ABSTRACT

JAWALKAR, UDDHAVA ANIL. Development of a 1.8L Heavy-Duty Optical Engine Test-Bed and Diesel Combustion Studies utilizing High-Speed Visualization. (Under the direction of Dr. Tiegang Fang).

A fully developed Bowditch type, heavy-duty optical diesel engine test-bed is utilized to conduct a comprehensive diesel combustion study employing high-speed imaging techniques. Predictions of global climate change have pushed stringent regulations on internal combustion engines to reduce emissions such as NOx, CO and particulate matter which are the driving force for research in this area. Optical Engines have proven to be successful in aiding engineers to study intricate details of combustion process and enable reduction in emissions. Newer combustion strategies for diesel engines utilizing alternative fuels and fuel blends have tremendous potential to fulfill today’s energy demand while adhering to EPA regulations. The optical engine test bed proves to be an ideal starting point to explore these possibilities. In this study the baseline combustion characteristics for SCE-903 optical engine test bed are established. Reliable and repeatable data acquisition systems, fuel control systems and combustion imaging techniques are devised. The results validate the proficiency the complete test bed and provide insight about diesel combustion behavior in an optical engine under various fuel injection pressure and timings.
Development of a 1.8L Heavy-Duty Optical Engine Test-Bed and Diesel Combustion Studies utilizing High Speed Visualization

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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Member of Advisory Committee
DEDICATION

Dedicated to my family especially my mother and sister. Their sacrifices and unwavering support throughout my life has made me what I am today.
BIOGRAPHY

Born in Pune, India in 1990, Uddhava always had a fascination with internal combustion engines from a very young age. Working on cars, and engines with his grandfather irked his curiosity for how things worked. He began his engineering education in 2009 at University of Pune where he graduated with a B.E. degree in Mechanical Engineering. During his bachelor’s degree he undertook endeavors like SAE BAJA and FSAE fueled his passion for mechanical design, validation and testing. Feeling somewhat ambitious, he decided to pursue a Master’s degree in Mechanical Engineering from North Carolina State University in 2014 in the area of engine research and development.
ACKNOWLEDGMENTS

I would like to express my gratitude towards my adviser Dr. Tiegang Fang for giving me this great opportunity to work on an optical engine, and for his words of encouragement during difficult situations. I have gained invaluable experience and knowledge under Dr. Fang’s guidance. I would also like to thank members of my committee, Dr. Richard Gould and Dr. Stephen Terry for their patience and continued support for me.

I would also like to thank my fellow researchers, Aaron McCullough for his hard work in building the initial components of the Optical Engine Test bed, my work would not have possible without it. Shawn Kim, Reuven Gomes, Ashutosh and Subhash Sriram for their invaluable inputs in the lab. It couldn’t have been possible to complete this without their assistance. Furthermore I would like to give a special thanks to Kyle Jonson for being a great friend and for his expert advice in putting together the test-bed. I’d like to additionally thank all of my fellow students under Dr. Fang. Each of them has been great to work with and they have always been eager to help with anything I may need. I would also like to thank Gary Lofton and Steve Cameron for all of the parts they have fabricated and machined for me even at short notice and also for their expert advice, I could not have been able to solve many issue without their support. The optical insert components are all Mr. Cameron’s craftsmanship. Most importantly I would like to thank the Chancellor’s Innovation Fund for supporting this project.

I would also like to acknowledge my family for supporting my dream to pursue graduate education in the U.S.A. despite their delicate health and my friends for supporting me through thick and thin and for understanding me so well.
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CHAPTER 1: INTRODUCTION

In the advent of alarming events like global climate change and global warming, the emission norms for internal combustion engines are progressively being tightened. These stringent emission norms have pushed for highly efficient engines, which deliver maximum performance demanded by the end-user while minimizing its carbon footprint. Diesel engines are able to offer both these features due to their inherent benefits of higher thermal efficiency and better reliability. But with these advantages, the pitfalls of exhaust emission that include NOx and particulate matter are a liability, especially in the face of progressively stricter EPA emission norms quantified in the graph below [9].

Figure 1: EPA Diesel Emission Standards
Diesel Engines have always been popular in the heavy-duty transportation industry and provide great possibilities for further development of eco-friendly engines. Although to tap into this resource, it was important to devise methods to better understand the process of combustion which is the most fundamental part of an internal combustion engine. Researchers have foraged to understand in-cylinder combustion process for over a century and yet many parts of this process are not fully understood. Complicated computational models are limited by their ability to account for the highly unsteady and turbulent nature of in-cylinder combustion. Often these models consume huge amounts of valuable computational time and resources. In an attempt to get quick results and reduce the disparity between the actual combustion processes and the assessments of a combustion process, experimental engines with optical access to the combustion chamber were developed since the 1930’s [2]. These powerful tools empowered engineers and researchers to be able to visually analyze the combustion process in great detail.

Various designs of optically accessible engines have been proposed since then, but the Bowditch design for optical engine has proven to be widely successful owing to simple yet robust design and extensive optical access to the combustion chamber. A Bowditch type design first developed in 1961 at General Motors Research Laboratories by Fred Bowditch, utilizes a window installed in an elongated piston crown. This piston-cylinder assembly is then extended and installed such that the piston oscillates around a stationary mirror which grants optical access through the piston crown window. The design can be better understood from Figure 2b borrowed from reference 2.
Figure 2: Bowditch type optical Engine

The original Bowditch editorial extensively discusses the various operating limitations and structural considerations for the design of such an optical engine. The most notable limitations are window fouling due to lubricating oil, lower firing rate due to optical parts and so on which shall be discussed in further detail subsequently.

The heavy duty optical engine utilized in development of the test-bed described in this thesis applies the design principles proposed by Bowditch. A brief description of the design
and development considerations is provided in chapter 2 and was published in reference 1. Originally modified from a Cummins marine engine for research purposes by Cummins it was designated as SCE-903 and employed for two known research studies. The first publication was by a research group at Rutgers University in New Jersey in late 1980’s titled “Instantaneous Heat Transfer over the Piston of a Motored Direct-Injection Type Diesel Engine” [4]. This study employed 10 thermocouples to gather instantaneous heat flux from 1/7th of the surface of the firing piston, while the engine was motored at varying engines speeds and intake pressures. It concluded that although the instantaneous heat flux data could be used for building heat transfer prediction models, it could not be used to verify heat transfer correlations with variation of location especially in a fired engine.

A second study utilizing the SCE-903 was for evaluating high temperature lubricants conducted by the United States Army for a period of three years until 1993. The research was aimed at developing novel lubricants for use in future low heat rejection (LHR) engines by simulating the SCE-903 as a LHR. After testing six lubricants it was concluded that test oil designated as ‘oil D’ successfully completed 200 hours of testing at 275° C without significant degradation or wearing of parts. [3]

Over the years various different optical engines utilizing the Bowditch design in addition to other optical accesses have been developed. It is useful to discuss the type of research carried out with these engine to establish a better understanding of the current work. It shall help provide insight to researchers at North Carolina State University about future possibilities for the SCE-903 Optical Engine Test-Bed.
One such Bowditch optical engine depicted in Figure 3 was utilized by researchers at Tianjin University, China in 2015 to understand the effects of physical and chemical properties of fuels on combustion in optical diesel engine. [12]

Figure 3: Optical Engine at Tianjin University

Liu et. al. [12] utilized a small 0.664 liter, Bowditch type optical diesel engine to conduct a comparative combustion study between diesel and two blends of diesel fuel. Butanol20 and DMF20 were two diesel blends which contained N-Butanol and DMF (2,5-dimethylfuran) oxygenated biofuels blended in a 20% volume fraction respectively. They also
varied EGR from zero to ~60% covering both conventional and low temperature combustion. Their results indicate that use of bio-oxygenated blends generates predominantly blue flames in the earlier phases of combustion and reduced soot luminosity. It is hence concluded that, while both fuels result in lower soot emissions as compared to diesel, the lower cetane number of DMF causes highest ignition delay, and hence lowest soot emission.

In France another Bowditch type optical diesel engine depicted in Figure 4 is available at the French Institute of Petroleum, abbreviated as IFP for the French name Institut Français du Pétrole. It is a 499cc single cylinder, light duty DI diesel with 15.6 compression ratio.

![Figure 4: IFP Optical Engine Set-up](image-url)
An article titled “Optical Investigation of Dual-fuel CNG/Diesel Combustion Strategies to Reduce CO2 Emissions” was published employing the IFP optical Engine in 2014 [10]. In an attempt to reduce CO₂, research for dual-fuel combustion strategies using compressed natural gas (CNG) and diesel fuel in the IFP optical engine was presented by Dronniou, N et. al. [10] Experiments for wide equivalence ratios were performed by supplying a premixed mixture of CNG, air and nitrogen while performing direct injection of diesel surrogate fuel that is a high reactivity, low sooting fuel blend. Optical diagnostics techniques employed were, high speed imaging of combustion luminosity to understand dynamic combustion events, and OH* chemiluminescence imaging. 2D planar laser-induced fluorescence (2D PLIF) was also performed to map fuel distribution at dual-fuel ignition. The study concluded that at low equivalence ratio combustion progressed slowly from near-wall region to the central part, which resulted in an increase in unburned hydrocarbons (UHC). Whereas at high equivalence ratios, backward flame propagation beginning in the near-wall region was observed in conjunction with mixture mixing mechanisms that aided the flame propagation. Dronniou et. al. [10] predicted that these remarkable flame propagation mechanisms would greatly reduce if not eliminate knocking in dual-fuel engines with high compression ratios running at high loads. These findings could open up new avenues for high power density engines which also can also be fuel efficient and environmentally friendly.

The optical engines discussed in these articles vary in size and optical access geometry greatly from the SCE-903. Although they are smaller in size, the research conducted employing them could certainly be replicated for a heavy duty engine like the SCE-903.
A heavy duty optical engine closer to SCE-903 in terms of size is located at Sandia National Laboratories’ Combustion Research Facility (CRF) in Livermore, California. Developed by Dr. Espey and Dr. Dec in 1993, this state of the art facility has empowered copious amounts of publications in the past two decades. The optical access engine at CRF is a single cylinder, direct injection, 4 stroke diesel engine with dimension listed in Table 1 and compared with the SCE-903 optical Engine.

<table>
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<tr>
<th>Table 1: Geometrical dimensions of N-14 and SCE-903 Optical Engines [11]</th>
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<tr>
<td><strong>Engine Type</strong></td>
</tr>
<tr>
<td>Combusion chamber</td>
</tr>
<tr>
<td>Number intake valves</td>
</tr>
<tr>
<td>Number of exhaust valves</td>
</tr>
<tr>
<td>Bore</td>
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<tr>
<td>Stroke</td>
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<tr>
<td>Displacement</td>
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<td>Compression Ratio</td>
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* In this optically accessible diesel engine, one of the two exhaust valves of the production cylinder head was replaced by a window and periscope.

It is based on a Cummins N-series heavy duty production engine modified to grant optical access to the combustion chamber through multiple windows explained in the Figure 5. The first optical access is through a quartz piston crown as introduced by Bowditch, second access is granted by a window and periscope assembly installed in the cylinder head replacing one of the two exhaust valves and third access is through a window in the liner. A unique
feature of this engine is the separating upper liner that can be dropped down as shown in Figure 5b for rapid cleaning of all windows.

The multiple windows allow use of various optical diagnostic techniques except high speed photography and it is worthwhile understand these techniques due to their direct applicability to the SCE-903 test bed.

Figure 5: Sandia National Labs Optical Engine in (a) closed and (b) open conditions
Utilizing this engine a paper titled “Diesel Engine Combustion Studies in a newly designed Optical-Access Engine using High-Speed visualization and 2D Laser Imaging” was published by Espey et. al. [11]. Three optical diagnostic techniques namely (a) natural flame high speed luminosity imaging, (b) laser induced incandescence imaging (LII) and (c) elastic (Mie) scattering imaging were used in this study. For a reference diesel fuel, at low fuel load they observed, first luminosity at 5.3 CAD (crank angle degrees) before TDC (top dead center) shortly after start of injection (SOI) coinciding with the start of premixed combustion and, the last flame luminosity at 28 CAD after TDC coinciding with the end of computed apparent heat release. During this period combustion was not seen to occur near the injector but the penetration of the leading edge of combusting fuel jet into the squish region was observed. This study is relevant to SCE-903 test bed because natural flame luminosity similar to this is expected in the SCE-903 combustion chamber and will be discussed later.

Based on this optical engine Dr. Mark P. B. Musculus has published a wide range of articles in recent years. Discussing some of his work in detail is of great benefit for the SCE-903 researchers and forms very good guidelines for further development. In 2009 Dr. Musculus and his team published an article titled “Optical Diagnostics and Multi-Dimensional Modeling of Spray Targeting effects in Late-Injection Low-Temperature Diesel Combustion”. In this paper a comparative study between experimental observations in the Sandia optical engine and multi-dimensional modelling of three different spray targeting strategies for low-temperature combustion (LTC) were presented. An 8-hole Cummins XPI, high pressure injector was used with three different injector tips and the engine operating conditions are listed in Table 2 [5].
The three late-injection strategies studied were (a) conventional piston-bowl-wall targeting using 152° included angle tip, (b) narrow-angle floor targeting using a 124° included angle tip and (c) wide angle piston-bowl-lip targeting using a 160° tip. The optical diagnostic techniques utilized in the optical engine included soot luminosity and planar laser-induced fluorescence (PLIF) excited by Nd:YAG laser beam of 284 nm or 355nm. This enabled detection of low temperature first stage ignition (T-PLIF), high temperature second stage ignition (OH-PLIF) and soot-formation consisting of poly aromatic hydrocarbon species (PAH-PLIF) respectively, which characterize the LTC combustion process. The setup is shown in Figure 6. Results from these imaging techniques were compared with the jet-bowl and jet-jet interactions predicted via a computational model. These were further utilized to predict jet-bowl interactions beyond the field of view for all three spray angles, and to understand their effects on UHC, CO and soot formation and oxidation. [5]
Table 2: Operating conditions for three spray targeting article [5]

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<tr>
<th></th>
<th>Combusting</th>
<th>Non-Combusting</th>
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<tr>
<td>Engine Speed, RPM</td>
<td>1200</td>
<td></td>
</tr>
<tr>
<td>Nominal GIMEP, kPa</td>
<td>400</td>
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<tr>
<td>Quantity of injected fuel, mg</td>
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<tr>
<td>Injection duration, ms (CAD)</td>
<td>0.94 (6.75)</td>
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<tr>
<td>Actual SOI, °ATDC</td>
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<tr>
<td>Intake oxygen, % (by volume)</td>
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<tr>
<td>Equivalent EGR rate, %</td>
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<tr>
<td>Global equivalence ratio</td>
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<tr>
<td>Estimated motored TDC density, kg/m³</td>
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<td>Estimated motored TDC temp., K</td>
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<td>16:1 BDC pressure, kPa abs.</td>
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</table>
The paper concluded that jet interactions with piston bowl and with other neighboring jets significantly affects pollutant formation and oxidation as follows; (a) for conventional piston-bowl-wall targeting emissions are reduced due to oxidation of soot, UHC and CO which is enhanced in the late oxidation phase due to rebounding of the jet head from the bowl. (b) for
narrow-angle floor targeting, soot and CO formation is higher near the cylinder head due to redirected fuel jet from the piston bowl & reduced jet-jet interactions. (c) for wide-angle piston-bowl-lip targeting, emissions are higher as oxidation of soot, UHC and CO is reduced owing to diminished jet rebounding from the bowl. Furthermore additional soot is formed in the squish region as fuel jet is deflected towards it.

This article provides a better understanding of application of different injection strategies for emission reduction of DI diesel engines. These findings are valuable for design and development of new injectors such as the one under development at the North Carolina State University. The diesel combustion data presented in this thesis for SCE-903 Optical Engine can be utilized for similar future investigations.

The most recent publication by Dr. Musculus and his colleagues published in June 2016 is entitled “Measurements of Liquid Length, Vapor Penetration, Ignition Delay and Flame Lift-Off Length for the Engine combustion network ‘Spray B’ in a 2.34L Heavy-duty Optical Diesel Engine”.
In this study Eagle et. al. [5] have conducted combusting and no-combusting optical diagnostics on a ‘Spray B’ fuel injector that has three 90 micron holes, details of which can be found in Table 3. This injector is designed by the Engine Combustion Network (ECN) which is a consortium of universities, research labs and industrial partners. [6, 7] Liquid Length, Vapor Penetration, Ignition Delay and Flame Lift-Off Length are the four main metrics presented to understand the fuel injector behavior. Mie-scattering [8] is used to acquire data for liquid length and Schileren diagnostics [5] are used to image vapor penetration as shown in Figure 7. Flame lift-off length is measured from OH* chemiluminescence imaging and ignition delay is calculated from heat release rate plots computed from in-cylinder pressure measurements.

<table>
<thead>
<tr>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Common rail fuel injector</strong></td>
</tr>
<tr>
<td>Nominal nozzle outlet diameter</td>
</tr>
<tr>
<td>Nozzle k-factor</td>
</tr>
<tr>
<td>Nozzle shaping</td>
</tr>
<tr>
<td>Mini-sac volume</td>
</tr>
<tr>
<td>Discharge coefficient at 10MPa pressure drop</td>
</tr>
<tr>
<td>Number of holes</td>
</tr>
<tr>
<td>Holes #1,2,3 angular position</td>
</tr>
<tr>
<td>Holes #1,2,3 exit diameter</td>
</tr>
<tr>
<td>Hole pattern included angle</td>
</tr>
</tbody>
</table>
The operating conditions of the Sandia Optical engine are listed in the Table 4 below. The engine operating parameters varied, fuel injection pressure from 500 bar to 1500 bar, TDC ambient gas temperature from 500K to 1000K and density of 15.2 kg/m$^3$ to 22.8 kg/m$^3$ were varied to build concise database about the fuel injector.
Table 4: Operating conditions of the Sandia Optical Engine for ‘Spray B’ article [5]

<table>
<thead>
<tr>
<th>Case name</th>
<th>SprayB</th>
<th>800K</th>
<th>1000K</th>
<th>15.2</th>
<th>13%</th>
<th>21%</th>
<th>500b</th>
<th>1000b</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature @ TDC [K]</td>
<td>900</td>
<td><strong>800</strong></td>
<td><strong>1000</strong></td>
<td>900</td>
<td>900</td>
<td>900</td>
<td>500</td>
<td>1000</td>
</tr>
<tr>
<td>Density @ TDC [kg/m³]</td>
<td>22.8</td>
<td>22.8</td>
<td>22.8</td>
<td><strong>15.2</strong></td>
<td>22.8</td>
<td>22.8</td>
<td>22.8</td>
<td>22.8</td>
</tr>
<tr>
<td>Non-Reacting O₂ [% vol.]</td>
<td>7.5</td>
<td>7.5</td>
<td>7.5</td>
<td>0</td>
<td>-</td>
<td>-</td>
<td>-</td>
<td>-</td>
</tr>
<tr>
<td>Reacting O₂ [% vol.]</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td>15</td>
<td>13</td>
<td>21</td>
<td>15</td>
<td>15</td>
</tr>
<tr>
<td>Injector rail pressure [bar]</td>
<td>1500</td>
<td>1500</td>
<td>1500</td>
<td>1500</td>
<td>1500</td>
<td>1500</td>
<td>500</td>
<td>1000</td>
</tr>
<tr>
<td>Temperature @ IVC [K]</td>
<td>380</td>
<td>340</td>
<td>454</td>
<td>400</td>
<td>392</td>
<td>396</td>
<td>397</td>
<td>390</td>
</tr>
<tr>
<td>Pressure @ IVC [bar]</td>
<td>2.25</td>
<td>2.01</td>
<td>2.61</td>
<td>1.54</td>
<td>2.27</td>
<td>2.26</td>
<td>2.28</td>
<td>2.28</td>
</tr>
<tr>
<td>Injected liquid mass [mg/cycle]</td>
<td>3.68</td>
<td>3.68</td>
<td>3.68</td>
<td>3.68</td>
<td>3.68</td>
<td>3.68</td>
<td>2.06</td>
<td>2.98</td>
</tr>
<tr>
<td>Engine speed [RPM]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>1200</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Start of Solenoid Energizing [CAD]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>355</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Start of Injection [CAD]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>357.25</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Duration of Solenoid Energizing [°CA], [μs]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>5.86, 795</td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td>Duration of Injection [°CA], [μs]</td>
<td></td>
<td></td>
<td></td>
<td></td>
<td>11, 1500</td>
<td></td>
<td></td>
<td></td>
</tr>
</tbody>
</table>

Conclusively the study incorporated actual engine combustion testing data for ‘Spray B’ fuel injector that included lift of length, ignition delay and so on are noted in the Table below. Here LL stands for liquid length, ID for ignition delay and H stands for Flame lift-off length.

Table 5: Results of ‘Spray B’ injector testing article.

<table>
<thead>
<tr>
<th>Case</th>
<th>$LL \pm \sigma_{LL}$ [mm]</th>
<th>$ID \pm \sigma_{ID}$ [μs]</th>
<th>$H \pm \sigma_{H}$ [mm]</th>
</tr>
</thead>
<tbody>
<tr>
<td>Spray B</td>
<td>9.2 ± 0.4</td>
<td>321 ± 9</td>
<td>15.6 ± 1.0</td>
</tr>
<tr>
<td>15.2</td>
<td>12.8 ± 0.4</td>
<td>352 ± 14</td>
<td>22.9 ± 1.4</td>
</tr>
<tr>
<td>1000b</td>
<td>8.8 ± 0.3</td>
<td>348 ± 12</td>
<td>13.7 ± 1.0</td>
</tr>
<tr>
<td>500b</td>
<td>8.3 ± 0.7</td>
<td>430 ± 12</td>
<td>11 ± 1.0</td>
</tr>
<tr>
<td>800K</td>
<td>11.9 ± 1.0</td>
<td>735 ± 25</td>
<td>24.7 ± 1.3</td>
</tr>
<tr>
<td>1000K</td>
<td>8.2 ± 0.6</td>
<td>192 ± 7</td>
<td>10.5 ± 0.6</td>
</tr>
<tr>
<td>13%</td>
<td>-</td>
<td>374 ± 10</td>
<td>19.4 ± 1.0</td>
</tr>
<tr>
<td>21%</td>
<td>-</td>
<td>200 ± 9</td>
<td>10.8 ± 0.7</td>
</tr>
</tbody>
</table>
This article provides SCE-903 researchers a glimpse of the best injector and combustion testing research carried out by the optical engine research community. The ignition delay estimations using the apparent heat release rate plots calculated are directly applicable to the diesel combustion data presented in this thesis and is very similar to the methods described in Chapter 4 of this thesis.

This synopsis of the recent literature only provides a small peek into the current optical engine research initiatives in the global research community. It is difficult to cover the extensive amount of publications available that detail this type of work. The various optical diagnostics techniques utilized by Dr. Musculus and his colleagues can be effectively implemented on the SCE-903 test bed. Moreover combustion studies of new eco-friendly fuels similar to the studies being conducted at IFP or the Tianjin University can also be carried out on the SCE-903 test bed. However the first step to be able to conduct these studies is to establish baseline operating characteristics and build combustion data for the SCE-903 Engine. This thesis presents the said baseline diesel combustion data and shall lay the foundation for future research on alternative fuels or new injectors utilizing this engine test-bed.
CHAPTER 2: EXPERIMENTAL SETUP

2.1. Cummins SCE-903

A heavy duty optical engine namely Cummins SCE-903 was developed from the Cummins VTA-903T commercial engine by separating a quarter of the original V8. The resulting single vee consisting of two cylinders was modified such that one cylinder would be the optical firing cylinder while the other is utilized for inertial balance. An exact description documented by the U.S. Army [3] is as follows:

“The SCE-903 block consists of Cylinder Nos. 7 and 8 separated from an 8-cylinder block at the midplane of Cylinder No. 5. This configuration allows for two main bearing webs, two camshaft bearing webs, the rear-mounted camshaft gear train, rear engine cover, flywheel, and flywheel housing to remain. The front of the engine is closed off with 16-mm (0.625-in.) steel plate fastened to the cylinder block with socket head cap screws.”

In addition to this all the essential components such as the crankshaft, camshaft, cylinder liners cylinder head, valves and the actuating mechanisms of the original VTA-903 were suitably modified for smooth operation of the SCE-903. A detailed description of the modifications to the sub-assemblies can be found in the U.S. Army’s report in reference 3. Technical Specifications for the SCE 903 are listed in Table 6.
Table 6: SCE-903 Engine Specifications

<table>
<thead>
<tr>
<th>Engine Type</th>
<th>Single-cylinder direct injection diesel</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bore</td>
<td>139 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>121 mm</td>
</tr>
<tr>
<td>Displacement</td>
<td>1.83 l</td>
</tr>
<tr>
<td>Compression Ratio</td>
<td>15:1</td>
</tr>
<tr>
<td>Connecting Rod Length</td>
<td>208.1 mm</td>
</tr>
<tr>
<td>Crank Arm Radius</td>
<td>61.5 mm</td>
</tr>
</tbody>
</table>

Figure 8: SCE-903 Test Bed
2.2. Optical Access

To enable researchers with optical access to the combustion chamber further modification to the firing piston of the SCE-903 were made by a former research fellow. These modifications are briefly discussed in the sections below and their detailed descriptions can be found in reference 1.

Figure 9: Optical Access on SCE-903
2.2.1 Optical Insert

The SCE-903 was further modified to grant optical access to the combustion chamber by using a Bowditch type design that requires a window on the piston crown [2]. This window was designed using a circular quartz insert mounted in a stainless steel retaining cup as illustrated in Figure 10 below. The dimensions of the quartz window were selected based on stress calculations using finite element analysis (FEA) for an in-cylinder pressure of 100 bar. A replaceable copper gasket was added under the quartz insert to cushion the high stress concentration observed on the lower periphery of the window. RTV adhesive was used to secure the quartz insert in place against, small inertial forces acting upwards due to piston acceleration at TDC. An optical access that is diametrically 2.5 inch wide is achieved using a quartz insert. [1]

![Figure 10: Quartz Insert Assembly](image)
A second optical insert was designed which would grant wider optical access to the combustion chamber using a larger window diameter. The larger diameter was possible because of the use of sapphire window material which is stronger as compared to quartz. The Sapphire insert grants an optical access 3.5 inches wide in diameter, which is 20% wider than the quartz window. The quartz window has been utilized for the research described in this thesis but for future research the sapphire insert utilized.
2.2.2 Extended Piston

The optical insert assembly discussed above is fastened in an extended piston using a bronze seal ring and a threaded cap mounted below the optical insert as shown in the Figure 12 below. The bronze seal ring greatly improves the sealing of the combustion chamber around the optical insert. A special spanner was also necessary for ergonomic mounting and dismounting of the optical insert assembly because a conventional tool could not be used on the custom designed threaded cap.

Figure 12: Extended piston with Optical Insert Assembly
2.3 Powertrain.

2.3.1 Driver Engine

The SCE-903 optically modified engine is driven by a 1.2 L single cylinder diesel engine equipped with a standard 3.8 kW electrical starter powered by a 12 Volt automotive battery. The technical specifications for the driver are listed in Table 7 and a photo can be seen in Figure 13.

Table 7: Driver Engine Specifications

<table>
<thead>
<tr>
<th>Model</th>
<th>CHANGFA ZS1115G</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Single-cylinder, four-stroke, direct injection</td>
</tr>
<tr>
<td>Bore</td>
<td>115 mm</td>
</tr>
<tr>
<td>Stroke</td>
<td>115 mm</td>
</tr>
<tr>
<td>Displacement</td>
<td>1.195 liters</td>
</tr>
<tr>
<td>Average piston speed</td>
<td>8.44 m/s</td>
</tr>
<tr>
<td>Compression ratio</td>
<td>17:1</td>
</tr>
<tr>
<td>Rated power</td>
<td>14.7 kW at 2200 rpm</td>
</tr>
<tr>
<td>Mean effective pressure</td>
<td>739.75 kPa</td>
</tr>
<tr>
<td>Fuel consumption</td>
<td>238 g/kW-hr</td>
</tr>
<tr>
<td>Oil consumption</td>
<td>1.47 g/kW-hr</td>
</tr>
<tr>
<td>Injection pressure</td>
<td>18.13±0.49MPa</td>
</tr>
<tr>
<td>Cooling method</td>
<td>Water cooled evaporation</td>
</tr>
<tr>
<td>Net weight</td>
<td>185 kg (408 lb.)</td>
</tr>
<tr>
<td>Overall dimensions</td>
<td>858 × 450 × 699 mm</td>
</tr>
</tbody>
</table>

2.3.2 Speed Reducer

A 3:1 speed reducer is installed between the two engines to step up the torque and stepdown the speed delivered by the driver engine. This permits the use of a smaller diesel
engine to drive the SCE -903 and also permits lower RPM of the SCE-903 for testing. The speed reducer is depicted in Figure 13 and its technical data can be found in Table 8.

Table 8: Speed Reducer Specifications

<table>
<thead>
<tr>
<th>Model</th>
<th>200</th>
</tr>
</thead>
<tbody>
<tr>
<td>Type</td>
<td>Single reduction parallel shaft drive</td>
</tr>
<tr>
<td>Gear type</td>
<td>Helical</td>
</tr>
<tr>
<td>Ratio</td>
<td>3:1</td>
</tr>
<tr>
<td>Input and output shaft diameters</td>
<td>1.5 in.</td>
</tr>
<tr>
<td>Dry weight</td>
<td>37 kg</td>
</tr>
<tr>
<td>Lubrication oil type</td>
<td>Synthetic 75W-90 or equivalent</td>
</tr>
<tr>
<td>Lubrication oil quantity</td>
<td>24 oz.</td>
</tr>
</tbody>
</table>

Figure 13: Driver Engine and Speed reducer assembly
2.3.3 Couplings and shaft design

The powertrain design from the driver to the SCE-903 utilizes a custom adapter, two commercial heavy duty couplings and a commercial muff coupling as shown in the Figure 14. The coupling between the driver engine and the speed reducer transmits power via a rubber center which not only ensures complete power transmission but also compensates for any opposing torques. On the output side the speed reducer has a 1.5 in driven shaft whereas the SCE-903 has a 2 in input shaft which necessitated the use of a heavy duty Lovejoy coupling. Advantage of using these couplings was the tolerance of misalignments in the shafts that they could handle. It is also meant for high torques and is specifically designed for frequent starting and stopping operation. The detailed tolerances and specifications of the couplings can be found in the Table 9 below.

<table>
<thead>
<tr>
<th>Table 9: Coupling Specifications</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Coupling between Driver and speed reducer</strong></td>
</tr>
<tr>
<td><strong>Coupling Model</strong></td>
</tr>
<tr>
<td>Max RPM rating</td>
</tr>
<tr>
<td>Max torque rating</td>
</tr>
<tr>
<td>Max misalignment parallel, axial</td>
</tr>
<tr>
<td>Max misalignment angular</td>
</tr>
<tr>
<td><strong>Coupling between the speed reducer and SCE-903</strong></td>
</tr>
<tr>
<td><strong>Coupling Model</strong></td>
</tr>
<tr>
<td>OD of the coupler</td>
</tr>
<tr>
<td>Max RPM rating</td>
</tr>
<tr>
<td>Max torque rating</td>
</tr>
<tr>
<td>Max misalignment parallel, angular</td>
</tr>
<tr>
<td><strong>Muff coupling for drive shaft extension</strong></td>
</tr>
<tr>
<td><strong>Coupling model</strong></td>
</tr>
<tr>
<td>Max torque rating</td>
</tr>
<tr>
<td>Max RPM rating</td>
</tr>
</tbody>
</table>
To bridge the distance between the speed reducer and the SCE-903 the driven shaft of the speed reducer was extended using a commercial muff coupling details of which are included in the Table 9. It is designed for high torque as the SCE-903 requires high torque.

2.4 *Lubrication and Coolant Systems*

The SCE-903 requires external lubrication and cooling systems have the capability to be heated so that we can simulate a warm engine running condition. The lubrication system is a closed loop pressurized system consisting of an oil reservoir, an oil pressure pump, a 0.5 HP WEG oil sump pump, an oil filter, a Cashco reducer, a Wika pressure regulator, an Omega
liquid level switch and a 1000W electric heater. Approximately 9 gallons of Shell Rotella 15W-40 engine oil is required and is supplied at 70 psig pressure and 170° F temperature via a 2HP Daytona rotary gear pump. The level switch drives the sump pump intermittently to ensure that the oil level in the reservoir as well as the engine oil pan is maintained at a pre-set level.

Similar to the lubrication system the coolant system is also a heated closed loop system that circulates standard 50% coolant, and 50% distilled water mixture at 170° F temperature. This systems consists of a 7 gallon coolant reservoir, a 1000W heater and a 0.5 HP WEG rotary pump. The schematic below depicts a detailed layout of both these systems with the cooling water circulation system for the in-cylinder pressure sensor discussed later.

Figure 15: Lubrication and Coolant Systems
2.5 Exhaust System

The exhaust gases from both diesel engines need to be routed out of the laboratory and a rotary blower and a 6 in diameter exhaust tube was installed to accomplish it. The driver engine generates majority of the exhaust gases because the SCE-903 will be operated in skip fire mode, which means it shall fire a couple of times every 10 motored cycles. Exhaust pipes for both the engines have been modified and routed via flexible exhaust ducts into the rotary blower.
The exhaust system for the SCE-903 was modified to route the gases outside the laboratory via the same root blower as shown in Figure 18. The original exhaust ducts were extended with a stainless steel extension. A 10 feet long flexible exhaust pipe was then installed and sealed onto this extension and the other end was secured to the 6-inch inlet of the root blower.
2.6 Intake System

The SCE-903 was retrofitted with a commercially available aftermarket intake manifold coupled with a conical air filter as shown in Figure 20 below. A custom aluminum mounting plate shown in Figure 19 below was designed to adapt the manifold assembly to the SCE-903 intake mounting holes along with commercially available manifold absolute pressure (MAP) sensor.
Table 10: Technical Specifications of the Intake System

<table>
<thead>
<tr>
<th>Intake Filter</th>
<th>Spectre Conical HPR Air filter</th>
</tr>
</thead>
<tbody>
<tr>
<td>Dimensions</td>
<td>6 in * 6.5in</td>
</tr>
<tr>
<td>Filter media</td>
<td>Pre-oiled cotton gauze</td>
</tr>
<tr>
<td>Intake manifold diameter</td>
<td>3 inches</td>
</tr>
<tr>
<td>MAP sensor Model</td>
<td>GM 12223861</td>
</tr>
<tr>
<td>MAP Operating pressure</td>
<td>40 to 304 kPa</td>
</tr>
<tr>
<td>MAP Operating Voltage</td>
<td>5.1 V ± 0.36</td>
</tr>
</tbody>
</table>

Figure 19: Mounting adapter for Intake assembly

A MAP sensor capable of digitally measuring up to 3 bar absolute pressure was selected. The reason for selection was good integration into the intake manifold so that the system is capable of measuring turbocharged intake pressures in the future. The details of the intake system can be found in Table 10. The calibration for the MAP sensor can be found in graph in Figure 22. The intake pressure for SCE-903 with the current intake system was recorded to be 96.56 kPa at a location just next to the intake port.
Figure 20: Intake system

Figure 21: Intake System and MAP sensor on the SCE-903
2.7 Fuel System

2.7.1 Mechanical Components of fuel system

Fuel system design is very crucial for research engines like the SCE-903, because they not only need to have durability and dependability similar to a normal diesel engine, but also require a great deal of flexibility and superior control. For this project flexibility of the fuel systems was very important because it needs to supply different types of fuels. Secondly high degree of control is necessary since the SCE-903 engine runs in skip fire mode, meaning there are 3 combustion cycles followed by 10 motored cycles. This skip fire operation mode is achieved by precise control over fuel injection by the experimenter. Figure 23 gives the details of the hardware of the fuel systems which includes a fuel tank, a low pressure pump, a fuel

Figure 22: Graph depicting the calibration of MAP sensor.
filter, a high pressure pump and a commercial fuel injector. Original VTA-903 used a 1980’s series injector but the SCE-903 is equipped with a modern 6 nozzle Bosch fuel injector mounted in a metal adapter to match the cylinder head dimensions. Figure 24 shows the custom designed adapter and fuel injector assembly and Table 11 gives the technical specifications of all the components.

**Table 11: Mechanical Components of Fuel Systems and Technical Data**

<table>
<thead>
<tr>
<th>Component</th>
<th>Model</th>
</tr>
</thead>
<tbody>
<tr>
<td>Common rail</td>
<td>Bosch 18985</td>
</tr>
<tr>
<td>High pressure fuel pump</td>
<td>Bosch CR/CP/1S3/R70/10 – 16S</td>
</tr>
<tr>
<td>Fuel pump motor</td>
<td>GE 5K49TN4333AX</td>
</tr>
<tr>
<td>Fuel filter</td>
<td>CAT 1R-0751</td>
</tr>
<tr>
<td>Injector</td>
<td>Bosch 0445110816</td>
</tr>
<tr>
<td>Injector Nozzle diameter</td>
<td>152 micrometer</td>
</tr>
<tr>
<td>Fuel spray cone angle</td>
<td>160°</td>
</tr>
</tbody>
</table>

![Figure 23: Fuel System Schematic](image_url)
Figure 24: Fuel Injector and Adapter assembly
2.7.2 Fuel Pressure Control System

The fuel system needs to enable the experimenter to have precise control over fuel rail pressure, injection timing and injection duration. It is very important to control these especially for an optical engine because these determine the combustion characteristics and hence the engine performance data. Fuel pressure is controlled using the National Instruments myDAQ controller and a fuel rail pressure sensor whose technical details are listed in Table 12.
The schematic in Figure 26 depicts the process of fuel pressure control that uses a virtual PID (Proportional-Integral-Derivative) control loop built in LabVIEW. A PID controller was chosen for its simplicity and flexibility over a wide range and capability to continuously monitor fuel pressure. The recommended P, I and D values that exhibited best control in practice are 0, 0.05 and 0 respectively.

Table 12: Common Rail Pressure Sensor and myDAQ technical data

<table>
<thead>
<tr>
<th>Common rail pressure sensor</th>
<th>Bosch 0281002283</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max voltage output</td>
<td>5V</td>
</tr>
<tr>
<td>Operating range</td>
<td>0 to 1500 bar</td>
</tr>
<tr>
<td>Full scale output</td>
<td>0.5 to 4.5 V</td>
</tr>
<tr>
<td>NI myDAQ</td>
<td></td>
</tr>
<tr>
<td>Input/Output Range</td>
<td>±10 V</td>
</tr>
<tr>
<td>Sampling rate</td>
<td>200 kS/s single channel</td>
</tr>
<tr>
<td>Accuracy</td>
<td>100 ppm of sample rate</td>
</tr>
</tbody>
</table>

The control circuit is a feedback loop that reads the rail pressure via the common rail pressure sensor and the myDAQ, and compares it with the set point (determined by the experimenter). The PID control program then calculates the error between these two data points and accordingly generates an output voltage between 0 to 10 volts. The myDAQ applies this voltage output to the high pressure pump solenoid which controls the fuel flow rate and hence the fuel pressure in the common rail. Once the fuel pressure reaches the set value the error reduces to zero but the voltage output is held constant as it directly controls the flow control valve in the high pressure pump which in turn controls the common rail pressure. A
detailed description and working principles of the high pressure pump can be found in reference 1. Ex. control voltage output is constant at 1.56V for fuel pressure setting of 400 bar.

**Figure 26: Fuel Rail Pressure Control System**
2.8 Crank Angle Degree and Exhaust TDC Detection

Dynamic location of the piston is measured using the angle of rotation of the engine crank shaft, otherwise known as crank angle degrees (CAD). It is measured by an optical shaft encoder with 2 output channels using a photodetector for accuracy. Channel B gives 720 pulses per revolution, i.e. one pulse per half CAD, while channel Z gives one pulse every 360 degrees. The revolutions of a single tooth disk rotating in a 1:2 ratio with the crank shaft are detected by a Hall Effect sensor. Combining these two digital signals gives one signal per 4-stroke cycle of the SCE-903 and greatly reduces the chance of TDC error.
Figure 28: Shaft Encoder, Speed sensor and Hall Effect sensor

The above mentioned sensors are synced with the piston such that the signal is obtained precisely when the piston is at the exhaust top dead center (Ex TDC). The two digital signals, one-per-rev signal from the encoder and once-per-cycle from the Hall Effect sensor, are combined via an AND operation to indicate the Ex TDC. A complete cycle begins from Ex TDC and is used as base reference to begin at zero CAD. Consequently compression TDC appears at 360 degrees which can be better understood from the diagram attached below. The window displays two complete cycles, and is set such that Ex TDC can be observed at the center on channel 2. Channel 1 displays injection trigger from the black box, and channel 3 displays the delayed injection signal from the function generator which are explained in further
detail in Section 2.9. The fuel injection signal is 10 degrees before compression TDC that is easily recognizable from the peak of the pressure signal recorded on channel 4.

Figure 29: Ex TDC and other signals observed on an oscilloscope (two cycles).

A circuit diagram illustrating the Ex TDC detection and the further operations is shown in Figure 30. The CAD signal from channel $\bar{B}$ of the encoder is boosted to 5V signal for precision in data acquisition and is also illustrated in the circuit diagram. These operations are carried out in a circuit enclosed in the black box shown in Figure 30.
2.9 Fuel Injection Control Circuit

Fuel injection for the SCE-903 is controlled via separate system that deliver’s flexibility and precise control over the quantity of fuel injected and the timing of injection. It is critical for the experimenter to have complete control over the quantity and timing, as this determines the operating characteristics of a compression ignition diesel engine [24]. Here timing refers to start of fuel injection with respect to the location of the crank and is typically described as number of CAD after top dead center (ATDC). For example when fuel is injected at -10 ATDC, it means that the start of injection (SOI) is 10 degrees before from compression
TDC. This can also be calculated in seconds using Equation 1 where \( t_d \) is the time between Ex TDC and the following SOI.

\[
t_d = \frac{\theta}{N \times 360^\circ}
\]

\text{.... Eqn 1}

where, 

\( N \) – Number of revolutions per second (RPM/60)

\( \theta \) – SOI in CAD from Exhaust top dead center

The quantity of fuel is controlled by changing the injection duration, and is calibrated to 46 mg per injection. This value was derived by experimentation of various operating conditions for the SCE-903, which will be explained in detail later in this document.

\[\text{Figure 31: Injection Control System Schematic}\]
The injection circuit consists of an injection switch, the black box, a function generator and an injector driver circuit as shown in Figure 31. As explained in the previous section CAD and Ex TDC are precisely detected and employed further for triggering fuel injection in the SCE-903. Injection switch was added with an AND operation and only when the switch, shown in the circuit diagram in Figure 31 above, is turned on, will fuel injection begin. Thus it lets the user decide, if and when fuel shall be injected into the combustion chamber. The AND operation reads the position of the injection switch as well as the Ex TDC and only if both inputs are high, it triggers the injection circuit.

But the precise timing and duration of fuel injection is controlled via the function generator. The injection duration ($t_{\text{inj}}$) and the injection timing are set in the function generator, and it is triggered via the injection switch. Once triggered the function generator delays the signal by $t_d$ calculated using Equation 1 inputted in delay A. The injection duration is controlled by input to delay B, and is set by the experimenter to $A + t_{\text{inj}}$. Injection duration for the various cases presented in this paper are calibrated so that precisely 46 mg of fuel is injected each time. The calibration shall be explained in detail in latter sections.

2.10 In-cylinder Pressure Data Acquisition System

In-cylinder pressure is measured using a water cooled piezoelectric pressure transducer, Kistler model 6067B shown in Figure 32 below. It is mounted using a mounting sleeve in a designated access hole drilled in the SCE-903 cylinder head. Technical details for the pressure transducer and its ancillary cables and instruments can be found in Table 13.
The connecting cable type 1919 is a high temperature and high sensitivity cable with two layers of metal braided shielding and a BNC connector on the other end.

**Table 13: Technical Data for Kistler 6067B Pressure sensor**

<table>
<thead>
<tr>
<th>Range</th>
<th>bar</th>
<th>0 …… 250</th>
</tr>
</thead>
<tbody>
<tr>
<td>Overload</td>
<td>bar</td>
<td>300</td>
</tr>
<tr>
<td>Operating temperature range</td>
<td>°C</td>
<td>-50 …. 350</td>
</tr>
<tr>
<td>Cooling water flow, pressure</td>
<td>l/min, bar</td>
<td>0.3 - 0.5, ≤6</td>
</tr>
<tr>
<td>Sensitivity</td>
<td>pC/bar</td>
<td>≈−25</td>
</tr>
<tr>
<td>Sensitivity shift (50° to 350° C)</td>
<td>%</td>
<td>≤±2</td>
</tr>
<tr>
<td>Linearity, all ranges</td>
<td>% FSO</td>
<td>≤±0.5</td>
</tr>
</tbody>
</table>

The dynamic pressure reading from the 6067B sensor are passed through a model 5004 Kistler charge amplifier, which converts the electrical charge output from the pressure transducer to voltage with a 20 bars per volt sensitivity. The output from the change amplifier
is then read and stored on the computer using National Instruments (NI) real time data acquisition (DAQ) hardware-software package with details in Table 13.

**Table 14: Technical Data for National Instruments DAQ Hardware**

<table>
<thead>
<tr>
<th>Feature</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>DAQ card</strong></td>
<td>PCI-MIO-16E-4</td>
</tr>
<tr>
<td><strong>Input Range</strong></td>
<td>±0.05 V to ±10 V</td>
</tr>
<tr>
<td><strong>Max sampling rate</strong></td>
<td>500 kS/s single channel</td>
</tr>
<tr>
<td><strong>Accuracy</strong></td>
<td>99.97% during a 24hr recording period</td>
</tr>
<tr>
<td><strong>Analog Inputs Outputs</strong></td>
<td>16 single ended, 8 differential channels</td>
</tr>
<tr>
<td></td>
<td>2 channels, all 12 bit resolution</td>
</tr>
<tr>
<td><strong>Counters/Timers</strong></td>
<td>2 up/down, 24-bit resolution</td>
</tr>
<tr>
<td><strong>Connector block</strong></td>
<td>BNC-2110 to instruments &amp; PCI bus to computer</td>
</tr>
<tr>
<td><strong>Application software</strong></td>
<td>LabVIEW, Measurement Studio etc.</td>
</tr>
</tbody>
</table>

The detailed data acquisition process is depicted in Figure 33 and in addition to the pressure transducer and charge amplifier, it includes a NI PCI-MIO-16E-4 data acquisition card, a NI BNC-2110 connector block and a LabVIEW program. In-cylinder pressure, dynamically sensed by 6067B transducer is amplified to a few volts by the charge amplifier and converted from analog to a digital signal by the combination of BNC-2110 board and 16E-4 DAQ card. This digital value is then co-related with the CAD, inputted to BNC-2110 from the black box and stored on the computer in a excel file by a LabVIEW program. This system performs this process at the rate of 500 Hz with 99.97% accuracy.
The LabVIEW virtual instrumentation or VI and the block diagram can be seen in Figure 34 and 35. The Ex TDC and CAD signals are inputted to the BNC-2110 block at user1 and AI Start respectively. Ex TDC signal triggers the LabVIEW program and it begins recording the in-cylinder pressure from the transducer. CAD signal acts as a marker for each pressure value received from the transducer and is used to co-relate and store pressure data. The encoder delivers 0.5 CAD resolution, which means 720 CAD signal per revolution and 1440 per cycle. As CAD is the marker, we are able to gather 1440 data points for every 4 stroke cycle of the SCE-903 from the existing DAQ system.
Figure 34: LabVIEW VI for In-cylinder Pressure recording.

Figure 35: LabVIEW block diagram for In-cylinder Pressure recording.
2.11 High Speed Camera Set up.

The SCE-903 is equipped with a Phantom v4.3 high speed camera that is capable of recording a 90,000 frames per second. A Nikkor 105 mm zoom lens with built in UV filters was used, typically set with an aperture of 11. The aperture setting and the exposure setting of 3 µs is kept constant across all experiments.

The high speed camera is synchronized with fuel injection in the combustion chamber using the injection trigger via the function generator as shown in Figure 36.

![Figure 36: High Speed Camera Schematic](image_url)

2.12 Speed Sensor

The SCE-903 is driven by a smaller diesel driver engine which lacks a speed indicator, hence installing an external speed indicator on the SCE-903 was necessary. A commercially available magneto resistive speed sensor, with an integrated 21 segment LED display was selected. A magneto resistive sensor uses the magnetoresistive effect, which is the change in resistivity of a current carrying magnetic material in the presence of an external magnetic field.
This change in resistivity causes a change in the output voltage from the sensor generating a square wave signal. A digital counter reads the frequency of the square wave and dynamically displays the speed on the 21 segment display. The external magnetic field is applied by a small circular permanent magnet fixed to a rotating plate on the crankshaft of the SCE-903. The sensor requires a 12V DC input voltage supplied from AC to DC voltage adapter. Technical data for the sensor can be found in Table 15 and the sensor mounting is displayed in Figure 37.

<table>
<thead>
<tr>
<th>Table 15: Technical Data for Speed Sensor</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Sensor make and model</strong></td>
</tr>
<tr>
<td><strong>Supply voltage</strong></td>
</tr>
<tr>
<td><strong>Operating range</strong></td>
</tr>
<tr>
<td><strong>Detection Range</strong></td>
</tr>
</tbody>
</table>

Figure 37: Speed sensor mounting on SCE-903
CHAPTER 3: METHODS AND EXPERIMENTAL PROCEDURE

3.1 Introduction

In this chapter, experimental procedures developed for operating the SCE-903 test bed are discussed in detail. The SCE-903 test bed is a large and complex system involving numerous sub-systems and hence the author has chosen to outline the methods, safety precautions and procedures separately. This chapter shall discuss fuel calibration and injection testing procedures, engine operating procedures and operating conditions. Two types of data for in-cylinder combustion is acquired from the current test bed – one is the in-cylinder pressure data and second is the natural combustion flame images. The experiments to acquire these two data sets and methods for data analysis are also discussed in this chapter.

3.2 SCE-903 Experimental Procedures

3.2.1 Safety precautions

The following precautions should be taken before running the test-bed for human safety and also to prevent system damage.

- Experimenters are required to wear safety goggles and ear protection during each run. Proper closed-toe footwear is mandatory and gas masks available in the lab are recommended for use while running the system.

- Experimenters are strongly advised to run any experiments only in the presence of at least two personnel in the laboratory.
• During operation all personnel should be positioned either behind the driver engine or well in front of the SCE-903 safety shield. Stand clear of any rotating parts, and ensure all personnel are outside the plane of rotation of flywheel and couplers.

• Ensure each bolt clamping base of the driver engine, speed reducer and the SCE-903 are torqued down to approximately 100 ft-lb. There are 17 of these 5/8” bolts clamping the mounting stands to the base plate.

• Ensure all mounting bolts for the driver engine and the speed reducer are tightened such that the damping pads are compressed slightly. Ensure the three flywheel adapter mounting bolts are tightened.

• Confirm that all set-screws for shaft keys and couplers are secured tightly. There are eleven of these – one on the driver engine adapter, two on each side of the rubber center coupler, 4 on the muff coupling sleeve between the speed reducer and SCE-903, and one on each side of the SCE-903 Lovejoy coupling.

• Check the optical piston cap, and confirm it is snug fit. Refrain from over-torqueing.

• Ensure the exhaust opening is free of any debris during operation, debris can be forced into the root blower and cause damage.

• Regularly check oil levels in the SCE-903 and the driver engine, they must be half full.

• Remove all tools and loose objects from the test-bed. Thoroughly check the system components each time before running.
• During fuel calibration carried out outside the engine precaution must be taken. Proper eye protection and safety equipment should be used, proper clothing is necessary as pressurized diesel particles can be injurious, if in contact with the skin.

3.2.2 Fuel Injector calibration

It is important to ensure that the quantity of fuel injected into the combustion chamber has been empirically verified using the existing experimental setup. Fuel quantity was calibrated for all the six cases presented in this thesis. Actual in-cylinder pressure during compression of 25 bar [1] was accounted for during fuel injector calibration, which yields accurate mass of fuel injected into the combustion chamber during experiments.

A highly sensitive Mettler AE100 weigh scale with a least count of 0.1 milligrams is used along with a closed container lined with fuel absorbing material. Procedure for fuel calibration is as follows:

1. Mount the fuel injector outside SCE-903 using proper supports. Ensure all the fuel lines are connected tightly and securely.
2. Test run the fuel system and test fuel injection for proper functioning.
3. Place the container on the scale and tar the dry weight of the container when the reading stabilizes.
4. Now carefully place the container under the Bosch fuel injector, ensuring that the nozzle is sufficiently deep inside the container.
5. Switch on the fuel pumps, fuel control systems and set the fuel pressure in LabVIEW virtual instrumentation panel to required value, accounting for motoring pressure. Set the appropriate injection duration on the function generator as discussed in Section 2.11.

6. Now inject fuel in the container 100 times which yields a measurable amount of fuel.

7. Weigh the container and record the weight of the injected fuel ($m_i$).

8. Repeat this procedure at least five times and average the values of mass of fuel injected to account for abnormalities and small pressure fluctuations.

Stable diesel combustion was observed at 1000 bar fuel pressure, with 1ms injection duration and hence was selected as a base case for fuel quantity calibration. The mass of fuel injected for this case was calculated to be 46 mg. The Table 16 below displays all fuel injection durations with their calibrated fuel quantities for approximately 46 mg of injected fuel. The air fuel ratio is calculated assuming 80% volumetric efficiency.

Table 16: Fuel injector calibration

<table>
<thead>
<tr>
<th>Fuel Pressure (bar)</th>
<th>Injection Duration</th>
<th>Weight of fuel per Injection (mg)</th>
<th>AF Ratio</th>
<th>Equivalence Ratio</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>Milliseconds</td>
<td>CAD (@ 600 rpm)</td>
<td></td>
<td></td>
</tr>
<tr>
<td>800</td>
<td>1.14</td>
<td>4.104</td>
<td>46.52</td>
<td>36.1511</td>
</tr>
<tr>
<td>1000</td>
<td>1.00</td>
<td>3.600</td>
<td>46.11</td>
<td>36.166</td>
</tr>
<tr>
<td>1200</td>
<td>0.905</td>
<td>3.258</td>
<td>45.85</td>
<td>36.679</td>
</tr>
</tbody>
</table>
3.2.3 Pre-run preparations

It is necessary to simulate actual engine operating conditions and ensure proper data acquisition during testing. To achieve this following pre-run preparations are necessary for both the sets of experiments:

- The oil and cooling systems are equipped with electric heaters. Engine coolant and oil are heated to 170° F and optical combustion chamber warmed up for at least 2 hours.
- Special care is necessary for the in-cylinder pressure sensor as output signal drift is a common problem posed by Kistler sensors used in oily environments. Sensor and connecting cable should be relatively oil-free during testing. If necessary the sensor may be dismantled and thoroughly cleaned with special electrical cleaners.
- Check the sensor signal for drifts after cleaning. If drifts higher than ±30 millivolts/sec are observed warm up the charge amplifier for 60 minutes [15] and check insulation of all wires and all BNC connections.
- Verify that the pressure sensor cooling water system is running and water is clean of any oil or coolant deposits. It should be cleaned at regular intervals.
- Switch on the function generator before connecting any ancillary circuits to protect other circuits from any output surges then connect all the necessary cables.
- Ensure trigger signals are relayed to the fuel injector. Check the fuse in injection driver circuit control box and the black box and verify all cables are properly connected.
- Verify the CAD and Exhaust TDC signal are correct and secured to the connector block.
- Ensure that there are no fuel leaks or air bubbles in the fuel systems.
3.3 Combustion Experiments

Experimental procedures followed during combustion testing in the SCE-903 test bed are outlined in the following Sections. The safety checklist and the pre-run preparations should be completed before each test run.

3.3.1 Experimental procedure for pressure data acquisition.

All data presented in this report was recorded at engine speeds of 600 rpm. To simulate actual engine operating conditions the optical combustion chamber is heated to 170° F. To prevent over-heating of the optical piston crown and avoid window fouling the engine was operated in skip fire mode, where optical piston was fired thrice every 10 motoring cycles. Cylinder pressure and fuel injection pressure were monitored and digitally recorded during all experiments. The first combustion cycle was selected from the skip fire operation mode. The in-cylinder pressure measurements were recorded at half crank-angle-degree (CAD) increments and averaged over 20 combustion cycles. The step by step experimental procedures are outlined in detail in Appendix A of this thesis.

3.3.2 Experimental procedure for combustion flame imaging.

Natural flame luminosity images of combustion were recorded through the optical access with the high sped camera setup described in Section 2.11, equipped with a 105 mm lens. Sampling rate was set to 5400 frames per second with an aperture of 11 and 3 µs exposure and was constant across all cases. The resolution of 320 x 312 pixels was seen to yield best
images throughout the piston stroke. At an engine speed of 600 rpm and the selected frame rate corresponds to 3 frames for every 2 CAD interval. For each test case three sets of combustion images were recorded and the typical combustion images will be presented. The high speed camera is synchronized by triggering both fuel injection and the camera at the same moment via the function generator depicted via a schematic in Figure 38. Consequently every combustion video begins recording from SOI. The procedure for recording combustion videos outlined in detail in Appendix A of this thesis.

Figure 38: Fuel Injection and High Speed camera trigger synchronization.
3.4 Data Analysis

In cylinder pressure data and combustion images obtained by from the experiments is processed via excel programming and MATLAB code. The formulae and functions utilized for combustion analysis are discussed in this Section.

3.4.1 Cylinder Volume Measurement

The in-cylinder volume is calculated utilizing the engine geometry and known crank angle degrees. The clearance volume ($V_c$) for the SCE-903 was physically measured to be 128.57 cubic centimeters. The formula used to calculate cylinder volume [24] derived using the slider-crank mechanism is at any crank angle ($\theta$) is:

$$\frac{V}{V_c} = 1 + \frac{1}{2} (r_c - 1) \times [R + 1 - cos \theta - (R^2 - sin^2 \theta)^{1/2}] \quad \text{…….. Eqn 2}$$

where
- $R$ is ratio of connecting rod length to crank radius
- $V$ is volume at crank angle $\theta$,
- $V_c$ is clearance volume,
- $r_c$ is compression ratio

3.4.2 In-cylinder Pressure data analysis

Pressure pegging

Piezoelectric transducers like the one used in the SCE-903 measure relative pressure, instead of absolute pressure. Hence the in-cylinder pressure measurements made using these transducers need to be referenced to a known absolute pressure value and the corrected in-cylinder pressure values need to be determined, this process is known as pressure pegging. It is necessary to perform pressure pegging to maintain the accuracy of absolute combustion
metrics like peak pressure and heat release rate. Several methods of pressure pegging have been useful to researchers before [14] the most common of which are: [13]:

1. Reference cylinder pressure measurements at intake BDC: In this method the in-cylinder pressure measurements at fixed crank angles near IBDC are equated with intake manifold absolute pressure (MAP). [15]

2. Fitting a polytropic compression index to the in-cylinder compression pressure measurement: This is a very popular and simple method but selecting a proper polytropic compression index can prove to be a challenge.

3. Use another pressure transducer lower in the cylinder to reference the piezoelectric pressure transducer: While this method is most accurate, it has severe limitations for practical application. Installing a second pressure transducer near BDC in the cylinder is difficult, time consuming and costly.

The first method is simplest and is known to yield good results at low speeds and part loads. As the engine rpm for all experiments presented is low (600 rpm) the data presented in this thesis is pegged by referencing in-cylinder pressure over the intake stroke with intake manifold absolute pressure.

Pressure correction formula used for every pressure data point is

\[ P_{actual} = P_{measured} - P_{correction} \quad \text{………… Eqn 3} \]

Where

\[ P_{correction} = P_a - P_i \]
Where $P_a$ is average intake pressure measured inside the combustion chamber and $P_i$ is the intake manifold absolute pressure measured by the MAP sensor.

This corrected pressure data is then further smoothened using a seven point average across all 1440 data points and then analyzed on a pressure versus crank angle plot.

**Indicated Mean Effective Pressure (IMEP)**

Indicated mean effective pressure is a relative engine metric which quantifies the engines ability to perform work and is independent of engine size. Excel calculates IMEP using the following formula:

$$imep = \frac{\Delta \theta}{V_s} \sum p \times \frac{dV}{d\theta}$$

……… Eqn 4

where $V_s$ is the swept volume, $p$ in cylinder pressure and $dV$ is instantaneous change in volume

This equation yields gross IMEP when equated across the 4 stroke cycle and pumping imep in the intake and exhaust stroke range. [16]

**3.4.3 Ignition Delay**

Ignition delay defined is defined as the period between the start of fuel injection and the start of combustion. [24] In this paper it is calculated based on the change in the slope of the in-cylinder pressure plots and the heat release rate diagrams. More accurate information for ignition delay would be available from the natural flame combustion imaging. But due to the small area of optical access on the SCE-903 these predictions were not found to be as accurate and hence the conventional methods have been utilized.
3.4.4 Heat Release Rate

Combustion chamber of a direct injection compression ignition engine like the SCE-903 can be modeled as a single open system, with only mass flow of fuel across the system boundary, assuming intake and exhaust valves are completely closed. Applying the first law of thermodynamics to this system the following Equation for apparent net heat release rate derived Heywood [24] is used:

\[
\frac{dQ_n}{d\theta} = \frac{\gamma}{\gamma - 1} \frac{dp}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta}
\]

\[\text{Eqn 5}\]

where \(Q_n\) is apparent net heat release rate in Joules per degree

\(P\) is in-cylinder pressure in Pascals

\(V\) is in-cylinder volume in cubic meters

\(\gamma\) is the ratio of specific heats taken to be 1.325

This Equation computes the amount of heat added by the fuel, which produced the measured pressure variations. It is based on the following assumptions:

- Contents of the cylinder can be modeled as an ideal gas, and \(R\) assumed constant
- Sensible enthalpy of injected fuel is negligible
- Mixture is assumed to be uniform across the combustion chamber.
- \(\gamma\) – the ratio of specific heat of combustion gases is assumed to be a mean across the combustion processes.

Further since the engine is operated in skip fire mode the heat transfer losses and crevice flow losses are estimated to be similar for all cases, until the time of significant heat release. Also the heat release rate curves are utilized for a comparative analysis during the presented
experiments. Hence the small errors (<1 Joules per CAD) due to crevice flows will not change the conclusions reached in this thesis and are excluded.

The above equation can be used to closely approximate the cumulative heat release rate by integrating the net heat release rate over the complete combustion process as follows:

\[ Q_c = \int_{\theta_{\text{start}}}^{\theta} \frac{dQ_n}{d\theta} \, d\theta \]

\[ \text{Eqn 6} \]

where \( Q_c \) is the cumulative heat released by the fuel during combustion

\( \theta_{\text{start}} \) is the start of fuel injection in CAD

\( \theta \) is the end of power stroke in CAD

In the future a fairly accurate estimate of the gross heat released over complete combustion process could be made with the incorporation of a heat transfer model like one’s presented by Woschni or Anand [17, 18, and 24]. Although the conclusions reached in this thesis will not change because the relative errors in heat release calculations shall be negligible. Thus heat transfer losses are also excluded during heat release approximations.

### 3.4.5 Natural Flame Combustion Imaging and Analysis

Combustion images gathered from the optical engine are natural flame luminosity images. The natural flame luminosity is due to chemiluminescence and soot luminosity. Chemiluminescence is characterized by relatively weak natural light emitted during early stages of low temperature combustion. According to Dec et. al. [20] the photons emitted by excited radicals, when they return to ground state from excited combustion state are responsible for this phenomenon of chemiluminescence [8]. Following chemiluminescence,
soot particles are formed and due to combustion heat begin burning with intense yellow light. The yellow color is detected in high speed images and is known as soot luminosity. [8, 9]

Theses combustion images further processed to yield a better resolution of the luminosity values of each frame. These characteristic combustion images at selected CAD for each case are presented in the results section.

3.5 Experimental conditions for diesel combustion testing

The data presented in this thesis is utilized to set baseline operating characteristics of the SCE-903 optical engine with the quartz insert assembly for diesel fuel. Since the SCE-903 combustion chamber with the quartz insert had never been fired, known operating regions for the engine were not available. Hence the optimum operating conditions were selected based on some combustion trials conducted on the SCE-903, by varying engine speed, fuel injection pressure and timings and heating temperatures of oil and coolant. For the settings presented in Table 17 below, best combustion images and pressures were observed.

<table>
<thead>
<tr>
<th>Table 17: SCE-903 Operating conditions</th>
</tr>
</thead>
<tbody>
<tr>
<td>Engine Speed</td>
</tr>
<tr>
<td>Coolant &amp; Oil Temperature</td>
</tr>
<tr>
<td>Fuel Injection Pressure</td>
</tr>
<tr>
<td>Injection duration</td>
</tr>
<tr>
<td>Injection advance</td>
</tr>
</tbody>
</table>
These operating conditions were chosen as a reference case for fuel quantity calibration for the SCE-903, and fuel mass injected was measured to be 46 milligrams. This yields an approximate equivalence ratio of 0.4, assuming a volumetric efficiency of 0.8. Hence the engine is operating at lean conditions, so as to control the peak in-cylinder pressure and nominal gross indicated mean effective pressure (IMEP) to avoid stresses on the optical window insert.
CHAPTER 4: EXPERIMENTAL CHALLENGES AND SOLUTIONS

Experimental combustion research is generally accompanied with practical challenges and issues and the SCE-903 test bed is no exception. The SCE-903 test bed is a recently constructed hefty system that requires close attention to multiple variables. It is the first time the test bed is utilized for rigorous testing and hence it is important to document any major problems. The major challenges faced as well as the solutions devised by the author while gathering combustion data presented in this thesis are discussed briefly in this chapter. Main issues like damage to the optical window insert, failures of heavy duty couplers, coolant system issues etc. are discussed below.

4.1 Optical Insert Damage

The optical insert for the SCE-903 is made of quartz glass, and was designed by assuming an in-cylinder pressure load of 100 bar. But in-practice the optical insert was seen to withstand a highest pressure of only 75 bar. The window material catastrophic failed during preliminary combustion tests designed to understand the operating characteristics of SCE-903. The reasons for failure of the window during experimentation are explained in further detail. Fortunately none of the other components of the system suffered any damage. The piston rings, cylinder liner and the pressure sensor were intact.
Engine Operating Conditions

The SCE-903 was originally developed by the U.S. Army for testing lubricants due to which combustion testing was never carried out in the modified combustion chamber. Hence it was necessary to understand the operating characteristics of SCE-903 with the new optical access geometry. During one such experiment with the operating conditions proposed in Table 18 resulted in catastrophic failure of the optical window. Fuel injection timing was advanced to 20 CAD BTDC, which typically increases in-cylinder pressure and hence power output of a normal diesel engine. To limit the pressure rise in the optical engine fuel quantity was reduced to 50% of the base case. Albeit these precautions the peak pressure observed during this experiment was 7.5 MPa, which is 2.5 MPa higher than the base case of 1000 bar fuel pressure at 10 CAD BTDC fuel injection timing.

Table 18: Operating conditions during window damage

<table>
<thead>
<tr>
<th>Engine Speed</th>
<th>600 rpm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Coolant&amp; Oil Temperature</td>
<td>~ 170° F</td>
</tr>
<tr>
<td>Fuel Injection Pressure</td>
<td>800 bar</td>
</tr>
<tr>
<td>Injection duration</td>
<td>0.8 millisecond</td>
</tr>
<tr>
<td>Injection advance</td>
<td>20 CAD BTDC</td>
</tr>
<tr>
<td>Fuel quantity</td>
<td>24.94 mg</td>
</tr>
</tbody>
</table>

In-Cylinder Pressure

The in-cylinder pressure recorded during this event is presented in Figure 39. A peak pressure of 75 bar was recorded just before the window cracked. The pressure curve depicts a high amount of knocking during the complete combustion process. The reason for such a sharp
increase of pressure can be attributed to the dominantly pre-mixed combustion mode observed in SCE-903, which is discussed in detail in later Sections. Advancing fuel injection timing to 20 CAD BTDC allowed the diesel fuel to vaporize and mix extremely well with the air in the combustion chamber. Furthermore the air temperature in the combustion chamber at 20 CAD BTDC, was not high enough for the diesel to combust, which resulted in a lean but extremely well mixed almost uniform fuel-air mixture.

When the piston reached compression TDC this fuel-air mixture caused instantaneous combustion, akin to a small explosion, releasing a high amount of heat. This caused the pressure to suddenly spike from 27 bar to an alarming 73 bar in only 3 CAD. Resulting in a pressure loading of 15.333 bar/CAD, which was higher than the design rating for the window. This coupled with other reasons caused the window to crack. Combustion images recorded during this event have also been presented in later Sections.

![Figure 39: In-cylinder pressure graph during window damage.](image-url)
Reasons for failure of Window Material

The original quartz window insert was recycled from another experimental setup that uses lasers in previous experiments. Laser beams are passed into a combustion chamber via quartz window inserts of small diameter. Sometimes due to errors in operation the laser beams cause miniscule damage to the optical inserts rendering them useless for laser based experimentation. As seen in Figure 40 the original insert used in the SCE-903 had these miniscule internal voids. During normal operation of the SCE-903 these flaws did not adversely affect combustion, but they did compromise the structural rigidity of the piston crown. These were weak spots in the window and served as starting points for cracks or other deformations during unexpected high pressure spikes such as the one experienced during this run. The resulting crack in the optical window is shown Figure 40b, the voids where it originated can be seen Figure 40a on the left.

(a) Internal voids caused by laser
(b) Starting point of crack

Figure 40: Quartz optical window insert defects and damages
The high speed combustion images shown in Figure 41, depict the crack developing in the optical window. The first image shows the start of combustion with the fuel injector tip visible at the center. In the second image a turbulent pre-mixed combustion flame is evident and in the last two images the window crack due to high in-cylinder pressure is seen.

**Modifications to the system**

The window insert was replaced with a new quartz window that does not have similar structural weak spots. Special care was taken while assembling the new insert so that no minor edge defects arise during assembly. RTV adhesive was used to hold the window in the cup holder and was cured for one week.

The experiments were redesigned such that in-cylinder pressure should never exceed 6.5 MPa. The advanced injection timing test cases were discarded owing to this limitation.
4.2 Powertrain Challenges

The original powertrain of the SCE-903 suffered from an alarming failure of the coupling rubber center during testing. The rubber center (highlighted by red box in the Figure below underwent fatigue failure as the operating conditions exceeded its design capacity.

Original powertrain design

The original powertrain design from the driver to the SCE-903 utilized a custom adapter, two commercial heavy duty couplings and a universal joint as shown in the Figure 42. The couplings with a rubber center not only ensured complete power transmission but also compensated for any opposing torque’s that may arise when the optical engine is fired. The detailed tolerances and specifications of the old couplings can be found in the Table 19 below. Although the tolerance on the engine were ensure to be within limits in static condition, the rubber center experienced excess vibrations and shock loading and resulted in fatigue failure.

Table 19: Original Coupling Specifications

<table>
<thead>
<tr>
<th>Coupling Model</th>
<th>Martin Quadra-Flex 6507K9</th>
</tr>
</thead>
<tbody>
<tr>
<td>Outer diameter of coupling</td>
<td>8.625 in</td>
</tr>
<tr>
<td>Max torque rating</td>
<td>3600 RPM</td>
</tr>
<tr>
<td>Max RPM rating</td>
<td>4530 in.-lbs.</td>
</tr>
<tr>
<td>Max misalignment parallel, axial (in)</td>
<td>0.032 in , 0.125 in</td>
</tr>
<tr>
<td>Max alignment angular</td>
<td>1°</td>
</tr>
</tbody>
</table>
Reasons for failure of rubber center

The universal joint (u-joint) used to transfer power from the speed reducer to the optical engine (highlighted in image above) created an excessive transverse force on the rubber center of the larger coupling on the right seen in Figure 42 above. The main reason was that the weight of the coupler could not supported by the u-joint. This unaccounted force, created an angular misalignment which was higher than the design rating of the rubber center that resulted in unbalanced rotation of the coupler and severe whirling. Furthermore the opposing torques acting on the rubber center during starting and stopping of both engines exerted a resultant
torque higher than the rated torque of the rubber center. Hence the sudden high jerk acting on the rubber center during every run, eventually resulted in fatigue failure shown in Figure 43.

![Figure 43: Sheared original coupler](image)

**Figure 43: Sheared original coupler**

*Modifications to the system*

This alarming event posed great safety concerns and the powertrain was modified, so that power from the speed reducer to the SCE-903 could be safely transmitted directly via a muff coupling and a new heavy duty coupling as shown in the Figure 44 below. The new coupling, specifically designed for frequent starting and stopping applications has a stronger hytrel rubber center which has a higher torque rating. Furthermore the replaceable rubber center does not actively transmit the torque, it only adds a cushioning effect during starting
and stopping the system. This design eliminated the root cause of failure of the rubber center by creating a secure support for the coupler from both sides and eliminating the high torques acting on the rubber center. The possibilities for shaft whirling were also eliminated ensuring that a recurrence of such unsafe event is prevented. The technical specifications of the new coupler can be found in Table 20 below.

Table 20: New coupling Specifications

<table>
<thead>
<tr>
<th>Coupling Model</th>
<th>Lovejoy Coupling 6408K22</th>
</tr>
</thead>
<tbody>
<tr>
<td>Max torque rating</td>
<td>4200 RPM</td>
</tr>
<tr>
<td>Max RPM rating</td>
<td>6225 in.-lbs.</td>
</tr>
<tr>
<td>Max misalignment parallel</td>
<td>0.015 in</td>
</tr>
<tr>
<td>Max alignment angular</td>
<td>0.5°</td>
</tr>
</tbody>
</table>

Figure 44: Modified powertrain
4.3 Oil system Challenges

Similar to most IC Engines the SCE-903 Engine also faced various issues with the oil system. The details and solution for these are discussed below.

4.3.1 Head Gasket leak

A sizable amount of oil leaking into the coolant system was observed when a thick layer of oil was seen in the coolant reservoir. The possible reasons for this leak were, a damaged head gasket, damaged oil lines or oil leaking past the threads of the head bolts. This was a serious problem and to verify, the cylinder head assembly was removed and closely examined. Some oil deposits were observed on the head gasket, leaking past the head bolts and can be seen highlighted by red circles in the Figure 45 below.

Figure 45: SCE-903 cylinder head and oil leaks
The smaller hole marked in blue are coolant passage ways and oil can be seen leaking into them past the heal bolts and the head gasket. Hence the head gasket was replaced with a new one cut out from the original VTA-903 head gasket at the mid-plane of the 5th cylinder. The head bolt threads were also sealed with thread locking material as a precautionary measure, to prevent any leaks past the threads into the cooling water jacket.

4.3.2 Other oil leaks and issues

In addition to the head gasket leak, oil leaks were also observed on the lower most part of the cylinder, past the oil sump return line. The reason for this leak was the improper fastening of the return oil plug on the bottom of the oil pan. The oil pan was detached and appropriate fastening created for the return plug along with sealants from both sides to prevent any further recurrence of such an event.

Oil line leaks and balance piston cap

The original return oil line used for the oil sump return pump was a simple high temperature rubber hose. The suction created by the oil sump pump was higher than the strength of the rubber hose and caused it to crimp and stunt the return oil supply. This caused excess oil build up in the SCE-903 sump. The excess oil in the sump lead to an increase in vibrations and other critical problems, which will be discussed in the next paragraph.

Hence the return oil rubber hose was replaced with a steel braided automotive oil hose
rated for 1000 psig pressure and 350° F temperature. An oil strainer was also installed to protect
the return pump from any metal pieces that rarely may pass through the sump.

Figure 46: New oil return line with oil strainer

*Balance piston cap*

The second balancing piston of the SCE-903 has four holes drilled in it to
remove pressure build up as shown in Figure 47 the head is replaced with an aluminum cap
which cannot with stand high pressures.
As mentioned above due to crimping of the oil return line excess oil was stored in the SCE-903 sump. This excess oil increased the oil splashed onto the cylinder liners and pistons as per the original oil system design of the SCE-903. The excess oil flow causes oil to pass through the holes drilled in the balance piston and accumulate in the chamber creating fluid pressure. The modifications to the balance piston are made such that the cap cannot withstand any pressure build-up and hence was dislodged from the engine. Such an alarming event could cause some serious injury or damage and hence to avoid it oil levels must be regularly checked.
4.4 Coolant system damage

The original coolant system for SCE-903 was retrofitted with transparent supply hoses rated for maximum temperatures of 165° F and 90 psig pressure. During early combustion experiments the input coolant line sheared due to fatigue causing a hazardous coolant leak in the laboratory. The main reason for this fatigue failure was the operating conditions of the coolant system. The SCE-903 coolant system is continuously operated at temperature ranges of 160° F - 170° F for several hours which was the maximum rated temperature for the clear hose. The long hours of operation, that were necessary for gathering combustion data, caused thermal stresses in hoses. Both the factors coupled with the wear due to corrosive coolant used caused the fatigue failure shown in Figure 48. Hence these were replaced with automotive grade radiator hoses rated for 200° F continuous operation.

(a) Old Ruptured Coolant Lines                (b) New high temperature lines

Figure 48: Coolant lines
4.5 Driver Engine Issues

The driver engine is a large bore single cylinder diesel engine with a horizontally oriented piston-cylinder. The horizontal orientation and the bulky piston assembly necessitate the use of heavy counterweights on the crank shaft. Such an engine design inherently generates high amount of secondary vibrations that may lead to several other problems. The SCE-903 test bed has faced a number of different problems owing to these unwanted secondary vibrations which will be discussed in detail in this Section.

4.5.1 Mounting base and fuel tank mounting bolt

The secondary vibrations particularly have a high impact on engine mounts. In an attempt to minimize the impact the driver engine is mounted on flexible damper pads. These are designed to absorb some of the vibrations generated by this engine. Despite the use of these mounting pads, due to the prolonged periods of operation of this engine, the secondary vibrations have caused serious damage to various components of the mounting assembly. The mounting bolts, which fasten the engine to the base, sheared due to fatigue over time. To prevent recurrence of such an event all the mounting bolts were replaced with higher grade bolts designed to yield under some amount to torque.

4.5.2 Fuel tank mounting bolt

The fuel tank mounting of the driver engine utilizes a long stud with an eye nut as shown in Figure. The eye nut is used to lift the engine while disassembly. The secondary
vibrations, induced vigorous, high amplitude low frequency, vibrations in the fuel tank which caused very high shear stress on this stud and resulted in its failure. The stud was replaced and to reduce fuel tank vibrations, fuel quantity in the tank was limited to half full.

![Image](image.png)

**Figure 49: Fuel tank mounting bolt failure**

4.5.3 **Mounting base**

The mounting base of the driver engine is fabricated by welding and bolting L-shaped brackets. One of the welds of these cast iron brackets were poor in quality and was broken due the secondary vibrations. This precarious mounting design was then replaced with a sturdier design equipped with professional welds.

4.5.4 **Radiator leaks and Exhaust muffler damage**

The driver engine radiator is a gravity flow, small tube heat exchanger. Due to prolonged use the small radiator passages were damaged. A permanent solution for this problem, was using a special sealant additive in the coolant system that enters these cracks sealing them.
As discussed in chapter 2 the exhaust gases of the driver engine are routed to the root blower via a flexible exhaust extension. The weight of this extension tube coupled with the excess vibrations of the driver engine ensued large cracks in the exhaust muffler shown in Figure 50.

![Exhaust muffler cracks](image)

**Figure 50: Exhaust muffler cracks**

A permanent solution is to completely eliminate the exhaust muffler and route the exhaust gases directly from the exhaust manifold to the large inlet pipe of the root blower. The large diameter of this pipe shall act as an exhaust muffler and dissipate any high intensity sound waves. Nevertheless implementing this solution requires fabricating special adapters which is time consuming. Hence until the adapters are fabricated a temporary solution was to use high temperature epoxy putty to close these cracks.

In conclusion the copious complications caused by the secondary engine vibrations need a permanent solution. For future scope new mounting for the driver engine that accounts for these vibrations is being designed. It should reduce difficulties caused by the driver engine greatly, and improve the stability of the SCE-903 test bed.
CHAPTER 5: RESULTS AND DISCUSSION

5.1 Operating Conditions

The SCE-903 was motored to a speed of 600 rpm with intake air supplied at room temperature and pressure through natural aspiration. Six operating cases with variations in fuel injection pressure and timing listed in Table 5.1 are presented in this thesis. The fuel quantity for injection all six cases is calibrated using the methods discussed in Section 3.2.2 to about 46mg for every fuel injection. The fuel used is a North American ultra-low sulfur diesel (ULSD) fuel and some properties listed in Table 5.2. [22] The engine is operated in skip fire mode with three injection cycles followed by ten flushing cycles.

**Table 21: Six Engine Operating conditions**

<table>
<thead>
<tr>
<th>Case Number</th>
<th>Injection pressure (bar)</th>
<th>Injection timing (CAD BTDC)</th>
<th>Injection duration</th>
<th>Injection duration</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td></td>
<td></td>
<td>Milliseconds</td>
<td>CAD (@ 600 rpm)</td>
</tr>
<tr>
<td>Case 1</td>
<td>800</td>
<td>10°</td>
<td>1.14</td>
<td>4.104</td>
</tr>
<tr>
<td>Case 2</td>
<td>1000</td>
<td>10°</td>
<td>1</td>
<td>3.6</td>
</tr>
<tr>
<td>Case 3</td>
<td>1200</td>
<td>10°</td>
<td>0.905</td>
<td>3.258</td>
</tr>
<tr>
<td>Case 4</td>
<td>800</td>
<td>5°</td>
<td>1.14</td>
<td>4.104</td>
</tr>
<tr>
<td>Case 5</td>
<td>1000</td>
<td>5°</td>
<td>1</td>
<td>3.6</td>
</tr>
<tr>
<td>Case 6</td>
<td>1200</td>
<td>5°</td>
<td>0.905</td>
<td>3.258</td>
</tr>
</tbody>
</table>

**Table 22: Ultra-Low Sulfur Diesel Fuel Properties**

<table>
<thead>
<tr>
<th>Property</th>
<th>ASTM Method</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cetane Number</td>
<td>D 976</td>
<td>43</td>
</tr>
<tr>
<td>Sulfur Content, ppm</td>
<td>D 5453</td>
<td>15 ppm</td>
</tr>
<tr>
<td>Density, Kg/m³</td>
<td>D 4052</td>
<td>876 Kg/m³</td>
</tr>
<tr>
<td>Flash Point, °C</td>
<td>D 93</td>
<td>52 °C</td>
</tr>
</tbody>
</table>
The newly developed in-cylinder pressure data acquisition system, fuel injection systems and the injection triggering systems were validated by carrying out preliminary combustion tests and ensuring optimum engine operation. The fuel system was effective in controlling fuel pressure and achieving precise fuel injection for the first fired cycle during skip-fire operation. The initial motoring pressure plots displayed in Figure 51 below correlate very well with the initial motored results published in reference 1. A peak motored pressure of 2.62 MPa is observed with the current set-up which is close to the 2.8 MPa published in the literature for SCE-903. The slight gap in motored pressure can be attributed to the different engine operating speeds of the two data sets and the poor resolution of the data acquisition system utilized.

![Figure 51: Motored In-Cylinder Pressure](image-url)
5.2 PV Diagram

The In-cylinder pressure is obtained based on 20 injection cycles. The PV diagrams for all the six cases displayed in Figure 52, depict the actual thermodynamic cycle observed in the SCE-903. This reveals a cycle closer to the fuel air cycles due to the low fuel injection quantities. It can be seen that for all six cases the work done is almost the same and is further evident from the IMEP comparisons shown in Figure 53.

![PV Diagram for all Six Cases](image)

Figure 52: PV Diagram for all Six Cases

The indicated mean effective pressures are calculate utilizing Equation 4, discussed in Section 3.4.2. A standard deviation of the 20 cycles recorded during each case is also displayed in Figure 53. The data for case 5 depicts the highest deviation of 39 kilo-pascal whereas case 3 shows the lowest deviation of 10 kilo-pascal. Overall the IMEP load for all six cases is within one standard deviation of case 5 which was the reference case for fuel calibration.
5.3 In-Cylinder Pressure

The In-cylinder pressure for all cases obtained based on 20 injection cycles and processed by utilizing methods described in Section 3.4.2 is shown in Figure 54 and Figure 55. As discussed in Section 2.10 data acquisition begins from exhaust TDC and hence zero CAD corresponds to the exhaust TDC. The compression curves are similar to the motoring run until fuel injection. After fuel injection, the peak pressure is slightly higher for case 2: 1000bar fuel pressure case with advanced injection timing. This occurs because advanced fuel injection allows better fuel vaporization and efficient fuel-air mixing, resulting in a premixed combustion. Moderate fuel pressure of 1000 bar controls the fuel atomization such that some pockets of rich fuel-air mixture are formed near the squish region.
A maximum pressure of 4.24 MPa is recorded at 11.5 CAD after compression TDC for case 2. A similar trend is observed for case 1 and case 3, where peak pressure of 3.95 MPa for both occurs at 12 CAD and 14 CAD after compression TDC respectively and are tabulated in Table 23. The delay in peak pressure is due to longer ignition delay for case 1, whereas for case 3 ignition delay is smaller than case 2 but heat release is slower due to stratified combustion. In general a faster combustion process is observed for advanced fuel injection timing as better fuel vaporization and mixing occurs well before TDC and hence slightly higher pressure is recorded. Furthermore a flat region in the combustion pressure curves as compared to motoring pressure after injection at 355 CAD is observed for advanced fuel injection timing.

Figure 54: In-Cylinder Combustion Pressures for advanced fuel injection (-10 CAD)
cases. It can be directly attributed to the heat absorbed by diesel fuel for vaporization. This is not as well pronounced in retarded injection cases shown below, but is surely present around 360° CAD.

The retarded injection cases 4 to 6 are shown in Figure 55 exhibit slower combustion rate as smaller amount of time is available for fuel vaporization and mixing. Hence for cases 4, 5 and

![Figure 55: In-Cylinder Combustion Pressures for late fuel injection (-5 CAD)](image)
6 peak pressures of 3.9 MPa, 4.0MPa and 3.82MPa respectively are observed at 15 CAD, 14 CAD and 13 CAD after compression TDC and are tabulated in Table 23.

Table 23: Results for Peak Pressure

<table>
<thead>
<tr>
<th>Case</th>
<th>Injection pressure (bar)</th>
<th>Injection timing (CAD BTDC)</th>
<th>Peak Pressure (in MPa)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Case 1</td>
<td>800</td>
<td>10°</td>
<td>3.95</td>
</tr>
<tr>
<td>Case 2</td>
<td>1000</td>
<td>10°</td>
<td>4.24</td>
</tr>
<tr>
<td>Case 3</td>
<td>1200</td>
<td>10°</td>
<td>3.95</td>
</tr>
<tr>
<td>Case 4</td>
<td>800</td>
<td>5°</td>
<td>3.9</td>
</tr>
<tr>
<td>Case 5</td>
<td>1000</td>
<td>5°</td>
<td>4</td>
</tr>
<tr>
<td>Case 6</td>
<td>1200</td>
<td>5°</td>
<td>3.82</td>
</tr>
</tbody>
</table>

**Effects of fuel rail pressure**

Fuel rail pressure has a significant impact on diesel combustion, but there is no direct correlation between the two. Higher pressure for 1000 bar fuel pressure cases is observed due to optimum fuel atomization, vaporization and mixing before combustion. Heat transfer rates and in-cylinder pressure also greatly affect this process. The resultant fuel vapor-air mixture in SCE-903 yields highest pressure for either injection timings. In case of 800 bar fuel rail pressure, fuel atomization is poorer and hence increases the ignition delay resulting in slower combustion and lower peak pressure. In case of 1200 bar fuel rail pressure fuel atomization is improved cause a shorter ignition delay, but the small amount of swirl induced by the 6 hole injector spray improves fuel vapor and air mixing greatly causing a broad heat release and reducing the peak pressure.
5.4 Apparent Heat Release Rate

The methods for heat release analysis employed in this thesis are described in Section 3.4.4. Thus utilizing Equation 5 heat release rate is computed using the 20 cycle average in-cylinder pressure data and plotted in Figure 56 below.

![Figure 56: Apparent Heat Release rate plots for all cases](image)

The negative net heat release value before compression TDC is due to the effects of heat transfer, blowby, and crevice flows. The main contributor to this term is the crevice flow which approaches zero near TDC. Assuming little change in the ring sealing effectiveness as the piston slows, the net heat release rate approaches zero which implies that blowby in SCE-903 is small.

For all six cases only single stage heat release process is observed. This indicates that premixed combustion mode is dominant for lean mixtures tested in the SCE-903 engine. The
reasons for this combustion mode is the effective fuel vapor-air mixture formed in the combustion chamber due to use of a six hole injector. Also the injector and the intake port geometries together induce a small amount of swirl in the combustion chamber which propagates fuel vaporization and fuel-air mixing leading to a premixed charge like compression ignition and rapid concentrated heat release process. Following this initial heat release a small tail of late combustion is also evident.

Case 2 exhibits the most rapid heat release rate indicated by the higher peak of 170 J/CAD at 367.5 CAD and smaller duration. Similar behavior is exhibited by case 1, 3 and 4 with a peak release rate of 145 J/CAD, 140 J/CAD and 162 J/CAD respectively. Except that the ignition delay for case 4 is longest. Efficient fuel-air mixing and excellent atomization are the main reasons for these combustion modes.

Case 3 and case 6 for 1200 bar rail pressure exhibit a broader heat release rate curve and smaller peak which indicate a slower combustion mode. It is attributed to the shorter injection duration coupled with higher fuel atomization. This results in stratified combustion, where pockets of rich fuel vapor-air mixture are smaller and hence broader heat release rate with lower peak is observed.
5.5 Ignition Delay

Ignition delay based on the change in slope of in-cylinder combustion pressure diagrams is evaluated and plotted in Figure 56. The longest ignition delay is observed for case 1 whereas the shortest ignition delay is observed for case 6. This shows that both fuel injection timing and rail pressure have a significant impact on ignition delay and subsequent peak pressure. Ignition delay is seen to progressively decrease with increase in fuel injection pressure. This is characterized by the slight shift of the pressure curves in Figure 54 and Figure 55 towards the left. Whereas ignition delay for first three cases is higher than for the last three cases as seen in the ignition delay plot in Figure 57 below. This can be explained by the excess amount of fuel vaporization time available during the first three cases as injection is well before TDC. However as fuel is injected closer to TDC shorted time is available for vaporization.

Figure 57: Ignition delay for all six cases.
5.6 Cumulative Heat Release

Integrating the apparent heat release rate over the expansion stroke estimates the cumulative heat released during the combustion process which gives an idea of the burning efficiency. The cumulative heat release plots in Figure 58 indicate ignition delay affects the cumulative heat release rate in addition to the net heat release value computed utilizing Equation 5 and 6. Since it is an integration beginning from SOI, longer ignition delay and retarded injection timing results in higher cumulative heat release as seen for case 4.

![Cumulative Heat Release](image)

**Figure 58: Cumulative Heat Release rates for all cases**
The highest cumulative heat release is observed for case 4, albeit having a low peak pressure. Whereas case 2 recorded highest peak combustion pressure has a very low cumulative heat release rate, which indicates low burning efficiency as indicated in Figure 59.

**Figure 59: Burning efficiency for SCE-903**
5.7 *High Speed Combustion Visualization*

The current optical access design to the SCE-903 combustion chamber is such that the quartz window and cup design grants a circular access to only 30% of area of the combustion chamber. The modular cup design of the window is almost half the diameter of the complete piston. The combustion high speed images for each of the six cases gathered via methods described in Section 3.3 are presented below in Figures 61 to 72. For each case three sets of complete cycle combustion images were recorded and the set with in-cylinder pressure close to average or IMEP within one standard deviation of the average were selected for presentation. All videos are recorded at 2µs exposure and an f/11 aperture for the phantom camera.

The CAD at which the image is recorded after compression top dead center (ATDC) is indicated in to top write corner of the image. Figure 61 below indicates this by example. This Figure also depicts the fuel spray from the 6 hole injector in the combustion chamber at TDC while giving a fair idea of the extent of optical access to the combustion chamber.

![Image of fuel spray high speed image]

*Figure 60: Fuel Spray High Speed Image*
A small very part of the valves is visible along with the injector tip in the center which corroborates how limited the field of view inside the combustion chamber is. The sequential arrangement of the every alternate CAD image gives a clear picture about the combustion flame for each case. Further processing of these images yields a color map of recorded luminosity values that provide quick visual information about soot development. Images from first to last luminosity are presented with lowest luminosity of 0 depicted in blue and highest of 255 depicted in red.
Figure 61: High Speed Combustion Images for case 1; 800bar, -10

Figure 62: Case 1: Comparative analysis of AHRR, In-Cylinder Pressure and combustion images
The combustion flame evolution for all the cases shows a small amount of anti-clockwise swirl due to quiescent DI combustion chamber design. Furthermore stronger combustion flames emerge from beyond the field of view for most of the cases which is why it is predicted that combustion begins near in the squish region along the periphery of the wall and then progresses towards the fuel injector. First luminosity for all cases is observed well after the end of injection thus confirming the inferences of ignition delay from the pressure and heat release plots. As a result no overlap of combustion flames with liquid fuel jet is visualized for the current operating conditions.

A weak combustion flame is observed for case 1, beginning from the periphery at 8 CAD ATDC. But the heat release plots indicate start of combustion to be 0 CAD ATDC creating a wide gap between two observations. This wide delay in luminosity visualization would be due limited view offered by the current optical access. It also strengthens the proposition that ignition occurs near the squish region and proceeds inwards to the center. Overall four ignition flames are visible, of which two opposing flames continue to be fairly luminous until 46 CAD ATDC. The top right flame weakly moves towards the center but dies out midway. Some residual luminosity can be observed until 60 CAD ATDC which coincides well with the zero heat release rate value as seen in AHRR plots in the previous section. Based on luminosity observations moderate soot generation is observed for case 1.
Figure 63: High Speed Combustion Images for case 2; 1000bar, -10

Figure 64: Case 2 - Comparative analysis of AHRR, In-Cylinder Pressure and combustion images
Stronger combustion flame is observed for the second case with highly luminous flames are first visible along the periphery of the window at 9.33 CAD ATDC. Similar to the first case a gap of about 9 CAD between SOC and luminosity visualization is due the limitations current optical view. Three flames progress inwards to the center of the combustion chamber which high localized luminosity until 44 CAD ATDC. These flames are observed until 64 CAD ATDC which agrees well with zero point on the heat release plots. Yet some luminosity for a single flame is observed until BDC as shown in the Figure above. This is due to residual oil leaking onto the optical piston when it reaches BTDC. Luminosity is highest for second case and implies higher soot formation which causes a significant rise in emissions and less efficient combustion.
Figure 65: High Speed Combustion Images for case 3; 1200bar, -10

Figure 66: Case 3 - Comparative analysis of AHRR, In-Cylinder Pressure and combustion images
A weaker combustion flame is observed for third operating case, which first luminosity beginning at 5.33 CAD ATDC. This is closer to the SOC predicted from the heat release plots but does exhibit a gap of about 7 CAD. A single flame along with small pockets of fuel-air ignition are recorded for this case. The single flame progresses along the periphery of the window until 63 CAD ATDC which also coincides with the end of combustion on the heat release rate plots.

Conclusively for advanced injection timing combustion is generally visualized between 8 CAD ATDC to 60 CAD ATDC. The soot formation in advanced fuel injection cases is higher as soot luminosity is dominant in the natural flame combustion images.
A combustion flame weaker than for case 1 is recorded during case 4. First luminosity is observed at 13.666 CAD ATDC. As compared to case 1, first light is delayed by 5 CAD which is the exact value by which fuel injection for case 4 is retarded. This shown that the ignition delay remains almost the same for 800bar fuel injection pressure. Some luminosity is observed until 73.33 CAD ATDC, which agrees with the heat release rate and in-cylinder
pressure plots. Crevice flow similar to that of second case may be observed for case 5. The combustion flame is better than case 4 with localized fuel rich flame viewed, but is in general a weaker flame. First and last luminosity for case 5 is observed at 13 and 72 CAD ATDC respectively. These observations are also seen on heat release rates and in-cylinder pressure.

Figure 69: High Speed Combustion Images for case 5; 1000bar, -5
Figure 70: Case 5 - Comparative analysis of AHRR, In-Cylinder Pressure and combustion images
Figure 71: High Speed Combustion Images for case 6; 1200bar, -5
The weakest and shortest combustion flame is visualized for case 6, which begins 14.33 CAD ATDC exhibiting a visualization delay of 13 CAD. The luminosity is observed at 74 CAD ATDC which is close to the end of combustion according to the heat release rate plots. In general for retarded injection timing luminosity is observed to begin around 13 CAD ATDC and continue until 75 CAD ATDC.
5.8 Limitations and Uncertainty Analysis

This section discusses the various limitations of the experimental processes and data analysis utilized in this body of work. Experimental methods may possess different types of sources of error which are broadly classified as systematic error and random error. Systematic errors arise due the uncertainty associated with the accuracy of a device and is known as systematic uncertainty \( (S_x) \). Whereas random error arises due to repeatability issues over time of a device and is known as random uncertainty or precision error \( (P_x) \). According to Coleman & Steele [23] these uncertainties can be combined to yield the overall uncertainty \( (U_x) \) of a device to a root sum square value.

\[
U_x = [S_x^2 + P_x^2]^{0.5} \quad \quad \quad \text{Eqn 7}
\]

Marvel B. T. [27] performed a similar uncertainty analysis on heat release rate which provides an in-depth understanding of the mathematics and the author refers the readers to this body of work for further explanations. The uncertainty in net heat release rate can be determined from the general formula (Equation 3.34 in reference 27) for uncertainty analysis to be as follows:

\[
U_{dQn}^2 = \left[ \frac{d}{d\gamma} \cdot dQ_n \right]^2 \cdot U_{\gamma}^2 + \left[ \frac{d}{dP} \cdot dQ_n \right]^2 \cdot U_{P}^2 + \left[ \frac{d}{dV} \cdot dQ_n \right]^2 \cdot U_{V}^2 \quad + \quad \left[ \frac{d}{d\delta P} \cdot dQ_n \right]^2 \cdot U_{\delta P}^2 + \left[ \frac{d}{d\delta V} \cdot dQ_n \right]^2 \cdot U_{\delta V}^2 \quad \quad \text{Eqn 8}
\]

Where \( Q_n \) is net heat release rate, \( P \) is in-cylinder pressure measurement, \( V \) is cylinder volume and \( \gamma \) is ratio of specific heats. These values are recorded as described I earlier sections.
based on CAD and pressure measurement hence the derivatives are expressed in terms of pressure and crank angle degrees (θ).

The systems utilized for pressure data acquisition consists of a pressure sensor with uncertainty of +/-0.5% FSV (Full scale value) and the electronic data acquisition cards that have an accuracy rating of 99.97% over a period of 24 hours. Hence the pressure data acquisition system has an uncertainty of 0.53% FSV which yields +/- 1.55 bar (Uₚ).

The volume is calculated based on the crank angle degree measurement from the shaft encoder that has an uncertainty of +/-5 bit over 8196 bit optical signals. Hence the uncertainty in volume measurements is small and would have a very negligible effect on the apparent heat release rate furthermore the uncertainty of dV shall be extremely small and can be neglected. Although the uncertainty in CAD measurements is 0.06 deg as determined from above information.

The heat release analysis presented in this thesis utilizes a ratio of specific heats (γ) that may vary with the change in temperature and composition of the charge. The air fuel ratio (AFR), charge temperature and pressure may also affect its value. Estimating a constant value of γ to be at 1.325 with an uncertainty of +/-0.02 to account for the neglected effects of AFR etc. and the random uncertainties.

Neglecting the effects of uncertainties in dP as they are numerical approximation and substituting the above uncertainties along with computed derivatives the uncertainty in net heat release rate is calculated to be +/-13.23% at the peak value.
Figure 73: Effects of uncertainty on AHRR

The net heat release rate is inversely affected by the uncertainty in net HRR and the uncertainty resulting for the net heat release calculations with constant value of specific heat ration provides reasonable confidence in the data presented in this thesis and validates the measurement systems and methodologies presented.
CHAPTER 6: CONCLUSIONS

A fully developed SCE-903 Optical Engine test bed with improvised sub systems and improved data acquisition is introduced at the North Carolina State University. Newly designed data acquisition and fuel injection and control systems are validated via combustion testing. Effective experimental procedures for combustion testing and analysis that are at par with the current industry standards are established for the SCE-903 test bed. High Speed visualization capabilities are also designed and validated.

A comprehensive set of diesel combustion results are acquired validating the functioning of the complete test bed. High speed images of natural flame luminosity and in-cylinder pressure data have provided key insights about the SCE-903 combustion mode which can be summarized as follows:

- Fuel injection pressure and injection timing have a significant impact on peak cylinder pressure and heat release rates. Ignition delay is reduced with increase in fuel injection pressure and by advancing fuel injection timing.
- An overlap of combustion flames with liquid fuel jet was not observed for the current operating conditions as the combustion begins after the end of injection.
- This delay causes effective fuel air mixing and results in a premixed combustion mode observed on the heat release rate diagrams.
- High speed combustion images are observed later than the peak of heat release owing to the limited field of view of the quartz optical access window. Further improvements to the window were proposed and are currently being evaluated.
The uncertainty analysis of net heat release rate indicated that the heat release calculations have a propagated uncertainty of \(\pm 13.23\%\). The highest contributor to this uncertainty is the approximation of the specific heat ratio.

The baseline operating characteristics and experimental methods for SCE-903 diesel combustion are hence established by the author. A fully functional Optical Engine test-bed is now available for combustion and fuel injector research for scholars at North Carolina State University. The baseline diesel combustion data along with the uncertainty analysis can be utilized to carry out research on alternative fuel combustion strategies or can also be utilized to calibrate new fuel injector designs.
7. REFERENCES


APPENDICES
Appendix A

Step by Step Experimental Procedures

*Fuel swap over*

Various types of fuels can be tested in SCE-903 due to its high compression ratio and flexibility of the fuel system. The procedure to change fuel is as follows:

- Drain the existing fuel from the SCE-903 fuel tank via the bottom outlet valve. Detach the fuel return line from the fuel tank and route it to a waste fuel collection container. Ensure old fuel is drained completely and refrain from operating the low pressure pump (LPP) with an empty fuel tank.
- Unscrew the fuel filter from the filter mounting plate while observing caution as filter shall be filled with fuel.
- Rinse the fuel tank with new fuel once and top up the tank.
- With the fuel filter disconnected, run the LPP for a short period of time while collecting run-off fuel from the fuel filter inlet on the mounting plate. This shall flush out the old fuel from the LPP and the fuel lines.
- Bypass the filter plate and run to LPP to flush out the old fuel from all the fuel lines.
- Screw on a clean new filter and reconnect the fuel lines. Switch on the LPP for a short while again, and collect run-off fuel from the return lines. This bathes the new filter in fuel and removes traces of old fuel from the fuel lines.
- To flush the common rail and the metal fuel lines disconnect the injector and collect the run-off fuel from the injector supply fuel line.
- Remove any air bubbles trapped in the fuel lines by bleeding from the highest point in the fuel lines, at the inlet of the High Pressure pump (HPP).

- Reconnect all the fuel lines and torque down the high pressure lines to prevent leakage. Test the fuel system with high pressure and ensure there the fuel system is free of any air bubbles, or leaks.
Appendix B

Experimental Procedure for Pressure Data Acquisition

1. Ensure both oil pumps and coolant pump are switched ON as outlined in the pre-run preparations, verify oil temperature and coolant temperature to be 170° F ± 2° F.

2. Run the LabVIEW programs for in-cylinder pressure data acquisition and the PID fuel pressure control. Select appropriate destination folder to save in-cylinder pressure data.

3. Switch ON the pressure sensor cooling water pump, and ensure the charge amplifier is in ‘operate’ mode.

4. Switch ON the black box and the speed sensor display panel. Ensure the injection switch is OFF.

5. Ensure the function generator was switched ON during pre-run check and set the injection delay and injection duration as per calculations in Section 2.9.

6. Switch ON the low pressure pump and the injection driver circuits.

7. Switch ON the high pressure pump and turn ON the exhaust root blower.

8. Gradually increase the fuel pressure to the desired fuel pressure value using the set-point input box on the PID fuel pressure control program in LabVIEW.

9. All personnel should be positioned either behind the driver engine or well in front of the SCE-903 safety shield and be clear of all rotating parts.

10. Set the driver engine throttle lever in start position and turn the key to start the engine.

11. Slowly increase the driver engine speed such that the SCE-903 attains 600 rpm ±2 rpm.

12. Allow the system to stabilize and closely observe for any abnormal behavior or sounds.

14. Switch ON the injection switch for only three combustion cycles after every 10 motoring cycles, this is skip fire operation.

15. Repeat skip fire operation for 60 combustion cycles (injection turned on 20 times). This records 20 third sequential combustion cycles from one experiment. Turn OFF the injection switch.

16. Gradually reduce the speed of the driver engine, and then turn it OFF by locking the throttle in OFF position.

17. Gradually decrease the fuel pressure until set point is set to ‘-500’. Then Turn OFF the high pressure pump.

18. Turn OFF all other systems except the one listed in the pre-run preparation list.

Experimenters are strongly advised to run any experiments in the presence of at least two personnel in the laboratory. Pay close attention to all systems during a run. If any abnormalities, excessive vibrations or unconfirmed sounds are observed abort the experiment.
Experimental Procedure for high speed Combustion imaging:

1. Complete the safety checklist and pre-run preparations.
2. Remove the mirror mounting and clean it thoroughly with optical cleaner.
3. Remove the optical window assembly and clean all surfaces thoroughly with surface cleaner and optical cleaner then blow dry with compressed air. This step is repeated before almost every combustion imaging run to minimize effects of soot build up.
4. Proceed to set up the high speed camera and its peripherals, carefully selecting the correct aperture, exposure and resolution.
5. Proceed with steps 1 through 13 outlined in above Section C.2 Ensure the camera is set to capture mode before beginning combustion.
6. Trigger fuel injection once for 3 sequential combustion cycles. Since fuel injection and high speed camera trigger signals are same, combustion for 3 cycles will be recorded.
7. While the system is running save the recorded video onto the computer drive and note the corresponding combustion pressures for reference.
8. Repeat steps 4 and 5 two more times recording 3 combustion videos in one experiment.
9. Gradually reduce the speed of the driver engine, and then turn it OFF by locking the throttle in OFF position.
10. Gradually decrease the fuel pressure until set point is set to ‘-500’. Then Turn OFF the high pressure pump.
11. Turn OFF all other systems except the high speed camera and the ones listed in the pre-run preparation list.