, as well as a standard nozzle. However, also the impingement angle was different for all three nozzles, (39°, 6° and 24°, respectively). Penetration was similar between the nozzles except for the one with internal impingement, which was lower. The impinging spray nozzles generated internally a more homogeneous mixture through its higher turbulence. The high fuel concentration typically observed at the centre of the spray with a conventional nozzle was made leaner and the distribution was made more homogeneous. The fuel spray tended to spread out more perpendicular to the plane containing the two nozzle holes, although the spray shape at 70 mm from the orifice showed a circular cross-section. Impingement at the orifice exit plane seemed most efficient in terms of air-entrainment.

Iwabuchi et al. [90] studied sprays from impinging spray nozzles, performed engine tests and examined the combustion and emissions from an impinging spray nozzle compared with a conventional non-impinging nozzle. They found that as the impingement angle was increased, the spray angle increased, the penetration decreased, and the fuel concentration within the spray became more uniform. The spray cone angle was in fact a linear function of the impingement angle. The Sauter-Mean-Diameter (SMD) increased slightly for collision angles 10°, 20° and 30°, but decreased for collision angles 40°, 50° and 60°.

A study of impinging sprays was made by Chiba et al. [91]. They studied the impact of impinging distance for impingement angles of 60° and 90°. Injection pressure was 23.5 MPa and ambient pressure was 1 MPa. With an impingement distance of 46 or 70 mm, the penetration was found to be longer with 60° impingement angle, than if the individual sprays were non-impinging. However, with impingement distance 20 mm, the penetration was lower than for the individual sprays. For 90° impingement angle, the penetration was lower than the non-impinging sprays regardless of impingement distance. Spray volume of the impinging spray was at maximum ~ 1.5 times larger than the non-impinging spray. The SMD was found to be slightly decreasing with increasing impingement distance. Spray volume was smaller with 90° than with 60° impingement angle.
INVESTIGATING IMPINGING DIESEL SPRAYS

The focus of this PhD-work was to test and evaluate means to provide mixture formation for HCCI diesel combustion. A challenge was seen in forming a lean or diluted mixture of fuel and in-cylinder gas during intake and compression stroke, but simultaneously avoiding excessive wetting of combustion chamber surfaces with diesel fuel. It was understood that this task could be achieved by injecting early during the compression stroke to provide sufficient time for mixing and evaporation, but too early injection would result in spray wall impingement as the spray might propagate through the low density, low temperature gas rather than evaporate and mix with the gas.

Thus the focus was on finding and evaluating concepts for diesel injection which make sprays with large volume, suited to fill the cylinder, but with small penetration in order to avoid wall contact. Conventional diesel nozzles were not considered a good option to avoid spray wall impingement at early injection timings, due to a high penetration rate. Therefore, the choice was on impinging sprays. A few papers had previously been published on the topic, and the concept appeared promising. Spray analysis and engine tests of impinging sprays thus formed the central part of this work. The goal was derived as the answer to the following questions:

1. What is the difference between standard- and impinging-sprays?
2. What is the effect of such sprays with respect to diesel HCCI?

The tools for answering the goal were a spray visualisation facility and a single-cylinder engine, described in the next section.

Experimental Equipment

A pressurised vessel with optical access was designed and built to allow for optical analysis of the sprays. The pressurized vessel consists of a metal cylinder with a rectangular pipe welded to the top, see Figure 19.
One injector holder and two window holders were mounted to the rectangular-shaped pipe, while one window holder was mounted to the bottom of the cylinder; the vessel has thus optical access from three directions, from below and from the sides. The vessel was operated at room temperature from 0.1 – 1.2 MPa absolute pressure.

The complete spray visualization facility is shown in Figure 20; it consists of the pressurized vessel, two 250 J studio flashes, a CCD camera and a data acquisition system. Imaging from the side is done with back illumination and imaging from below is done with side illumination. When imaging is from the side, the two flashes are pointed at a 10 mm thick sheet of cellular plastic to create an evenly illuminated background. When imaging from below, one flash is located on each side of the vessel to achieve illumination which is as even as possible. The flashes are then pointed directly at the sprays, with no additional diffusers between them. The CCD camera receives the image through a mirror. To suppress reflecting light from the walls, the inside of the vessel is painted matte black.
Investigating impinging diesel sprays

Figure 20. The spray visualization facility.

Engine tests were made with a 4-valve single cylinder engine based on a Scania D12 cylinder with characteristics according to Table 3.

Table 3. Engine specifications

<table>
<thead>
<tr>
<th>Engine type</th>
<th>Scania single-cylinder</th>
</tr>
</thead>
<tbody>
<tr>
<td>Displacement</td>
<td>1.95 dm³</td>
</tr>
<tr>
<td>Bore</td>
<td>127 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>154 mm</td>
</tr>
<tr>
<td>Connecting rod</td>
<td>255 mm</td>
</tr>
<tr>
<td>Swirl ratio</td>
<td>1.78</td>
</tr>
<tr>
<td>Piston type</td>
<td>Shallow bowl</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>12</td>
</tr>
<tr>
<td>Intake valve open</td>
<td>-366 CAD</td>
</tr>
<tr>
<td>Intake valve close</td>
<td>-137 CAD</td>
</tr>
<tr>
<td>Exhaust valve open</td>
<td>141 CAD</td>
</tr>
<tr>
<td>Exhaust valve close</td>
<td>370 CAD</td>
</tr>
</tbody>
</table>

A Drawing of the used piston is provided in Figure 21. Spray picture from the pressurized vessel are included for reference.
An external compressor provides supercharging. Exhaust gaseous emissions were measured with a Horiba EXSA 1500 and smoke emissions with an AVL 439 opacimeter. The EGR source was cooled exhaust gases from a catalyst-equipped SI engine operating at stoichiometric conditions. A picture of the engine test bed is provided in Figure 22.
A summary of the different injector nozzles used in this work is provided in Table 4.

**Table 4. Tested nozzles. The definition of included angle and impingement angle is provided in Figure 17.**

<table>
<thead>
<tr>
<th>Nozzle #</th>
<th># of holes</th>
<th>(d_{\text{hole}}) [mm]</th>
<th>Impingement Angle [°]</th>
<th>Included Angle / 2 [°]</th>
<th>(C_D)</th>
<th>Used in</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>10</td>
<td>0.16</td>
<td>-</td>
<td>45 / 75</td>
<td>0.66</td>
<td>Paper IV / Paper V</td>
</tr>
<tr>
<td>2</td>
<td>10</td>
<td>0.16</td>
<td>30</td>
<td>45 / 75</td>
<td>0.66</td>
<td>Paper IV / Paper V</td>
</tr>
<tr>
<td>3</td>
<td>10</td>
<td>0.12</td>
<td>60</td>
<td>30 / 90</td>
<td>0.71</td>
<td>Paper III</td>
</tr>
<tr>
<td>4</td>
<td>10</td>
<td>0.16</td>
<td>60</td>
<td>30 / 90</td>
<td>0.64</td>
<td>Paper V</td>
</tr>
<tr>
<td>5</td>
<td>4</td>
<td>0.12</td>
<td>0 (parallel)</td>
<td>60</td>
<td>0.73</td>
<td>Paper I</td>
</tr>
<tr>
<td>6</td>
<td>4</td>
<td>0.2</td>
<td>0 (parallel)</td>
<td>60</td>
<td>0.66</td>
<td>Paper I</td>
</tr>
<tr>
<td>7</td>
<td>4</td>
<td>0.12</td>
<td>60</td>
<td>30 / 90</td>
<td>0.70</td>
<td>Paper I</td>
</tr>
<tr>
<td>8</td>
<td>4</td>
<td>0.2</td>
<td>60</td>
<td>30 / 90</td>
<td>0.68</td>
<td>Paper I / Paper IV</td>
</tr>
<tr>
<td>9</td>
<td>4</td>
<td>0.16</td>
<td>60</td>
<td>30 / 90</td>
<td>0.69</td>
<td>Paper IV</td>
</tr>
</tbody>
</table>

The work performed with this equipment has been described and published in papers which are appended to the thesis. An overview of the work and a summary of the papers are given in the following chapter.

**Work progress**

This section is a brief history of the work progress; which steps were taken and why. Each major step has been expressed in a report. The major findings from each step are provided below, as a summary of papers.

The initial task was to design and test the impinging-spray nozzles. Optical tests in the pressurized vessel confirmed that the manufacturing process was precise enough for the sprays to impinge properly. The first effort to understand the nature of the impinging spray and its impact on mixing, resulted in *Paper I*. Methods used for image evaluation were developed and are described in *Paper II*. As can be seen from the conclusions from the spray studies in *Paper I*, the concept of impinging sprays appeared promising. The next step was therefore engine tests with impinging sprays; the results were reported in *Paper III*. Variations of engine speed and injection pressure were made, which gave information on the relevance on such parameters. It was concluded that the impinging spray nozzle was successful with regard to mixture formation for HCCI operation. However; there were still many unanswered questions regarding properties such as cone angle, penetration and volume of impinging sprays. The understanding of ordinary diesel sprays had previously been significantly advanced through the use of scaling laws in combination with a penetration correlation based on conservation of momentum and mass, example in
Experimental Investigation of Impinging Diesel Sprays for HCCI Combustion

ref. [61]. It was therefore decided to try a similar procedure with the impinging sprays. The attempt was successful, and the result was a unique insight in impinging sprays. These results, which are the main contribution of this work, are presented in Paper IV. The impinging sprays had earlier been operated at low load in the engine. The next step was therefore to extend the investigation to higher loads, which is presented in Paper V. It was concluded that the impinging spray is successful with respect to mixing performance also at high load.

Paper I

This paper is a result of the first investigation of impinging sprays in the pressurized vessel. The approach was to investigate the sprays at EOI with respect to penetration, volume and air / fuel ratio, since these were considered important spray features for mixture preparation in the engine. The investigated nozzles were impinging with 0° (parallel) and 60° impingement angle. Two different hole diameters were chosen, 0.12 and 0.2 mm. Different injection pressures and injected fuel masses were tested. Major conclusions from the study were:

- The impinging sprays appear as one homogeneous spray with no trace of its individual sprays. The manufacturing of the impinging nozzles therefore seems reliable with respect to precision, i.e. the sprays actually impinge.
- The 0° nozzles could not be evaluated at EOI with higher injection pressure than 25 MPa, due to an excessive penetration rate which resulted in wall impingement in the vessel.
- The 60° nozzles could be operated at high injection pressure due to their lower penetration rate, which increased their volume and air / fuel ratio.
- The lowest penetration with respect to injected fuel mass was acquired with the nozzles with large hole diameter, since the mass flow rate through an orifice is proportional to the orifice diameter to the power of two, while the spray penetration is proportional to the square root of the orifice diameter.
- It was noted that the spray volume of the 60° nozzles decreased with increasing backpressure, while the spray volume of the 0° nozzles were relatively unaffected.

Experiments, data processing and evaluation were made by the author. Software for image evaluation was provided by Andreas Cronhjort.
Investigating impinging diesel sprays

Paper II

The second paper is a description and evaluation of an algorithm for automated segmentation of the spray images that was used by the author in the spray investigations. The algorithm was found to be robust to variations in image quality and illumination.

Derivation and evaluation of the algorithm was made by Andreas Cronhjort. The author provided spray images.

Paper III

Paper III is the result of the initial engine tests of impinging sprays. The used nozzle was a 10-hole 60°-impinging with hole diameter 0.12 mm. The tests were run at low load with no EGR. The approach was to run the engine at a constant low NO\(_X\), and evaluate required mixing time (ignition delay) and emissions with respect to different injection pressure and engine speeds. The tests showed that the impinging nozzle worked well in the engine with respect to emissions. Major conclusions from the study were:

- The air/fuel mixing time was shortened with increasing injection pressure and engine speed.
- Injection pressure and engine speed had no impact on the HC emission for injection pressures over 1000 bar. Under 1000 bar, the HC emission increased with higher engine speeds and lower injection pressures.
- The effect of injection pressure on CO emission was weak, with lower emissions at higher injection pressures. The CO emission increased with higher engine speeds.

Experiments, data processing and evaluation were made by the author. Software for extracting the mixing time was provided by Andreas Cronhjort, Software for heat release analysis by Hans-Erik Ångström. Ulf Olofsson supported the design of experiments.

Paper IV

This paper is the result of a fundamental investigation of impinging sprays. Different impinging and non-impinging spray nozzles were run in the pressurized vessel. Data were reduced with scaling laws that accounted for parameters like ambient pressure,
density, injection pressure, etc. Special modifications for exit velocity and fuel flow area were made for the impinging spray nozzles. The collapsed data were matched to a non-dimensional penetration correlation, which was used for the understanding of the characteristics of impinging sprays. Major conclusions from the study were:

- The cone angle of high-velocity impinging diesel sprays was confirmed to increase with the impingement angle.
- The effect of ambient density on the cone angle of the impinging sprays appeared weak, in contrast to the cone angle of non-impinging sprays which increased with ambient density. For the impinging sprays, at low ambient density, the cone angle was slightly smaller at the plane containing the two nozzle holes, than perpendicular to that plane. The difference vanished at higher ambient density.
- The penetration and volume of impinging sprays could be predicted in a manner similar to non-impinging sprays, using a penetration correlation based on conservation of momentum and mass.
- The spray volume can be described as a function of cone angle and penetration. Any increase in cone angle will cause a subsequent decrease in penetration due to conservation of momentum. The spray volume vs. cone angle has a maximum due to these competing phenomena.
- The volume of the impinging spray decreases rapidly with increased ambient density, since penetration becomes lower while the cone angle is relatively unaffected. The volume of the non-impinging sprays shows a smaller decrease with increased ambient density, since cone angle increases while penetration decreases.
- At low ambient density, the volume of the impinging spray is generally larger than the volume of the non-impinging sprays. As ambient density increases, the difference becomes smaller, and at a certain density the volume of the non-impinging sprays surpasses the volume of the impinging spray.

Experiments, data processing and evaluation were made by the author. Software for image evaluation and spray parameter extraction was provided by Andreas Cronhjort.
Investigating impinging diesel sprays

**Paper V**

The purpose of this paper was to evaluate the mixing performance of impinging- and non impinging-spray nozzles through engine tests. Tests were conducted with nozzles of different impingement angle: 0°, 30° and 60°. The tools used for evaluation were emissions and calculated heat release. The tests were run at low load without EGR and at high load with EGR. The difference between the emissions at low load can be explained with respect to the different spray penetration rates for the different nozzles. At low load, all of the nozzles provided sufficient mixture quality.

Running at high load with 47% EGR and lambda 1.05, anything less than a homogeneous mixture were expected to result in CO, HC and/or smoke emissions. The 30° nozzle performed very well at this operating point in terms of emissions and efficiency. The 0° nozzle performed well in terms of emissions, but the combustion efficiency with respect to heat release was inferior to the 30° nozzle. This was interpreted as a fuel loss to the crankcase via the impingement on the liner. The 60° nozzle presumably suffered from a low penetration rate which resulted in high CO emissions, except at very early injection timings. However, at these early injection timings, the HC emissions were high for the 60° nozzle. At the high load case, the 30° nozzle was preferable.

It is concluded that impinging nozzles can be used as a tool for influencing the mixing performance for early direct-injected premixed combustion.

The study was performed by the author.

**Summary**

The main conclusions from each individual paper are found under the Work progress section. A contribution to the understanding of impinging sprays has been made; the difference between impinging and non impinging sprays can now to some extent be answered for with respect to their macro-scale characteristics. Impinging sprays are found to be powerful with respect to the task of providing well-mixed air-fuel mixtures while avoiding wall impingement.
APPENDIX

The appendix contains derivations of orifice coefficients and the model for spray penetration. The orifice coefficients are a requirement in the penetration model.

Modelling the orifice coefficients

Nurick [96], who was studying the impact of cavitation on mixing between impinging jets, performed experiments and studied cavitation with scaled-up transparent nozzles. His model that explains the behaviour of the discharge coefficient with respect to cavitation has proved very useful, and is often adopted in literature. For example, the model is used for spray modelling in the computational fluid dynamics (CFD) code FIRE [97].

Comparison of the model with experimental data is provided in Figure 24. The model is reproduced below. Consider the nozzle flow model in Figure 23. The non-cavitating flow first contracts in the vena-contracata (point C), and then expands. Assuming that diffusion losses occur only in the reattachment process the flow will be ideal from the plenum (f) and cavity (C).

![Figure 23. Model of flow in sharp-edged orifice. (From [96])](image)

![Figure 24. Comparing experimental results and one-dimensional theory. (From [98])](image)

The Bernoulli equation for ideal flow, velocity at entrance $f$ assumed to be zero, is:

$$ P_f = P_c + \frac{1}{2} \rho_f Uc^2 $$  

(3)

From continuity:

$$ Uc A_c = \bar{U} u A_0 $$  

(4)
The contraction coefficient is defined as:

\[ C_c = \frac{A_c}{A_0} \]  \hspace{2cm} (5)

The value of \( C_c \) can be empirically estimated as:

\[ C_c = \left[ \left( \frac{1}{C_{c0}} \right)^2 - 11.4 \cdot \frac{r}{d} \right]^{\frac{1}{2}} \]  \hspace{2cm} (6)

where \( C_{c0} \) equals 0.611 (from [98]), \( r \) is the hole inlet radius and \( d \) is the hole diameter.

Combining eq. 3, 4 and 5.

\[ \frac{P_f - P_c}{\frac{1}{2} \rho_f U_a^2} = \left( \frac{1}{C_c} \right)^2 \]  \hspace{2cm} (7)

If the discharge coefficient is defined in terms of total losses occurring to the exit:

\[ \overline{U}_a = C_D \sqrt{2(P_f - P_a)/\rho_f} \]  \hspace{2cm} (8)

eq. 5 now becomes:

\[ \frac{P_f - P_c}{P_f - P_a} = \left( \frac{C_D}{C_c} \right)^2 \]  \hspace{2cm} (9)

This equation is of little practical use since the cavity pressure is unknown. However, as the flow increases, \( P_c \) eventually reaches the fluid vapour pressure \( P_v \), and the coefficient of discharge become:

\[ C_D = C_e \sqrt{\frac{P_f - P_v}{P_f - P_a}} = C_c \sqrt{K} \]  \hspace{2cm} (10)

The correlation predicts a low \( C_D \) at high injection pressures and a relatively low backpressure. \( C_D \) increases with \( K \) until the nozzle no longer cavitates. At this point the analysis is no longer valid and \( C_D \) is not necessarily a function of \( K \). In fact,
analogous one-dimensional analysis for non-cavitating orifices can be performed. Instead of fixing the contraction pressure, it is assumed that the flow fully expands to fill the nozzle. The predicted coefficient of discharge for non-cavitating nozzles using these assumptions is a constant of about 0.84 [98]. Figure 24 illustrates the compliance of the theory with measurements.

Further, if the mass flow at vena contracta is calculated as:

\[ \dot{m} = \rho_f \cdot A_c \cdot U_c \]  \hspace{1cm} (11)

Then the mass flow through the cavitating orifice can be calculated by combining eq. 1, 2 and 3 (assuming \( P_c \) equals \( P_r \)):

\[ \dot{m} = A_0 C_e \sqrt{2 \rho_f (P_f - P_r)} \]  \hspace{1cm} (12)

The mass flow of a cavitating nozzle is thus independent of backpressure, and is considered choked.

Nurick’s model was applied by Schmidt and Corradini [98], who extended the model to predict the flow velocity and effective area at the orifice exit. This was accomplished by using integral momentum and mass balances built on the assumption that wall shear may be neglected during conditions where the cavitating region is long compared to the nozzle length. This approach is called “zero wall shear” model. In contrast stands the “slug flow” exit condition, where the velocity at nozzle exit is assumed uniform over the entire orifice area (\( A_0 \)). This approach seems appropriate when the cavitating region is short compared to nozzle length. The two models are represented in Figure 25.
Figure 25. (a) Sketch of the physical process represented by the slug flow exit condition. Wall shear is assumed to be responsible for the change in velocity profile from point C to point a. (b) Sketch of the physical process represented by the zero wall shear model. The flow downstream of the cavities has low velocity and does not transfer momentum to the walls. All losses are the result of the uncontrolled expansion of the liquid into vapour (from [98]).

The models bound the range of exit momentum; the slug flow exit condition has the least possible momentum flux, and the zero wall shear assumption gives the greatest possible momentum. The exit conditions of the zero wall shear model is represented as a slug flow with an effective cross-sectional area $A_f$ and a velocity of $U_f$. Conservation of mass and momentum gives:

$$U_f = \frac{2C_c P_f - P_a + (1 - 2C_c)P_v}{C_c \sqrt{2\rho_f (P_f - P_v)}}$$

where this velocity occupies an effective area, $A_f$, given by:

$$A_f = \frac{2C^2_c (P_f - P_v)}{2C_c P_f - P_a + (1 - 2C_c)P_v} A_0$$

Since mass flow is a fixed parameter, $A_f / A_0$ equals $\bar{U}_a / U_f$. K is plotted as a function of $C_d$, $U_f / U_{Bern}$ and $A_f / A_0$ in Figure 26. It appears that at high injection pressures (compared to downstream pressure) the vena contracta measured
by $C_C$ dominates the behaviour of the nozzle. In the limit as $K$ approaches unity, both $C_D$ and $A_f/A_0$ approach $C_C$. This means that for high injection pressures the $C_D$ represents the fraction of area occupied by the liquid jet. However, for values of $K$ larger than approximately 1.8, the slug flow model appears superior.

To be able to use the previous analysis, the value of $C_C$ must be known. The value can of course be calculated using eq. 6, however in reality the value is very sensitive to nozzle geometry. Instead, $C_D$ can be measured experimentally; the $C_C$ can be calculated using eq. 10. Once $C_C$ is known, the exit momentum, effective area and effective velocity can be calculated.

![Figure 26. Predicted orifice coefficients of the exiting flow. The coefficient of contraction for these calculations is assumed to be 0.611.](image)

The nozzle velocity- and area coefficients are $C_v = U_f/U_{Bern}$ and $C_a = A_f/A_0$, where $C_D = C_C \cdot C_a$.

The above technique allows for the calculation of the orifice coefficients using the measured $C_D$ in combination with the zero wall shear model. The zero wall shear model gains credibility from measurements of exit velocities in diesel nozzles. For example Chaves et al. [69] measured $C_D$ of 0.7, however exit velocities were close to ideal Bernoulli estimate. Winklhofer et al. [66] measured exit velocities between 90 – 95% of Bernoulli estimate using LIF. The nozzle geometries were however rectangular with glasses on two sides to allow for optical access, which may increase the wall shear compared to the diesel nozzle geometry.
An alternative technique to achieve the nozzle coefficients is to use spray momentum measurements in addition to the rate measurements. This was done by, for example Siebers [99] and Desantes et al. [100]. The momentum measurements allow for a direct calculation of either the velocity- or the area-coefficient. For example, dividing the momentum flux (obtained from a pressure transducer placed at the orifice exit) with the mass flux (obtained from rate measurements), the outlet velocity is estimated. Dividing with the Bernoulli velocity, the velocity coefficient is obtained. With the discharge- and velocity coefficient determined, the area coefficient can be obtained. See example of derived orifice coefficients in Figure 27.

\[ C_{M} = C_{D} \cdot C_{V} \]

As K decreases, the velocity coefficient increases, since the effective area decreases due to increased cavitation. It can also be seen that the maximum value at low K is quite far from the ideal Bernoulli.

Figure 27. Orifice coefficients obtained from momentum and rate measurements (from [100]). The momentum coefficient is defined as

\[ C_{M} = C_{D} \cdot C_{V} \]
Derivation of the spray model

Naber and Siebers [61] have derived a spray model based on conservation of mass and momentum using integral control surface techniques. The development has two steps: (1) derivation of a relationship for the spray tip velocity, and (2) integration of the velocity relationship to obtain a correlation for tip penetration time versus penetration distance. A schematic of the model is shown in Figure 28.

Figure 28. Schematic of the spray model for the penetration correlation.

Major assumptions made in the analysis are that the spray has:

1. Uniform velocity profile.
2. Constant injection velocity with instantaneous start.
3. No velocity slip between fuel and entrained air.
4. Quasisteady flow with uniform growth rate. (i.e. constant angle $\alpha$).

The first assumption is a gross simplification of reality. The second assumption excludes conditions where the injection rate “ramps up” over a significant period of time, or does not remain constant during the injection. This is generally no problem with common rail systems, which usually has a “square” rate profile. The third assumption is doubtful near the nozzle tip, but is assumed to hold for larger distances. However, air entrainment measurements suggest an existing slip also in the spray tip, see section on air entrainment. The fourth assumption is supported
since the sprays are found to have virtually axially uniform spray angles, except at
the initial transient period. ([61], [101]).

Using the control surface, fuel mass and momentum balances can be written:
\[
\rho_f \cdot A_f(0) \cdot U_f = \rho_f \cdot A_f(x) \cdot U(x) \tag{15}
\]
\[
\rho_f \cdot A_f(0) \cdot U_f^2 = \rho_f \cdot A_f(x) \cdot U(x)^2 + \rho_a \cdot A_a(x) \cdot U(x)^2 \tag{16}
\]
where \( \rho_f \) and \( \rho_a \) are densities of injected fuel and entrained ambient air. \( U_f \) and
\( A_f(0) \) are axial velocity and cross-sectional area of the fuel at the exit of the
orifice. \( U(x), A_f(x) \) and \( A_a(x) \) are the spray velocity and the cross sectional
areas of the fuel and air at any location \( x \) in the spray.

Additionally, a third equation is added, defined as:
\[
A_a(x) = A(x) - m \cdot A_f(x) \tag{17}
\]
where the area \( A(x) \) is the total cross-sectional area of the jet at \( x \), \( m \) is a
parameter with value 1 or 0. Setting \( m \) to zero, as is done later, is the equivalent to
neglecting the cross-sectional area of the spray occupied by the fuel.

Solving eq. 15 – 17 with respect to \( U(x) \) results in:
\[
U(x) = \frac{U_f \cdot A_f(0)}{2 \cdot A(x)} \left( \frac{\rho_f}{\rho_a} - m \right) \cdot \left\{ \frac{A(x) \frac{\rho_f}{\rho_a}}{1 + 4 \cdot \frac{A_f(0) \frac{\rho_f}{\rho_a}}{\left( \frac{\rho_f}{\rho_a} - m \right)^2} - 1 \right\} \tag{18}
\]
The spray velocity in Eq. 18 applies at all \( x \) in the “model” spray. If the following
substitutions and non-dimensionalizations are made:
\[
\tilde{\rho} = \frac{\rho_f}{\rho_a} \tag{19}
\]
\[
x_0 = \frac{m \cdot d_f}{2 \cdot \tan(\alpha / 2)} \tag{20}
\]
\[
x' = x + x_0 \tag{21}
\]
\[
\frac{dx'}{dt'} = \frac{dx}{dt} = U(x) \tag{22}
\]
Appendix

\[ A(x) = \pi \cdot \left[x' \cdot \tan(\alpha / 2)\right]^2 \]  
(23)

\[ A_f(0) = \frac{\pi}{4} \cdot d_f^2 \]  
(24)

\[ x^+ = d_f \cdot \bar{\rho}^\frac{1}{2} \cdot \left(\frac{\bar{\rho} - m}{\bar{\rho}}\right) \cdot \frac{1}{a \cdot \tan(\alpha / 2)} \]  
(25)

\[ t^+ = d_f \cdot \bar{\rho}^\frac{1}{2} \cdot \left(\frac{\bar{\rho} - m}{\bar{\rho}}\right)^2 \cdot \frac{1}{U_f \cdot a \cdot \tan(\alpha / 2)} \]  
(26)

\[ \tilde{x} = x' / x^+ \]  
(27)

\[ \tilde{t} = t' / t^+ \]  
(28)

then, after some algebraic manipulation, eq. 18 simplifies to:

\[ \frac{d\tilde{x}}{d\tilde{t}} = \frac{2}{\sqrt{1 + 16\tilde{x}^2} + 1} \]  
(29)

In the above equations, the distance \( x_0 \) is the location of the orifice exit relative to the projected origin of the spray. The diameter \( d_f \) is the diameter of the fuel stream exiting the orifice. The angle \( \alpha / 2 \) is defined in Figure 28. The terms \( t' \) and \( x' \) are the coordinates referenced to the projected origin. \( t^+ \) and \( x^+ \) are scaling parameters. \( a \) is an arbitrary constant.

The final step in the derivation is to assume that \( U(x) \) is equal to the spray tip velocity and integrate eq. 29 from \( \tilde{x} = 0 \) to \( \tilde{x} = \tilde{S} \), where \( \tilde{S} = S' / x^+ \) and \( S' \) is the spray tip location relative to projected origin. The integration yields the dimensionless penetration time, \( \tilde{t} \), as a function of the dimensionless penetration distance, \( \tilde{S} \), both referenced to the projected spray origin in Figure 28.

\[ \tilde{t} = \frac{\tilde{S}}{2} + \frac{\tilde{S}}{4} \cdot \sqrt{1 + 16 \cdot \tilde{S}^2} + \frac{1}{16} \cdot \ln \left(4 \cdot \tilde{S} + \sqrt{1 + 16 \cdot \tilde{S}^2}\right) \]  
(30)

Integrating eq. 30 in the limit of \( \tilde{t} \) approaching zero (i.e. \( \tilde{x} \) approaching zero):

\[
\begin{array}{c|c}
\text{Limit} & \tilde{S} = \tilde{t} \\
\tilde{t} \to 0 & \\
\end{array}
\]  
(31)
The long time limit is derived by integrating eq. 30 in the limit of \( \tilde{t} \) approaching infinity (i.e. \( \tilde{x} \) approaching infinity):

\[
\text{Limit } \quad \tilde{t} \to \infty \quad \tilde{S} = \tilde{t}^{\frac{1}{2}}
\]  

(32)

The transition of a linear to square root dependence of penetration with time occurs in the vicinity of \( \tilde{t} = 1 \). In the transition region, the spray evolves from one dominated by the injected fluid to one dominated by entrained air.

Given the small value of \( x_0 \), (which is typically 0.5 mm) and the unphysical fact that air entrainment would occur inside the nozzle, when the projected spray origin is inside the nozzle, the fuel flow area in eq. 15 is neglected. This is done by setting \( m = 0 \) in eq. 17 – 30. The result is that the projected spray origin in Figure 28 shifts to the orifice exit plane and the injected fuel is treated as a point source of momentum.

The velocity \( U_f \) and effective orifice diameter \( d_f \) in the previous equations is determined using the following relationships:

\[
U_f = C_v \cdot \sqrt{2(P_f - P_a)/\rho_f}
\]  

(33)

\[
d_f = \sqrt{C_a \cdot d}
\]  

(34)

Equation 33 is the Bernoulli relationship for orifice exit flow, where \( P_f \) is the fluid pressure, \( P_a \) is the ambient gas pressure, \( C_v \) and \( C_a \) are the velocity and area coefficients discussed in the previous section.

The relationship between the model spray angle \( \alpha \) and the “measured” spray angle \( \theta \) obtained from spray images is defined as:

\[
\tan(\alpha) = a \cdot \tan(\theta)
\]  

(35)

A value of 0.66 of the arbitrary constant \( a \), was found to provide the best agreement between data and model in the investigation by Naber and Siebers [61]. A later investigation by Siebers et al. [102] resulted in a value of 0.75, due to a difference in how \( C_a \) was measured. In the study by Wählin and Cronhjort [103], a value of 0.71 gave best agreement. There are several other possible reasons for a deviation in this value: The pressure drop from the rail (where the injection pressure is measured) to the sac is not known, differences in how the cone angle and penetrations are estimated from spray pictures and how the pictures are thresholded.
## Abbreviations

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>CAD</td>
<td>Crank angle degree</td>
</tr>
<tr>
<td>CFD</td>
<td>Computational fluid dynamics</td>
</tr>
<tr>
<td>CO</td>
<td>Carbon oxide</td>
</tr>
<tr>
<td>DI</td>
<td>Direct injection</td>
</tr>
<tr>
<td>DPF</td>
<td>Diesel particulate filter</td>
</tr>
<tr>
<td>EGR</td>
<td>Exhaust gas recirculated</td>
</tr>
<tr>
<td>EOI</td>
<td>End of injection</td>
</tr>
<tr>
<td>GDI</td>
<td>Gasoline direct injection</td>
</tr>
<tr>
<td>H$_2$SO$_4$</td>
<td>Sulphuric acid</td>
</tr>
<tr>
<td>HC</td>
<td>Hydrocarbon</td>
</tr>
<tr>
<td>HCCI</td>
<td>Homogeneous charge compression ignition</td>
</tr>
<tr>
<td>IMEP</td>
<td>Indicated mean effective pressure</td>
</tr>
<tr>
<td>LNT</td>
<td>Lean NO$_x$ trap</td>
</tr>
<tr>
<td>MTBE</td>
<td>Methyl tertiary butyl ether</td>
</tr>
<tr>
<td>N$_2$</td>
<td>Nitrogen</td>
</tr>
<tr>
<td>NMHC</td>
<td>Non-methane hydro-carbon</td>
</tr>
<tr>
<td>NO</td>
<td>Nitrogen oxide</td>
</tr>
<tr>
<td>NO$_x$</td>
<td>Oxides of nitrogen</td>
</tr>
<tr>
<td>PAH</td>
<td>Polycyclic aromatic hydrocarbons</td>
</tr>
<tr>
<td>PCCI</td>
<td>Premixed-charge compression-ignition</td>
</tr>
<tr>
<td>PI</td>
<td>Port injection</td>
</tr>
<tr>
<td>PIV</td>
<td>Particle imaging velocimetry</td>
</tr>
<tr>
<td>PM</td>
<td>Particulate matter</td>
</tr>
<tr>
<td>SCR</td>
<td>Selective catalytic reduction</td>
</tr>
<tr>
<td>SI</td>
<td>Spark-ignited</td>
</tr>
<tr>
<td>SMD</td>
<td>Sauter mean diameter</td>
</tr>
<tr>
<td>SO$_2$</td>
<td>Sulphur dioxide</td>
</tr>
<tr>
<td>SOC</td>
<td>Start of combustion</td>
</tr>
<tr>
<td>SO$_x$</td>
<td>Oxides of sulfur</td>
</tr>
</tbody>
</table>
**Experimental Investigation of Impinging Diesel Sprays for HCCI Combustion**

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
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<tbody>
<tr>
<td>TDC</td>
<td>Top dead center</td>
</tr>
<tr>
<td>VCO</td>
<td>Valve covered orifice</td>
</tr>
<tr>
<td>VOC</td>
<td>Volatile organic compounds</td>
</tr>
<tr>
<td>VVT</td>
<td>Variable valve timing</td>
</tr>
</tbody>
</table>
NOMENCLATURE

\( A_c \)  Area at vena contracta  \\
\( A_a \)  Area of air in spay cross-section  \\
\( A_f \)  Effective area of liquid  \\
\( A_0 \)  Nozzle hole geometrical area  \\
\( C_a \)  Orifice area contraction coefficient  \\
\( C_D \)  Orifice discharge coefficient  \\
\( C_v \)  Orifice velocity coefficient  \\
\( C_M \)  Momentum coefficient  \\
\( C_c \)  Contraction coefficient  \\
\( C_{C0} \)  Contraction constant  \\
\( d \)  Nozzle hole diameter  \\
\( d_f \)  Effective orifice diameter  \\
\( l/d \)  Nozzle hole length-to-diameter ratio  \\
\( m \)  Parameter with value 1 or 0  \\
\( \dot{m} \)  Mass flow  \\
\( P_f \)  Injection pressure  \\
\( P_a \)  Ambient pressure  \\
\( P_v \)  Vapor pressure  \\
\( P_c \)  Static pressure at vena contracta  \\
\( \bar{t} \)  Dimensionless time  \\
\( t^+ \)  Time scalar  \\
\( t \)  Time  \\
\( \overline{U_a} \)  Average velocity at orifice exit  \\
\( U_f \)  Fuel axial exit velocity  \\
\( U_{Bern} \)  Ideal Velocity given by Bernoulli equation  \\
\( U \)  Spray velocity
Experimental Investigation of Impinging Diesel Sprays for HCCI Combustion

\[ U_c \] Velocity at vena contracta
\[ \tilde{x} \] Dimensionless penetration
\[ x^+ \] Penetration scalar
\[ x \] Spray penetration
\[ CN \] Cavitation number
\[ K \] Cavitation parameter
\[ a \] Constant
\[ \alpha \] Model spray angle
\[ \theta \] Spray cone angle
\[ \phi \] Nozzle impingement angle
\[ \rho_a \] Ambient density
\[ \rho_f \] Fuel density
REFERENCES

Experimental Investigation of Impinging Diesel Sprays for HCCI Combustion


Experimental Investigation of Impinging Diesel Sprays for HCCI Combustion


[89] Yamamoto H. and Niimura K. “Characteristics of fuel sprays from specially shaped and impinging flow nozzles” SAE paper No. 950082


