ABSTRACT

PADHIARY, ABHIJIT. Study of the Effect of Injection Pressure and Timing on the Combustion in an Optical Diesel Engine through Experiments and Simulations (Under the direction of Dr. Tiegang Fang).

A fully developed Bowditch type, 1.8 L heavy duty single cylinder optical diesel engine has been used to conduct experiments on Diesel fuel under varying injection timing and pressure to understand the in-cylinder combustion evolution. Numerical simulation of the same setup is performed using commercial CFD software CONVERGE to gather further data to explain the resulting trends. This thesis analyzes the fuel air mixture evolution in typical reacting engine like conditions. The study finds that increase in injection pressure enhances the fuel air mixing and thus leading to more premix type combustion. It is also found that there can be two regions of ignition depending upon the mixture formation. Furthermore, it is also found that retarding the start of injection towards top dead center leads to smoother combustion with lower rates of heat release and lower peak pressure.
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Study of the Effect of Injection Pressure and Timing on the Combustion in an Optical Diesel Engine through Experiments and Simulations

By
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A thesis submitted to the Graduate Faculty of North Carolina State University in partial fulfillment of the requirements for the degree of Master of Science

Mechanical Engineering

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APPROVED BY:

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Dr. Stephen Terry                                           Dr. Tarek Echekki

_______________________________
Dr. Tiegang Fang
Committee Chair
DEDICATION

Dedicated to Krishna

And to my brother, Tuna
BIOGRAPHY

Abhijit was interested in mathematics and engineering from a very young age. After graduating with a Bachelor’s degree in Mechanical Engineering from Birla Institute of Technology and Science - Pilani, he pursued his interest to study combustion through a Master in science degree at NC State University. He hopes to add to the understanding of combustion by pursuing a Ph.D.
ACKNOWLEDGMENTS

First, I would like to thank Dr. Fang for giving me the opportunity to work on combustion. The knowledge, skill and understanding that I have developed during this period of research is very important in my quest to understand combustion physics. He has been very patient, supportive and guided me during this project. Without his push and guidance, I would not have been able to develop the skills that helped me to carry out this work. I would like to thank Dr. Tarek Echekki and Dr. Stephen Terry for being understanding and accommodating during my thesis work. I specially thank Dr. Echekki for his lectures on fundamentals of combustion which have shaped my understanding.

I would like to take this opportunity to thank all my colleagues who have helped me with their inputs and support. Thanks a lot, Reuven, Libing, Kaushik, Fujun and Akash.

Most of all I am indebted forever to my parents and my brother for their love, faith and support for me despite the limited availability of resources. I would also like to thank God for his blessings throughout my life.
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<th>Description</th>
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<td>TDC</td>
<td>Top Dead Centre</td>
</tr>
<tr>
<td>BDC</td>
<td>Bottom Dead Centre</td>
</tr>
<tr>
<td>CI</td>
<td>Compression Ignition</td>
</tr>
<tr>
<td>BTDC</td>
<td>Before Top Dead Centre</td>
</tr>
<tr>
<td>ASOI</td>
<td>After Start Of Injection</td>
</tr>
<tr>
<td>SOI</td>
<td>Start Of Injection</td>
</tr>
<tr>
<td>ID</td>
<td>Ignition Delay</td>
</tr>
<tr>
<td>SCE</td>
<td>Single Cylinder Engine</td>
</tr>
<tr>
<td>NL</td>
<td>Natural Luminosity</td>
</tr>
<tr>
<td>LHV</td>
<td>Lower Heating Value</td>
</tr>
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CHAPTER 1: INTRODUCTION

1.1. Background

Internal combustion engines are the power house of modern society. Diesel engines have been the most preferred engine due to its higher efficiency and almost complete combustion of fuels leading to less emission of Carbon Dioxide (CO2). This is due to leaner combustion of fuels achieved through spray mixing of fuels. But in this process, they emit a large amount of other harmful pollutants like Nitrogen dioxide and particulate matter. There is a need to reduce the emissions while increasing the efficiency. Recent developments of high pressure injection systems (Common rail injection system) and turbocharging has made significant progress targeting this need. But there is still a long way to go to meet the upcoming emissions standards EPA TIER III, while still being very efficient. This motivates the researchers to gain the fundamental understanding of the spray combustion.

There is another method to reduce emissions, i.e. capturing the pollutants after they have been produced, classified as exhaust gas after treatment techniques. These include Diesel particulate filters and selective catalytic reduction techniques. But there is significant interest in reducing raw engine emissions while increasing the efficiency. Several methods like exhaust gas recirculation, high pressure injection system, multiple injections, injection rate shape alterations, combustion chamber geometry etc. are used by the research community to enhance the spray combustion in diesel engines.

Fuel injection timing and pressure play significant role in controlling the spray mixing that can lead to significant control in the air-fuel mixture formation. Air fuel mixture formation and the conditions (pressure and temperature) prevailing during the combustion can impact the heat released and emissions formed during the process. This thesis focusses on understanding the
effects of the fuel injection timing and pressure on the air fuel mixture preparation in a diesel engine through experiments and simulations.

1.2. Overview of combustion in C.I. Engines

The objective of this section is to introduce different processes in compressed ignition combustion engines and various parameters that can be used as indicators during the process. Different phases of CI combustion is shown in Figure 1.

**Figure 1: Phases of modern diesel engine, adopted from [1]**

Fuel spray is injected into the hot compressed gas near the end of compression stroke. Fuel droplets in the spray absorb the heat and evaporate to form air fuel mixture. After a relatively richer mixture is formed, ignition occurs. This period, the time between the fuel injection and ignition is called as ignition delay (ID). Generally, the fuel is injected few CAD before the TDC, to allow to the liquid fuel to mix with air which is quantified as injection timing. ID can be seen in the dip of Apparent Heat Release Rate which is because of the heat absorbed by the liquid fuel during
evaporation and mixing. During the ignition delay, near to stoichiometric fuel air mixture is formed that leads to combustion initiation. Later the combustion can be classified into two types. Because of the ignition delay some of the fuel has already mixed with the air and thus it leads to premix combustion. As the premix combustion increases the temperature, there is a thin sheet of diffusion flame is formed where liquid fuel is vaporized and diffuses into the sheet to burn. This part of combustion is called diffusion flame. This is evident from the two peaks in the heat release curve, the first one is because of the premix combustion and the second peak is because of diffusion combustion.

One of the most important aspect of the combustion in diesel engines is the mixture preparation from the spray. When high pressure spray is injected into the chamber, the liquid spray breaks up because of the instabilities and shearing of the high pressure and temperature gasses. This breaks the liquid spray consisting of ligaments like structures into droplets which are then evaporated because of the surrounding temperature, thus leading to the formation of air and fuel mixture. When the air fuel mixture reaches a suitable ratio first reaction begins. The first reactions are slow, but the onset of spontaneous combustion provides further energy for accelerating this first set of reactions for remaining bulk of the fuel. This mixture formation shapes the combustion and subsequent emissions production.

Ignition can be further classified into different stages. The first stage is a slow invisible process. The onset of rapid combustion is marked by formation of excited radicals (OH, CH etc.) which emit photons (in the visible light band) when they come back to their ground states. This can be observed during the early stages of combustion in the natural flame luminosity. In later stages this luminosity is dominated by the radiation from soot particles, which is a function of both the soot
temperature and volume fraction. Natural luminosity is an effective way to visualize combustion to interpret ignition and qualitative soot concentration.

1.3. Literature review

Effect of injection pressure on the diesel combustion was studied in an optical engine [2], which led to the conclusion that injection pressure increased the mixing and decreased the burn duration. This was even more evident with reducing impact of swirl on the chemiluminescence with the increasing injection pressure. The ignition was observed to occur near the wall.

Effect of injection pressure and timing on the spray mixing has been studied extensively among other parameters through computational models [3]. Lower injection pressure is found to have two peak distribution in terms of mixture fractions. Higher injection pressure gives a single peak of mass fraction in equivalence ratio space towards the spray tip showing higher air entrainment and mixing. It is shown that Injection timing advance can improve mixing but as we move towards TDC there is a rich distribution close to piston surface due to shorter clearance and more near the axis. Even though the model only studies non-combusting spray it gives a fair idea on the effects on injection pressure and timing on the mixture formation. In the numerical study [4] it has been found that increase in injection pressure leads to decrease in droplet size in the spray. The two peaks of droplet size distribution is also evident. Similar observation with respect to ignition delay and combustion duration is also found in [5].

In general higher injection pressure tends to have more premixing as droplets are broken into much finer diameter leading to better air fuel mixing[6]. In fact, the author mentions the need of retarding to reduce NOx emissions. Similar observations of vapor distribution (air-fuel mixture) is found in [7]. The work demonstrates increase in pressure leads to increased mixing especially in presence of wall. Similarly [8] describes the mixture formation and vapor fraction in a free jet. The
presence of wall increases the mixing, which at higher injection pressures is magnified. This observation of fuel air mixing is higher aided by interaction with the wall is supported by [9]. Similar observation in parts is also found in [10], regarding the distribution of equivalence ratio. Similar conclusions of better mixing with increase in injection pressure is reached in [11],[12].

Ignition delay and combustion duration is generally found to decrease with increase in injection pressure [6],[13]. This is attributed to reduced droplet size and increase in air entrainment at higher injection pressure [4],[14],[15]. Moreover, injection timing has crucial role to play in the mixture formation as advanced injection timing provides more time for the fuel to mix and thus showing a higher premix combustion fraction. This gives rise to higher pressure rise rate and peak pressure[3],[13],[16]. But the temperature and density in engine conditions are not constant in engine conditions. Early injections face lower density, low temperature and longer penetration. Very early injection may lead to large increase in total ignition delay [17].

Ignition location in a spray and with the effect of flat plate is studied in [18]. This shows the offset of ignition in axial direction location because of the increased mixing near the wall. It is found to be at the tip of vapor spray closer to wall due to enhanced mixing [19],[20],[21]. Natural Luminosity can be an effective way to qualitatively compare the soot formation and flame structure[22],[23]. The studies in[24],[25],[26] show that increase in pressure effectively reduces soot NL. Similar to our experiments the work in [27] use NL images to compare different modes of combustion, showing the impact of injection pressure. The paper [23] provides guidelines for interpreting NL images.

1.4. Problem Statement

Based on the literature survey there is still a need to understand the evolution of spray combustion in engine combustion with respect to important parameter: Injection pressure and
timing. In this thesis work we will study the effect of injection pressure and timing through experiments and simulations. Extensive study on the other quantitative parameters is done using simulations in commercial software package CONVERGE.
CHAPTER 2: EXPERIMENTAL SETUP

2.1. Cummins SCE 903

A Cummins VTE 903T engine, originally modifies by the US military for research purposes designated as SCE 903. The original V8 turbocharged engine was modified to a single cylinder, the details can be found in [28],[29]. All the essential components are modified for smooth running of the engine. Please note the compression ratio has changed as the piston was modified to flat top to provide optical access. For simplicity only, relevant details are provided in Table 1.

Table 1: Specifications of the optical engine

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Naturally Aspirated, Direct Injection</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>139 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>117 mm</td>
</tr>
<tr>
<td>Displacement</td>
<td>1.78 liters</td>
</tr>
<tr>
<td>Geometric Compression Ratio</td>
<td>12.9</td>
</tr>
<tr>
<td>Connecting Rod length</td>
<td>208.1 mm</td>
</tr>
<tr>
<td>Crank Arm Radius</td>
<td>58.5 mm</td>
</tr>
</tbody>
</table>

2.2. Optical access

The SCE 903 engine has a Bowditch type optical access. The Bowditch design was developed by Fred Bowditch in 1961 at GM research laboratory which utilized a window in an elongated piston to gain optical access into the combustion chamber. Extensive studies can be found in literature to generate optical data explaining spray and combustion in diesel engines.
A classical Bowditch type of optical access is made whose details can be found in the thesis of previous students who worked on this setup [30],[31]. A schematic diagram shows the optical access in the engine and the optical insert.

2.3. Drive train

The optical engine is not able to power itself because of skip fire mode and low load-based injections. An external engine was used to drive the optical engine at constant speed of 600 RPM. Details about the engine can be found in [31].

2.4. Exhaust TDC signal and shaft encoder

A shaft encoder is coupled to the crank shaft, which is used to measure the transient crank angle degrees (CAD) for the data acquisition and control system. In simple lines the encoder generates a pulse every 0.5 CAD rotation, thus forming the basic grid for data acquisition and control.

The exhaust top dead center (TDC) signal is used as reference for the injection control and data acquisition. This is generated by a single tooth disc which rotates at half the speed of crank shaft and an optical switch circuit. The single tooth is synchronized to the TDC of optical engine, such that it generates a pulse at exhaust TDC. The pulse is generated by an optical switch whenever the tooth passes through it. More details can be found in [31].

2.5. Lubrication and coolant conditioning systems

The optical engine is supplied with external lubrication and coolant conditioning systems. In order to simulate warm engine conditions, the coolant and lubrication system maintains the temperature of 170 F. Further details can be found in [31],[30].
2.6. Fuel Injection System

The stock common rail system was modified to fit a commercial common rail and a 6 nozzle Bosch injector having 160 degrees spray angle. The system composed of a low-pressure pump followed by a high-pressure pump capable to inject the fuel above 1200 bars. Further details on the fuel injection system can be found in [31],[30].

The optical engine needs to run at low loads and skip fire mode (skip a few cycles between two batches of injection events) to prevent damage to the optical windows. For this purpose a control system was designed [31]. A feedback PID control loop consisting of rail pressure sensor controlled using NI myDAQ controller was used to achieve the target rail pressures. The controller reads the current rail pressure and calculates an error based on which a signal is given to the high-pressure pump to control the flow rate to achieve the target pressure.

A fuel injection control system was developed to account for skip fire mode injection [31]. Further improvement was also made to account for parameters

1. No. of skipped cycles
2. No. of injection cycles
3. Injection timing based on CAD.

This system also removed the dependency on the pulse generator by controlling the injection solely using NI 6061 card. This was achieved by using 20 MHz clock to time perform a gated counting and pulse generation. Figure 2 shows the interface of the control system.
This control system however was not used wholly during the experimentation, but its features were included in the injection system used in experiments. More details on the exhaust TDC and CAD shaft encoder can be found in [30],[31].

2.7. Data acquisition system

In cylinder pressure is acquired through a Kistler 6067B water cooled piezoelectric pressure transducer through type ‘1919’ with two layers of metal raided shielding and Kistler 5004 charge amplifier. PCI MIO 16 E4 DAQ card is used to record the analog pressure signals referenced with exhaust TDC and CAD signals. Details of the acquisition program can be found in [31]. It was
further modified to enable to capture the time between two CAD, so as to correlate with the high-speed images, two counters from NI 6601 card. Figure 3 shows the front panel of the VI designed for this purpose.

![Figure 3: Front panel of data acquisition program](image)

### 2.8. High Speed Image acquisition system

A phantom v4.3 high speed camera with 5400 frames per second settings was used to record images during combustion. A Nikon 105 mm zoom lens with inbuilt UV filters was used with aperture settings of F11. The exposure used was 8 µs for combustion images. The image resolution used is 320X312. The camera was triggered along with the same signal that is used to send the injection pulse.
CHAPTER 3: NUMERICAL MODEL

3.1. Overview of CONVERGE

Numerical modelling of the in-cylinder physics was performed using CONVERGE version 2.4. Diesel engine includes multiple processes like spray injection, droplet evaporation, fuel air mixing, combustion, emissions modelling, heat transfer, turbulence modelling etc. The method followed here is to solve the partial differential equation for pressure, density, temperature, species concentrations, velocities and turbulence parameters for gases and liquids at each grid for the whole domain. Converge uses finite volume method to solve for gas phase properties using Navier-Stokes equation. It uses the pressure implicit splitting of operator method (PISO) algorithm for the velocity pressure coupling. A variable time step is used subject to several accuracy and stabilizing limits. Sub grid scale modelling is used in spray modelling as it length and time scales are small. Further details on CONVERGE numerical methods can be found in the manual [32]. More details about the numerical model of the diesel engine is provided in the upcoming subsections. We have used CONVERGE studio to set up the case for simulation. All the case setup description will be based on the studio.

3.2. Geometry

All the simulations were performed in a 1/6th sector geometry of the combustion chamber. Each sector has the corresponding piston bowl to accurate profile and one nozzle hole of the injector. The sector geometry is generated from the CONVERGE tool of ‘Make engine sector surface’. This tool enables to create a sector geometry that is defect free, boundary matched and periodic face matched[32]. The bowl profile is generated from the CAD drawings of the optical piston assembly. Figure 4 shows the tool window with the bowl profile for the case. Base grid size of 1mm was used because of restrictions in computational capabilities. Adaptive mesh
refinement size and sub grid cell size of 0.25 mm is used near the injector as per the recommendations [33]. The cylinder head profile is not accurately modeled because of the lack of profile information. The sector geometry used is depicted in Figure 5. Once the geometry is created we must flag boundaries and define surfaces for CONVERGE to know for further process explained in section of boundaries.
Figure 4: ‘Make engine sector surface’ tool
Swirl ratio used for the simulation is 0.8. Since the swirl ratio is not measured on this cylinder head, we have used the swirl ratio in [34], as the engine model used in their study is same as Cummins VT 903. The swirl profile used is standard profile recommended in [32]. The engine speed is 600 RPM.

Because of lack of measured data and drawings of the piston crevice and rings a standard model of crevice data is used. The effect of movement of piston rings is also included in the model. This model is then calibrated with various parameters to match the motoring curve. These parameters
include the crevice parameters, intake temperature, etc. More details on crevice parameters can be found in [32]. Application type is used as IC engines with CAD based simulation.

3.3. Physical properties and Reaction Mechanism under ‘Materials’ section

CONVERGE uses polynomial coefficients to obtain the gas properties which also can be obtained from an external therm.dat file (in NASA 7 or NASA 9 format) or a tabular format input file. A gas.dat file containing the viscosity and conductivity data is also required. A transport.dat file is also required. Since we will be using a surrogate fuel n-heptane to model Diesel 2 we will have to coerce the model to match the LHV of the original fuel. For this we use the LHV model available in CONVERGE. The LHV of n-heptane is coerced to 42.9 MJ/Kg.

The Liquid properties are set by the tabular format and Diesel 2 properties are used. A mech.dat file is used to provide the software with the reaction mechanism file that contains all the gas phase species and reaction data in it as we will be using SAGE detail chemistry model. Each reaction specified in mech.dat lists the pre-exponential factor $A_i$, the temperature exponent $b_i$, and the activation energy $E_i$ in the Arrhenius equation

$$k_{fi} = A_i T^{b_i} \exp \left( \frac{-E_{ai}}{R_u^* T} \right)$$

There is also provision to provide the third body constants for three body reactions, pressure dependent reaction constants and isomer lumping reaction reduction method. We can also define species that are not present in the mechanism file through the species.in file. Typically, it is used to specify various species (parcels, gas, liquid, solid, passives and non-transport passives). They are also used to introduce species and passives which are monitored in the output file. For our simulation we used the n-heptane skeletal mechanism available with the example of CONVERGE which is based on the mechanism in [35] with soot modelling derived from [36].
3.4. Simulation parameters

Here we specify which file to get the geometry from, what properties to solve for and how the simulation will be controlled in terms of files and solver. All the files should be present in one directory for the solver to run. A transient solver with CAD temporal resolution is used. Simulation mode is ‘Full Hydrodynamic’ with compressible gas flow solver and incompressible liquid solver. We also specify to solve momentum and energy.

Simulation time is set from -180 CAD to 180 CAD. 0 CAD specifies the compression TDC of the engine. We used a variable time step algorithm where the minimum time step will be based on the models that are active at a particular point of time[32] and user defined CFL numbers. CFL numbers define the number of cells through which a related quantity can travel in one-time step. This gives freedom to accelerate the computation of certain models while going slowly for important models. Table 2 shows the time step parameters used for this model.

<table>
<thead>
<tr>
<th>Table 2: Time-step control parameter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Initial Time step</td>
</tr>
<tr>
<td>Minimum Time step</td>
</tr>
<tr>
<td>Maximum Time step</td>
</tr>
<tr>
<td>Maximum Convection CFL limit</td>
</tr>
<tr>
<td>Maximum Diffusion CFL limit</td>
</tr>
<tr>
<td>Maximum Mach CFL limit</td>
</tr>
<tr>
<td>Droplet motion time step control multiple</td>
</tr>
<tr>
<td>Chemical Time step Control Multiple</td>
</tr>
<tr>
<td>Collison grid time step multiple</td>
</tr>
<tr>
<td>Moving Boundary time step multiple</td>
</tr>
</tbody>
</table>
Because of our computational limitations we have used recommended values for the solver parameters[32]. We have used strict conserve for all scalars and passives.

3.5. Boundary Conditions

As the name suggests in this setup we give the boundary conditions to the partial differential equations that will be solved. All the boundaries are given the temperature of coolant i.e. 170 F ~ 350 K.

I. Piston

Piston is given as ‘WALL’ boundary with translating motion specified as ‘piston motion’ in the case. The law of wall is used for the temperature boundary condition. It is generally used when turbulent boundary layer is not sufficient [32].

II. Front and Back Face

They are given ‘PERIODIC’ boundary as it allows us simulate only a sector of engine geometry[32].

III. Cylinder wall and cylinder head

They are given as stationary ‘WALL’ boundary and law of wall temperature boundary condition.

3.6. Initial conditions and events

In this section we can give initial conditions to the in-cylinder volume. Note, we are conducting numerical study in only the in-cylinder volume with the outside boundaries with wall boundary conditions. Since the effect of turbulence was not our key focus, we used typical values for the simulation as prescribed by CONVERGE (Turbulent Kinetic Energy (TKE) = 62.0271\(m^2/s^2\) and Turbulent Dissipation = 17183.4\(m^2/s^3\)). Here we set the intake pressure at the end of intake stroke as 0.9 bar, while the intake temperature was varied during the motoring match. The intake
temperature then used is 308 K (for 800 bar case). Further we can specify the intake air composition which also includes pollutant scalar compositions. We set the intake air composition as 79% Nitrogen and 21% Oxygen and specify the NOx scalar and soot scalar as 0.

3.7. Physical Models

In this section we specify which models to incorporate in our simulations. CONVERGE has a variety of models (which are based on various sub models). These include Spray, Combustion, emissions, turbulence, VOF, Fluid structure interaction etc. For our case we only used Spray, combustion, emissions and turbulence modelling. We will go through in details to explain our model. These details are very necessary for reproducing our results.

3.7.1. Spray Modelling

This section is explained in CONVERGE manual as ‘Discrete Phase Modelling’. They are in spray.in file. To simulate the spray in which liquid at high pressure is forced through nozzle leading to formation of droplets, CONVERGE uses parcels which are identical droplets (radius, composition, temperature etc.) to be placed at the injector nozzle location. The parcels statistically represent the spray by using different models to predict or given as input by the user. This is computationally very efficient. For simplicity we will only explain the models used for our case. Other models and their description can be found in [32].

Liquid injection can be done through nozzles and injectors. Injector is basically a group of nozzles. The nozzles and injectors are embedded units which will be described in grid control subsection a chosen compared other advanced models owing to limited computational resources. Boiling model is not activated for similar reasons. Once the droplets are evaporated our surrogate fuel i.e. n-heptane must replace the diesel 2. This can be set by specifying the evaporated species as n-heptane. Please note that the n-heptane should be specified as per represented in the reaction
mechanism file. To calculate the Spray penetration, we used 0.99 of the liquid mass for liquid spray penetration and 0.001 of vapor mass for fuel vapor penetration. The mass diffusivity constants are used that of diesel 2 with 1.0 scaling constants.

The Collison model used is NTC Collison model which is faster and more accurate than the only other Collison model available in CONVERGE (O’Rourke model)[37]. Because of computational limitations we have used less complex wall interaction (just rebound and slide) model. There was no available rate shape data on our injector, so we used existing rate shape profile which is almost a square rate shape, typical for the commercial injector. This profile is scaled based on the injection duration and the injection mass for the simulation. There may be variations in the simulation data with different injection rate shape profile. The rate shape profile is shown in the figure 6. A KH-RT breakup model is used in this case. Details about the model can be found in [32]. The parameters used are that of recommended for diesel injectors. They are also presented in Table 2. The injection timing and quantity was varied on the various cases which will be discussed the section cases setup. 1/6th of the total injection quantity was used since we are simulating a sector of the engine. The nozzle diameter used was 152 micrometer, spray cone angle 9 degrees (measured). The injection duration was calculated using the spray tool where max pressure, injection quantity and coefficient of discharge was used as input.
Figure 6: Rate Shape profile of the injector

Table 3: KH and RT parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>KH Shed Constant</td>
<td>1.0</td>
</tr>
<tr>
<td>KH Model Size constant</td>
<td>0.6</td>
</tr>
<tr>
<td>KH Model velocity constant</td>
<td>0.188</td>
</tr>
<tr>
<td>KH Model breakup time constant</td>
<td>7.0</td>
</tr>
<tr>
<td>RT Break up time constant</td>
<td>1.0</td>
</tr>
<tr>
<td>RT Model size constant</td>
<td>0.1</td>
</tr>
<tr>
<td>Coefficient of discharge</td>
<td>0.86</td>
</tr>
</tbody>
</table>

3.7.2. Combustion Modeling

The combustion model is activated from 10 CAD BTDC (0 specifies TDC) to 80 CAD. Since it was observed in experiments that the combustion stops much before 80 CAD [31]. Further combustion temperature cutoff is 600K and minimum HC mole fraction is 1e-8. This means that the combustion model will switch off below temperature of 600K and HC mole fraction of 1e-8. SAGE detail chemistry solver with constant volume calculations is used for the combustion model.
In order to accelerate the computation we have used the option for analytical Jacobian and option to resolve rate constants only if temperature increases by 2K besides adaptive zoning (temperature bin size 5.0K, reaction ratio bin size 0.05) [32].

3.7.3. Turbulence modeling

For this case RANS RNG turbulence model is used with typical coefficients[32]. The focus of this work was not to account for the effects of various turbulence model so, we preferred to use less computationally expensive turbulence model. A heat transfer model based on [38] is used.

3.7.4. Grid Control

Base grid size used was 0.001 m. CONVERGE has the option to incorporate adaptive mesh refinement to resolve detail physics when there is a need to. A sub grid criterion level makes sure that the change in that property is resolved to maximum of this level in one cell. If required, this helps to add more cells till the change in one cell is below this level. Embedding level ensures how low the cell sizes can be. Velocity uses 2m/s sub grid criterion level with embedding of scale 2 (1/4 of cell size) throughout the simulation. Temperature uses sub grid criterion as 5 K and embedding scale of 2 but it is only activated during the combustion event. An embedding scale of 2 is used for the nozzle. This makes the smallest grid sizes to be 0.00025 m (0.25 mm) for the fixed embedding i.e. nozzle. A grid convergence study conducted in [39] finds this grid size to be grid convergent with SAGE model. For the piston and cylinder head a scale of 1 is used.

3.7.5. Output processing

In this section we can control what we can write to our output files. It comes at the expense of computational time. For this case we monitored various species and used heat transfer output files. We used a 2 CAD duration for writing 3D data files and 1 CAD duration for writing property data file. A restart file was saved every 10 CAD.
The Case setup was then solved in parallel processing with 8 cores of CPU. It took approximately 60 hours for one simulation run. The simulation results are very close to experimental results.
CHAPTER 4: RESULTS AND DISCUSSIONS

4.1. Design of Experiments

The experiments were designed to study the effect of injection pressure with three design points. The timing was studied with two design points for each injection pressure. The injection quantity was approximately constant for all the cases. Table 4 shows the experiments design very well.

Table 4: Design of Experiments

<table>
<thead>
<tr>
<th>Case</th>
<th>Injection Pressure</th>
<th>Injection Timing</th>
<th>Injection Quantity</th>
<th>CAD duration</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>800 bars</td>
<td>10 CAD BTDC</td>
<td>46.29 mg (in experiment)</td>
<td>6.22</td>
</tr>
<tr>
<td>2</td>
<td>1000 bars</td>
<td>10 CAD BTDC</td>
<td>46.26 mg (in experiment)</td>
<td>5.56</td>
</tr>
<tr>
<td>3</td>
<td>1200 bars</td>
<td>10 CAD BTDC</td>
<td>46.22 mg (in experiment)</td>
<td>5.08</td>
</tr>
<tr>
<td>4</td>
<td>5 CAD BTDC</td>
<td>5 CAD BTDC</td>
<td>7.715*6 mg (in simulation)</td>
<td></td>
</tr>
<tr>
<td>5</td>
<td>5 CAD BTDC</td>
<td>7.71*6 mg (in simulation)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>6</td>
<td>5 CAD BTDC</td>
<td>7.71*6 mg (in simulation)</td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

The figure-7 shows the injection pressure and total injection mass used in the simulations. The DAQ system records the In-cylinder pressure referenced with 0.5 CAD. Simultaneously we use high speed camera to record the images. Further the data processing is done using EXCEL and MATLAB to understand the different parameters.
4.2. Data Analysis Methods:

4.2.1. In-cylinder Volume

The volume of the cylinder (V) used for experimental data analysis was calculated using the equation given in [40]

$$\frac{V}{V_c} = 1 + \frac{1}{2} \times (r_c - 1) \times [R + 1 - \cos \theta - (R^2 - \sin^2 \theta)^{0.5}]$$  \hspace{1cm} (2)$$

Where, \(V_c = \text{Clearance Volume} \ (149.16 \ \text{cc})$$

\(r_c = \text{compression ratio}\)

\(R = \frac{\text{Connecting rod length}}{\text{crank radius}}\)

4.2.2. Pressure referencing

Pressure pegging with respect to intake manifold pressure is used to reference the relative pressure data recorded from the Kistler pressure sensor [31]. Average pressure during the intake is used to coerce the pressure measured at the end of intake stroke to generate absolute pressure.
data. Further to remove noise from the pressure data it is filtered with a seven-point moving average.

4.2.3. Apparent Heat Release (AHR) and Mass Burned Fraction (MBF)

During the combustion when fuel burns heat is released which is transferred to work into the piston. Of the total heat released a part of it is lost as heat to the walls. Remaining heat shows up in the work done by the pressure trace. AHR($Q_n$) can be calculated with realistic assumptions by[40]

\[
\frac{dQ_n}{dt} = \left(\frac{\gamma}{\gamma-1} \times p \times \frac{dv}{dt} \right) + \left(\frac{1}{\gamma-1} \times V \times \frac{dp}{dt}\right)
\]

(3)

Where, \( \gamma \) is the ratio of specific heat \( \frac{C_p}{C_v} \) (1.325 used in our calculations). \( \frac{dQ_n}{dt} \) Can be integrated to find the total heat released (THR) during the cycle. Based on AHR a mass fraction burned fraction can be calculated. This can be analyzed to get a qualitative idea of progress of combustion in the cylinder.

\[
MFB(X_b) = \frac{AHR(CAD)}{AHR(Total)}
\]

(4)

The heat transfer losses to the wall, crevice losses is assumed to have similar trends in all cases so that it will not have significant impact on the trends recognized in this thesis.

4.2.4. Burn Duration (CA 10, CA 50, CA 90)

Burn duration is defined as the duration (in CAD) that is taken to burn a specific amount of fuel. CA 10 signifies the duration (in CAD) taken to burn 10% of the total amount of fuel. Similarly, CA 50, CA 90 define the duration taken to burn 50% and 90% of the fuel. This is calculated from the heat release curve.
4.2.5. Ignition Delay (ID)

Ignition delay in this work is measured using the Apparent Heat release curves (AHR) where it is quantified by the duration taken after the fuel injection to show the first sign of reaction[1],[40]. We use AHR curve to calculate the duration in CAD between the start of injection and when AHR curve becomes positive signifying that heat release has already started. Luminous delay is analogous to ID but it is defined as the duration between the start of injection and first visible light showing heat release reactions[19]. But because of inability of our setup to cover wide viewing area we were unable to calculate luminous delay effectively. We will be using the ID calculated from AHR for our analysis.

4.2.6. NL Combustion images

NL combustion images as discussed earlier can give us a fair idea of the onset of combustion and soot volume and temperature. Even though our optical access is not able to capture all the flame structure, but the combustion images add up to the other parameters to explain the trends. We have further processed the images to generate a good comparison with other experimental and simulation data to explain the structure of the flame. For example, the figure 8 shows a sample image, which shows some extremely luminous regions (bright red in the processed image) which can be qualitatively interpreted as sooty flame[11],[24]. For showing the effectiveness in using processed image we have also shown unprocessed image in Figure 9. Moreover, we have used representative cases to show our observations. The Number on the figure represents CAD corresponding to the NL image.
4.2.7. Simulation Snapshots

Simulation results are presented to show the evolution of combustion in all of the 6 cases. The pictures are made translucent to show the evolution along the same line of sight as experiments. A side view of the middle plane is also provided to further strengthen such kind of representation. The semicircle indicates the extent of viewing area available through the experiments. OH concentration and temperature are meant to show the flame and its evolution. Equivalence ratio is intended to indicate more details on the prevailing conditions of flame. They can also be used for qualitative comparison with the NL images obtained through experiments.
4.2.8. Mass fractions vs Equivalence ratio at different CAD

We have grouped cells with a close range of equivalence ratio and computed the mass fraction of the group. Such a mass fraction can give us an idea of the equivalence ratio distribution at various CAD. Such a plot provides us with snapshots of premixing.

4.3. Effect of Injection Pressure:

To study the effect of injection pressure we kept the fuel mass, injection timing and all other parameters constant but varied the injection pressure. SOI for two set of cases are 350 CAD (10 CAD BTDC) and 355 CAD (5 CAD BTDC). Figure 10 shows the Pressure curves for 350 CAD injection (10 CAD BTDC). Due to lack of ambient conditioning system we were not able to keep the intake temperature at a constant value but since the experiments are conducted one after another, the intake temperatures are very close to each other (308 for 800 bar cases to 315 K for 1200 bar). We expect this will not impact the trends we are trying to observe.

The red circle in Figure 10 signifies the SOI, after which we will compare the NL images with simulations in terms of structure of flames. In both the SOI cases, the peak pressure rises with the injection pressure. This is because of the heat release rates (shown in figure 12), being higher with increase in injection pressure. Increase in injection pressure may help in breaking of the liquid spray and thus leading to greater mixing. This is evident from the temporal profiles of mass fractions vs equivalence ratio in figure 20 and 21. Indeed increase in pressure moves the peak mass fraction towards richer equivalence ratio signifying increase in mixing. Further this peak value increases which is another indicator of increased mixing. Figure 13 shows the mass fraction burned (MFB) during the combustion based on apparent heat release rates. It clearly shows that with increase in injection pressure the MFB of fuel becomes faster. There can be seen a flatness in the experimental MFB curve, which may be extreme leaning of combustion due to significant
portion of fuel consumed during premix burning at earlier stages. One interesting observation is that n-heptane when used as a surrogate fuel represents diesel fuel very well in premixed combustion stage. The increased mixing with increase in injection pressure may be explained by ignition delay, shown in Figure 14 which decreases with increase in injection pressure. Further, the duration of combustion also reduces with increase in injection pressure. CA 50, CA 90 follow a decreasing trend with increase in injection pressure, shown in figure 15. Figures from simulations suggest that most of combustion happens in premix stages as equivalence ratio is drawn closer to 1 before any rise of temperature is seen. This is in line with the single peak heat release rate for all the cases. The NL images in figure 24-figure 29, suggest that stronger flame development for higher injection pressure cases. Considering the narrow view of the window, simulation snapshots will be used to explain the ignition and combustion development. We will be using the OH concentration to mark the point of ignition and along with temperatures (above 2400 K) to mark the flame[1]. In all of the cases, first ignition is found near the wall where spray is mixed well. This cannot be observed in the experiments as the view is narrow. The NL images for higher injection pressure case indicate faster flame development. Spray mixing near the wall is the key phenomenon that can be seen during the study. Because of which we can also see at least two position of auto ignition: One in the squish region and another in the bowl region. This can be seen in first trace of OH concentration rise, at 4-6 CAD (Figures 30,31,37,38,44,45,52,53,58,59,64,65). We find, there is rich mixture found upstream of the wall. Further since there is some mixture escaping outside the bowl region into the squish region they form rich mixtures in the squish region and thus form the second ignition point. The structures of the flame seen in NL images match well to the structures in terms of temperatures observed from simulation results. (Note: The NL flame
structures include chemiluminescence from OH and soot radiation, which depends on soot temperature).

4.4. Effect of injection timing:

Simulations and experiments are carried out to understand the influence of injection timing for two different cases 10 CAD BTDC and 5 CAD BTDC. Earlier injection timing gives enough time to the fuel to mix thus leading to larger premix fractions and peak pressures. This is because of faster heat release rates. At 5 CAD BTDC SOI cases leaning of the mixture is not seen as in 10 CAD BTDC SOI injection cases. Because of absence of leaning due to large premixing of charge there is smooth combustion (evident in mass burnt fraction). This may result in low heat transfer losses thus leading to more apparent heat release. Most of combustion in 5 CAD BTDC SOI occurs outside our view in the squish region. The cases with SOI 5 CAD BTDC have slightly larger burn duration with lower peak pressures and heat release rates.

The distribution of the equivalence ratio has a tendency to for two peaks at the retarded SOI. This can be confirmed in figure 20 and 21. Further since premixing is less in retarded injection cases the temperatures achieved in the mixture are less.
Figure 10: Apparent Heat Release Rates vs CAD curve for cases with SOI 10 CAD BTDC

Figure 11: Apparent Heat Release Rates vs CAD curve for cases with SOI 10 CAD BTDC
Figure 12: Apparent Heat Release vs CAD for cases with SOI 10 CAD BTDC

Figure 13: Mass Burnt Fraction vs CAD for cases with SOI 10 CAD BTDC
Figure 14: Ignition Delay vs CAD

Figure 15: Burn Duration vs CAD
Figure 16: Pressure vs CAD curve for cases with SOI 5 CAD BTDC

Figure 17: Apparent heat release rates vs CAD curve for cases with SOI 5 CAD BTDC
Figure 18: Apparent heat release vs CAD curve for cases with SOI 5 CAD BTDC

Figure 19: Mass Burn Fraction vs CAD curve for cases with SOI 5 CAD BTDC
Figure 20: Equivalence Ratio distribution snapshots in CAD for cases with SOI 10CAD BTDC
Figure 21: Equivalence Ratio distribution snapshots in CAD for cases with SOI 5CAD BTDC
Figure 22: Mean Temperature with CAD for SOI 10 CAD BTDC

Figure 23: Mean Temperature with CAD for cases with SOI 5 CAD BTDC
Figure 24: Natural Luminosity Images for Injection Pressure 800 bar, SOI 10 CAD BTDC
Figure 25: Natural Luminosity Images for Injection Pressure 1000 bar, SOI 10 CAD BTDC
Figure 26: Natural Luminosity Images for Injection Pressure 1200 bar, SOI 10 CAD BTDC
Figure 27: Natural Luminosity Images for Injection Pressure 800 bar, SOI 5 CAD BTDC
Figure 28: Natural Luminosity Images for Injection Pressure 1000 bar, SOI 5 CAD BTDC
Figure 29: Natural Luminosity Images for Injection Pressure 1200 bar, SOI 5 CAD BTDC
Figure 30: Combustion Development in case of 800 bar, SOI 10 CAD BTDC-1
Figure 31: Combustion Development in case of 800 bar, SOI 10 CAD BTDC-2
Figure 32: Combustion Development in case of 800 bar, SOI 10 CAD BTDC-3
Figure 33: Combustion Development in case of 800 bar, SOI 10 CAD BTDC-4
Figure 34: Combustion Development in case of 800 bar, SOI 10 CAD BTDC-5
Figure 35: Combustion Development in case of 800 bar, SOI 10 CAD BTDC-6
Figure 36: Combustion Development in case of 800 bar, SOI 10 CAD BTDC-7

40 CAD

50 CAD
Figure 37: Combustion Development in case of 1000 bar, SOI 10 CAD BTDC-1
Figure 38: Combustion Development in case of 1000 bar, SOI 10 CAD BTDC-2
Figure 39: Combustion Development in case of 1000 bar, SOI 10 CAD BTDC-3
Figure 40: Combustion Development in case of 1000 bar, SOI 10 CAD BTDC-4
Figure 41: Combustion Development in case of 1000 bar, SOI 10 CAD BTDC-5
Figure 42: Combustion Development in case of 1000 bar, SOI 10 CAD BTDC-6
Figure 43: Combustion Development in case of 1000 bar, SOI 10 CAD BTDC-7
Figure 44: Combustion Development in case of 1200 bar, SOI 10 CAD BTDC-1
Figure 45: Combustion Development in case of 1200 bar, SOI 10 CAD BTDC-2
Figure 46: Combustion Development in case of 1200 bar, SOI 10 CAD BTDC-3
Figure 47: Combustion Development in case of 1200 bar, SOI 10 CAD BTDC-4
Figure 48: Combustion Development in case of 1200 bar, SOI 10 CAD BTDC-5
Figure 49: Combustion Development in case of 1200 bar, SOI 10 CAD BTDC-6
Figure 50: Combustion Development in case of 1200 bar, SOI 10 CAD BTDC-7
Figure 51: Combustion Development in case of 800 bar, SOI 5 CAD BTDC-1
Figure 52: Combustion Development in case of 800 bar, SOI 5 CAD BTDC-2
Figure 53: Combustion Development in case of 800 bar, SOI 5 CAD BTDC-3
Figure 54: Combustion Development in case of 800 bar, SOI 5 CAD BTDC-4
Figure 55: Combustion Development in case of 800 bar, SOI 5 CAD BTDC-5
Figure 56: Combustion Development in case of 800 bar, SOI 5 CAD BTDC-6
Figure 57: Combustion Development in case of 1000 bar, SOI 5 CAD BTDC-1
Figure 58: Combustion Development in case of 1000 bar, SOI 5 CAD BTDC-2
Figure 59: Combustion Development in case of 1000 bar, SOI 5 CAD BTDC-3
Figure 60: Combustion Development in case of 1000 bar, SOI 5 CAD BTDC-4
Figure 61: Combustion Development in case of 1000 bar, SOI 5 CAD BTDC-5
Figure 62: Combustion Development in case of 1000 bar, SOI 5 CAD BTDC-6
Figure 63: Combustion Development in case of 1200 bar, SOI 5 CAD BTDC-1
Figure 64: Combustion Development in case of 1200 bar, SOI 5 CAD BTDC-2
Figure 65: Combustion Development in case of 1200 bar, SOI 5 CAD BTDC-3

12 CAD

14 CAD
Figure 66: Combustion Development in case of 1200 bar, SOI 5 CAD BTDC-4
Figure 67: Combustion Development in case of 1200 bar, SOI 5 CAD BTDC-5
Figure 68: Combustion Development in case of 1200 bar, SOI 5 CAD BTDC-6
CHAPTER 5: CONCLUSION

Experiments and simulations were conducted with diesel for 3 cases of different pressure and 2 cases of injection timing with the aim to understand the evolution of combustion via experimental and numerical approaches. Numerical experiments were seen to closely follow experimental results showing important trends of combustion in diesel fuel. The conclusions are as follows.

1. Ignition is found to occur in the vicinity of wall where the spray interacts with the wall with enhanced mixing. Our experimental setup is not able to capture the ignition and combustion at early stages due to narrow view. But our simulation results show the point of combustion and its evolution.

2. Increasing the injection pressure increases the spray break up and mixing. Thus resulting in a faster and more premix type of combustion. This led to larger heat release rates and peak pressures. Equivalence ratio profiles obtained show enhanced mixing in a reacting spray with increase in injection pressures.

3. Injection timing close to TDC leads to gradual combustion (lower heat release rates and slightly larger burning duration). Further the tendency to form two peaks in equivalence ratio distribution is seen with retardation of SOI at lower injection pressures.
REFERENCES


[27] S. Han, J. Kim, and C. Bae, “Effect of air–fuel mixing quality on characteristics of


