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Fuel type estimation using fuel system parameters

Shambhavi Balasubramanian

Purdue University

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By SHAMBHAVI BALASUBRAMANIAN

Entitled
FUEL TYPE ESTIMATION USING FUEL SYSTEM PARAMETERS

For the degree of Master of Science in Mechanical Engineering

Is approved by the final examining committee:

DR. PETER H. MECKL
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DR. GREGORY SHAVER

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Approved by Major Professor(s): DR. PETER H. MECKL

Approved by: DR. JAY P. GORE 8/4/2016

Head of the Departmental Graduate Program Date
FUEL TYPE ESTIMATION
USING FUEL SYSTEM PARAMETERS

A Thesis
Submitted to the Faculty
of
Purdue University
by
Shambhavi Balasubramanian

In Partial Fulfillment of the
Requirements for the Degree
of
Master of Science in Mechanical Engineering

May 2016
Purdue University
West Lafayette, Indiana
To,

*Appa and Amma,*

*And Pati,*

*And beloved Apru.*
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# TABLE OF CONTENTS

<table>
<thead>
<tr>
<th>LIST OF TABLES</th>
<th>vi</th>
</tr>
</thead>
<tbody>
<tr>
<td>LIST OF FIGURES</td>
<td>vii</td>
</tr>
<tr>
<td>SYMBOLS</td>
<td>xii</td>
</tr>
<tr>
<td>ABBREVIATIONS</td>
<td>xiii</td>
</tr>
<tr>
<td>ABSTRACT</td>
<td>xiv</td>
</tr>
</tbody>
</table>

1. INTRODUCTION                                     | 1   |
   1.1 Diesel vs. Biodiesel                         | 2   |
   1.2 Objectives and Motivation                    | 2   |
   1.3 Thesis Overview                              | 3   |

2. LITERATURE REVIEW                                | 4   |
   2.1 Bulk Modulus                                 | 4   |
   2.2 Density                                      | 7   |
   2.3 Sonic Speed                                  | 9   |
   2.4 Viscosity                                   | 11  |
   2.5 Biodiesel Fuel Property Variation            | 12  |
   2.6 Concluding Comments                          | 17  |

3. SIMULATION MODEL AND DATA COLLECTION             | 18  |
   3.1 Need for a Simulation Model                  | 18  |
   3.2 Fuel Modeling in GT-ISE                      | 19  |
       3.2.1 GT-ISE Library Fuel (Diesel)             | 19  |
       3.2.2 Defining Custom Fuel (Biodiesel)        | 21  |
   3.3 GT Models                                    | 24  |
       3.3.1 Pressure Rise and Bulk Modulus          | 25  |
       3.3.2 Pump Model                               | 27  |
       3.3.3 Pump Cutout                              | 32  |
       3.3.4 Combined Model                           | 35  |
           3.3.4.1 Combined Model with Pumping and Injection | 36  |
           3.3.4.2 Pumping Only                         | 39  |
   3.4 Consistency of GT Model with Test Rig Data   | 40  |
       3.4.1 Test Rig Data                            | 41  |
       3.4.2 Frequency Spectrum                       | 42  |

4. BULK MODULUS ESTIMATION                          | 45  |
4.1 Overview of the Strategy ........................................... 45
4.2 Assumptions ........................................................... 46
4.3 Filtering ............................................................... 46
  4.3.1 Signal Frequency Analysis .................................. 47
  4.3.2 Low-Pass Filtering ............................................ 49
    4.3.2.1 Effect of Filter Order ................................. 49
  4.3.3 Moving Average Filter ...................................... 53
    4.3.3.1 Effect of filter length ............................... 54
  4.3.4 Parks-McClellan Low-Pass Filter ......................... 56
4.4 Pressure Rise Estimation ......................................... 58
  4.4.1 One Moving Window Approach .............................. 60
  4.4.2 Two Moving Windows Approach ............................ 62
5. STATISTICAL ANALYSIS .................................................. 65
  5.1 Pressure Rise Estimation ...................................... 65
  5.2 Bulk Modulus Estimation ...................................... 69
  5.3 Fuel Type Determination ...................................... 70
6. CONCLUSIONS AND PROPOSED FUTURE WORK ...................... 75
  6.1 Summary ........................................................... 75
  6.2 Contributions .................................................... 76
  6.3 Recommendations for Future Work ............................ 77
LIST OF REFERENCES ...................................................... 79
A. PRESSURE RISE ESTIMATES - RIG DATA ............................ 81
  A.1 Rail Pressure = 2400 bar .................................... 81
  A.2 Rail Pressure = 1800 bar .................................... 82
  A.3 Rail Pressure = 1600 bar .................................... 83
  A.4 Rail Pressure = 1200 bar .................................... 85
  A.5 Rail Pressure = 800 bar ..................................... 86
# LIST OF TABLES

<table>
<thead>
<tr>
<th>Table</th>
<th>Description</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>3.1</td>
<td>Injector firing order for the 6-cylinder Cummins Common Rail.</td>
<td>33</td>
</tr>
<tr>
<td>5.1</td>
<td>Pressure rise estimation mean and standard deviation results for test rig data of 2400 bar, 1000 RPM, 0 SOI.</td>
<td>69</td>
</tr>
<tr>
<td>5.2</td>
<td>Best and worst case pressure rise and bulk modulus estimates.</td>
<td>70</td>
</tr>
<tr>
<td>5.3</td>
<td>Comparison of bulk modulus estimates for D2 and B100.</td>
<td>71</td>
</tr>
<tr>
<td>A.1</td>
<td>Pressure rise estimation results for case: 2400 bar, 1000 RPM, 5 SOI.</td>
<td>81</td>
</tr>
<tr>
<td>A.2</td>
<td>Pressure rise estimation results for case: 2400 bar, 1000 RPM, 10 SOI.</td>
<td>82</td>
</tr>
<tr>
<td>A.3</td>
<td>Pressure rise estimation results for case: 1800 bar, 1000 RPM, 0 SOI.</td>
<td>82</td>
</tr>
<tr>
<td>A.4</td>
<td>Pressure rise estimation results for case: 1800 bar, 1000 RPM, 5 SOI.</td>
<td>83</td>
</tr>
<tr>
<td>A.5</td>
<td>Pressure rise estimation results for case: 1800 bar, 1000 RPM, 10 SOI.</td>
<td>83</td>
</tr>
<tr>
<td>A.6</td>
<td>Pressure rise estimation results for case: 1600 bar, 1000 RPM, 0 SOI.</td>
<td>84</td>
</tr>
<tr>
<td>A.7</td>
<td>Pressure rise estimation results for case: 1600 bar, 1000 RPM, 5 SOI.</td>
<td>84</td>
</tr>
<tr>
<td>A.8</td>
<td>Pressure rise estimation results for case: 1600 bar, 1000 RPM, 10 SOI.</td>
<td>84</td>
</tr>
<tr>
<td>A.9</td>
<td>Pressure rise estimation results for case: 1200 bar, 1000 RPM, 0 SOI.</td>
<td>85</td>
</tr>
<tr>
<td>A.10</td>
<td>Pressure rise estimation results for case: 1200 bar, 1000 RPM, 5 SOI.</td>
<td>86</td>
</tr>
<tr>
<td>A.11</td>
<td>Pressure rise estimation results for case: 800 bar, 1000 RPM, 0 SOI.</td>
<td>86</td>
</tr>
<tr>
<td>A.12</td>
<td>Pressure rise estimation results for case: 800 bar, 1000 RPM, 5 SOI.</td>
<td>87</td>
</tr>
<tr>
<td>A.13</td>
<td>Pressure rise estimation results for case: 800 bar, 1000 RPM, 10 SOI.</td>
<td>87</td>
</tr>
</tbody>
</table>
### LIST OF FIGURES

<table>
<thead>
<tr>
<th>Figure</th>
<th>Title</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>2.1</td>
<td>Bulk modulus - isothermal curves [8].</td>
<td>5</td>
</tr>
<tr>
<td>2.2</td>
<td>Bulk modulus for tested fuels (PRO, RME and diesel) calculated by experimental values of density and sonic speed [9].</td>
<td>6</td>
</tr>
<tr>
<td>2.3</td>
<td>Density values for isothermal curves [14].</td>
<td>8</td>
</tr>
<tr>
<td>2.4</td>
<td>The density for tested fuels (PRO, RME and diesel) from experimental values [9].</td>
<td>8</td>
</tr>
<tr>
<td>2.5</td>
<td>Sonic speed as a function of increasing pressure [8].</td>
<td>10</td>
</tr>
<tr>
<td>2.6</td>
<td>Sonic speed for tested fuels (PRO, RME and diesel) from the experimental values from atmospheric pressure to 160 MPa and expected values for higher pressures [9].</td>
<td>10</td>
</tr>
<tr>
<td>2.7</td>
<td>Variation of density with pressure [22].</td>
<td>14</td>
</tr>
<tr>
<td>2.8</td>
<td>Variation of bulk modulus with pressure [22].</td>
<td>14</td>
</tr>
<tr>
<td>2.9</td>
<td>Variation of sonic speed with pressure [22].</td>
<td>15</td>
</tr>
<tr>
<td>2.10</td>
<td>Estimated probability density functions of the density of a group of methyls. Red: a Gaussian distribution with a mean and standard deviation set to the estimated values from the data. Purple: estimated probability density function from the measurements. Number of observations of density (N) = 1000 and bin width = 0.2 kg/m$^3$.</td>
<td>16</td>
</tr>
<tr>
<td>2.11</td>
<td>Estimated probability density functions of the isentropic bulk modulus of a group of methyls. Red: a Gaussian distribution with a mean and standard deviation set to the estimated values from the data. Purple: estimated probability density function from the measurements. Number of observations of density (N) = 1000 and bin width = 50 MPa.</td>
<td>16</td>
</tr>
<tr>
<td>2.12</td>
<td>Estimated probability density functions of the speed of sound of a group of methyls. Red: a Gaussian distribution with a mean and standard deviation set to the estimated values from the data. Purple: estimated probability density function from the measurements. Number of observations of density (N) = 1000 and bin width = 3 m/s.</td>
<td>17</td>
</tr>
<tr>
<td>3.1</td>
<td>Variation of density with temperature and pressure for the GT diesel D2 object ‘FluidLiqCompressible’.</td>
<td>20</td>
</tr>
<tr>
<td>Figure</td>
<td>Description</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>------------------------------------------------------------------------------</td>
<td>------</td>
</tr>
<tr>
<td>3.2</td>
<td>Variation of isentropic bulk modulus with temperature and pressure for the GT ‘FluidLiqCompressible’ diesel D2 object.</td>
<td>20</td>
</tr>
<tr>
<td>3.3</td>
<td>Variation of isothermal bulk modulus with temperature and pressure for the GT ‘FluidLiqCompressible’ diesel D2 object.</td>
<td>21</td>
</tr>
<tr>
<td>3.4</td>
<td>Variation of density with temperature and pressure for the custom-built methyl ester GT ‘FluidLiqCompressible’ object.</td>
<td>23</td>
</tr>
<tr>
<td>3.5</td>
<td>Variation of isothermal bulk modulus with temperature and pressure for the custom-built methyl ester GT ‘FluidLiqCompressible’ object.</td>
<td>23</td>
</tr>
<tr>
<td>3.6</td>
<td>Variation of isentropic bulk modulus with temperature and pressure for the custom-built methyl ester GT ‘FluidLiqCompressible’ object.</td>
<td>24</td>
</tr>
<tr>
<td>3.7</td>
<td>GT model to verify bulk modulus calculation.</td>
<td>25</td>
</tr>
<tr>
<td>3.8</td>
<td>Pressure rise observed in accumulator element (‘Acc’ in GT model).</td>
<td>25</td>
</tr>
<tr>
<td>3.9</td>
<td>Volume flow rate observed through the orifice (‘Orifice’ in GT model).</td>
<td>26</td>
</tr>
<tr>
<td>3.10</td>
<td>Observed changes in isentropic bulk modulus with time in the GT model.</td>
<td>27</td>
</tr>
<tr>
<td>3.11</td>
<td>Block diagram of high pressure pump.</td>
<td>29</td>
</tr>
<tr>
<td>3.12</td>
<td>Observed poppet lift of outlet control valve (OCV1) for a target pressure of 2400 bar in GT high pressure pump model.</td>
<td>30</td>
</tr>
<tr>
<td>3.13</td>
<td>Observed poppet lift of outlet control valve (OCV2) for a target pressure of 2400 bar in GT high pressure pump model.</td>
<td>30</td>
</tr>
<tr>
<td>3.14</td>
<td>Observed volume flow rate out of pump-to-rail orifice for a target pressure of 2400 bar in GT high pressure pump model. The rail flow volume is a combination of flows out of outlet valves OCV1 and OCV2.</td>
<td>31</td>
</tr>
<tr>
<td>3.15</td>
<td>Block diagram of the pump cutout model. The pump cutout contains the common rail and injectors.</td>
<td>32</td>
</tr>
<tr>
<td>3.16</td>
<td>Rail pressure variation with time in the GT pump cutout model. Because there is no pump in this model, with each injection event there is a drop in rail pressure.</td>
<td>34</td>
</tr>
<tr>
<td>3.17</td>
<td>The volume flow rate through injector nozzles in the GT pump cutout model.</td>
<td>34</td>
</tr>
<tr>
<td>3.18</td>
<td>Schematic of the Cummins Common Rail XPI fuel system [25].</td>
<td>35</td>
</tr>
<tr>
<td>3.19</td>
<td>Block diagram of the combined model with pump, rail and six injectors.</td>
<td>36</td>
</tr>
<tr>
<td>3.20</td>
<td>Pressure signal when combined model is run with pumping and injection.</td>
<td>37</td>
</tr>
</tbody>
</table>
Figure | Page
---|---
3.21 The transient portion of the pressure signal when the combined model is run with pumping and injection. | 37
3.22 The saturation portion of the pressure signal when the combined model is run with pumping and injection, along with injector needle lift to indicate the injection events and pumping mass flow rates to indicate pumping events. | 38
3.23 Pressure signal when combined model is run in a pumping only condition. | 39
3.24 Volume flow rate from pump to rail. Blue - Pump Chamber 1, Red - Pump Chamber 2. | 40
3.25 Rail pressure signal for rig data case: 2400 bar, 0 SOI, 1000 RPM. Red - indicates starts of pumping events. Dotted green - indicates starts of injection events. | 41
3.26 Frequency spectrum of test rig data for case: 2400 bar, 0 SOI, 1000 RPM. | 42
3.27 Frequency spectrum of the GT data (saturation region) for case: 2400 bar, 1000 RPM. | 43
4.1 Bulk modulus estimation strategy from rail pressure signal. | 46
4.2 Pumping and injection events over 720°. Blue upward arrows indicate pumping events. Red downward arrows indicate injection events. | 47
4.3 Power spectral density showing frequencies in rail pressure signal at 1000 RPM. The injection and pumping frequencies are at 50 Hz and 66 Hz respectively. | 48
4.4 Magnitude response for filter orders N = 20 (blue), 50 (red) and 75 (magenta). The green dashed line indicates the filter cut-off at -3 dB down (indicated by the orange dashed line). | 50
4.5 Low-pass filtered rail pressure signals for filter orders N = 20, 50 and 75. As the filter order increases, the higher frequencies are better filtered. | 51
4.6 Power spectral density showing frequencies present in the low-pass filtered signal for case 1: filter order N = 20. | 52
4.7 Power spectral density showing frequencies present in the low-pass filtered signal for case 2: filter order N = 50. | 52
4.8 Power spectral density showing frequencies present in the low-pass filtered signal for case 3: filter order N = 75. | 53
4.9 Moving average filtered rail pressure signals for filter orders L = 49 and 99. The moving average filter distorts the original signal. | 54
<table>
<thead>
<tr>
<th>Figure</th>
<th>Page</th>
</tr>
</thead>
<tbody>
<tr>
<td>4.10</td>
<td>55</td>
</tr>
<tr>
<td>4.11</td>
<td>55</td>
</tr>
<tr>
<td>4.12</td>
<td>57</td>
</tr>
<tr>
<td>4.13</td>
<td>57</td>
</tr>
<tr>
<td>4.14</td>
<td>58</td>
</tr>
<tr>
<td>4.15</td>
<td>59</td>
</tr>
<tr>
<td>4.16</td>
<td>60</td>
</tr>
<tr>
<td>4.17</td>
<td>61</td>
</tr>
<tr>
<td>4.18</td>
<td>62</td>
</tr>
<tr>
<td>4.19</td>
<td>63</td>
</tr>
<tr>
<td>4.20</td>
<td>63</td>
</tr>
<tr>
<td>4.21</td>
<td>64</td>
</tr>
<tr>
<td>5.1</td>
<td>66</td>
</tr>
<tr>
<td>5.2</td>
<td>66</td>
</tr>
<tr>
<td>5.3</td>
<td>67</td>
</tr>
<tr>
<td>Figure</td>
<td>Page</td>
</tr>
<tr>
<td>--------</td>
<td>------</td>
</tr>
<tr>
<td>5.4</td>
<td>Histogram of pressure rise estimates for two moving windows case 4: $M = 10^\circ$. The red marker indicates the mean.</td>
</tr>
<tr>
<td>5.5</td>
<td>Histogram of pressure rise estimates for two moving windows case 5: $M = 20^\circ$. The red marker indicates the mean. This case represents the maximum variability in pressure rise estimates.</td>
</tr>
<tr>
<td>5.6</td>
<td>Histogram of pressure rise estimates for two moving windows case 6: $M = 30^\circ$. The red marker indicates the mean.</td>
</tr>
<tr>
<td>5.7</td>
<td>Fuel type determination using bulk modulus approach for best case: one moving window $M = 48^\circ$.</td>
</tr>
<tr>
<td>5.8</td>
<td>Fuel type determination using bulk modulus approach for worst case: two moving windows $M = 20^\circ$.</td>
</tr>
</tbody>
</table>
SYMBOLS

\( \beta \)  bulk modulus
\( c \)  speed of sound
\( \rho \)  density
\( m \)  mass
\( V \)  volume
\( P \)  pressure
\( T \)  temperature
\( \gamma \)  specific heat ratio
### ABBREVIATIONS

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
</tr>
</thead>
<tbody>
<tr>
<td>B100</td>
<td>100% Biodiesel Fuel</td>
</tr>
<tr>
<td>CCR</td>
<td>Cummins Common Rail</td>
</tr>
<tr>
<td>D2</td>
<td>Diesel Fuel</td>
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<tr>
<td>ECM</td>
<td>Electronic Control Module</td>
</tr>
<tr>
<td>FAME</td>
<td>Fatty Acid Methyl Ester</td>
</tr>
<tr>
<td>FIR</td>
<td>Finite Impulse Response</td>
</tr>
<tr>
<td>FFT</td>
<td>Fast Fourier Transform</td>
</tr>
<tr>
<td>GT</td>
<td>Gamma Technologies</td>
</tr>
<tr>
<td>HPP</td>
<td>High Pressure Pump</td>
</tr>
<tr>
<td>IIR</td>
<td>Infinite Impulse Response</td>
</tr>
<tr>
<td>IMV</td>
<td>Inlet Metering Valve</td>
</tr>
<tr>
<td>LPF</td>
<td>Low-pass Filter</td>
</tr>
<tr>
<td>LPP</td>
<td>Low Pressure Pump</td>
</tr>
<tr>
<td>MA</td>
<td>Moving Average</td>
</tr>
<tr>
<td>PRO</td>
<td>Pure Rapeseed Oil</td>
</tr>
<tr>
<td>RME</td>
<td>Rapeseed Methyl Ester</td>
</tr>
<tr>
<td>RPM</td>
<td>Revolutions Per Minute</td>
</tr>
<tr>
<td>SOI</td>
<td>Start-of-injection</td>
</tr>
<tr>
<td>XPI</td>
<td>Extreme-high Pressure Injectors</td>
</tr>
</tbody>
</table>
ABSTRACT

Balasubramanian, Shambhavi MSME, Purdue University, May 2016. Fuel Type Estimation Using Fuel System Parameters. Major Professor: Peter H. Meckl, School of Mechanical Engineering.

A number of factors are responsible for an increased interest in alternative fuels for transportation and other uses. Alternative fuels, such as biodiesel and vegetable oil, are widely available and both fuels can be used to power a diesel engine. Biodiesel has higher bulk modulus, sonic speed, density and viscosity than regular diesel fuel. The fueling, injection timing, and fuel spray and consequently the emission characteristics of biodiesel are affected by these properties, when used in a diesel engine.

Estimation of fuel type is critical to the performance of the engine. Knowing the fuel type allows the engine controller to determine the proper fuel quantity, injection timing, injection duration, amongst other settings, to provide the best balance of performance, emissions and fuel economy. Based on the fuel type, the calorific value will change. Knowing this, the engine controller can use the best air-fuel mix and injection pressure for that type of fuel. The type of injection system and the fuel properties (such as the density of fuel, viscosity, bulk modulus and sonic speed) affect fuel injection characteristics.

The objective of this project is to estimate fuel properties that can determine whether the fuel is diesel or biodiesel. Bulk modulus is the critical parameter used for characterizing the fuel. This thesis seeks to devise suitable strategies to estimate the bulk modulus to determine the fuel type. Using sensors available on the Cummins XPI fuel system, this is accomplished by observing the effect that bulk modulus has on pressure rise associated with pumping events for different fuels. In this research, the pressure rise is extracted from filtered raw rail pressure data. It has been found
that the separation between diesel and biodiesel bulk modulus estimates is small, and with high variability in estimation of bulk modulus, the separation would reduce. This would cause ambiguity in the determination of fuel type. In the future, we also hope to be able to differentiate between different diesel and biodiesel fuel blends.
1. INTRODUCTION

A number of factors are responsible for an increased interest in alternative fuels for transportation and other uses. These include the ability to generate biofuels from locally grown feedstocks, thereby reducing dependence on foreign oil, as well as the perceived decrease in greenhouse gas emissions from fuels that are sourced from biological materials.

Alternative fuels, such as biodiesel and vegetable oil, are now widely available. Both fuels can be used to power a diesel engine. As biodiesel is developed from vegetable or animal fats, it is functionally identical to petroleum diesel. However, biodiesel generates less particulate matter (PM) and carbon monoxide (CO) than regular diesel, although nitrogen oxide (NOx) emissions generally increase. Biodiesel is most commonly sold in blends with regular diesel.

Currently, in diesel engines, fuel blends containing up to twenty percent biodiesel in volume are being used as fuel. Blends with higher biodiesel content may lead to malfunctioning of the engine and may require modifications in the current engine design in order to operate with the same efficiency [1]. Biodiesel is a renewable fuel, and biodiesel blends have the advantage of lower carbon dioxide (CO₂) emissions compared to pure diesel fuel. The most prevalent biodiesel fuels happen to be from methanol and soybean oil or rapeseed-oil-based esters.

Biodiesel fuels generally have a shorter ignition delay, a higher ignition temperature, and greater ignition pressure and peak heat release than diesel fuels [1]. However, there is similarity in the engine power outputs and combustion characteristics of biodiesel and diesel fuels. Biodiesel blends typically have reduced particulate matter and carbon dioxide emissions, but an increase in NOx emissions [2,3].
1.1 Diesel vs. Biodiesel

Biodiesel has higher bulk modulus, sonic speed, density and viscosity than regular diesel fuel. The fueling, injection timing, and fuel spray and consequently the emission characteristics of biodiesel are affected by these properties, when used in a diesel engine [4]. Biodiesel has the advantages of a high cetane number that produces a shorter ignition delay. The oxygen in the biodiesel also enhances combustion. However, biodiesel causes some loss of power as compared to diesel because of its lower heating value, and hence more fuel is required to be injected during combustion. This results in longer injection duration due to changes in the injection timing [1].

Crude oil is refined and distilled to obtain different hydrocarbon compounds including diesel fuel. On one end of the spectrum of the compounds obtained from the refining process are gases such as methane and propane, and at the other end are heavy tar and asphalt. In between the two are commercial fuels such as gasoline, kerosene, diesel fuel (D1, D2, D4) and other petroleum products like heating oil (D5, D6), and lubricants.

The type of injection system and the fuel properties (such as the density, bulk modulus, viscosity, and sonic speed) affect fuel injection characteristics. [1,5,6].

In the chapter that follows, we describe in detail various studies on the properties, characteristics and behavior of diesel and biodiesel fuel.

1.2 Objectives and Motivation

Fuel type determination is one of the first exploratory steps to complete adaptive injection control. Fueling parameters including injection quantity, injection duration and timing can be better controlled knowing the fuel type, which results in higher efficiency and performance of the fuel system.

This thesis outlines the work towards determination of fuel type (D2 vs. B100) by estimating the bulk modulus of fuel for Cummins Scania XPI injectors (high pressure,
solenoid-actuated injectors). The strategy for estimating the bulk modulus, and the corresponding confidence in its measurement, is the main focus of the thesis.

1.3 Thesis Overview

This thesis has been organized into six chapters. The current Chapter 1 provides an introduction to the problem, and also briefly discusses the differences in diesel and biodiesel characteristics. Chapter 2 provides some detail about relevant diesel and biodiesel characteristics in the form of a literature review. Chapter 3 describes simulation of the Cummins XPI fuel system using the GT-Power simulation software, and also the different simulation models that were used. Chapter 4 highlights the different approaches that were implemented to estimate the bulk modulus. Some results for different cases are discussed in this chapter, along with the performance of our strategy for the data from the model and test rig data. Chapter 5 provides a statistical analysis of the results, and provides insight for potential to use bulk modulus to determine fuel type. Chapter 6 summarizes this thesis and lists recommendations for future work. The Appendix A tabulates the results from this research for different test cases.
2. LITERATURE REVIEW

Biodiesel (B100) comes from a different source than diesel (D2). Therefore, many fuel properties vary as well, which in turn affects the injection, combustion and emission characteristics when it is used in a diesel engine. Since the main focus is on the injection characteristics, the following will outline the fuel properties of relevance to the injection process. This chapter focuses on basic definitions of these fuel properties, findings of other researchers, and some preliminary discussion on the usefulness of the fuel properties in determination of fuel type. The primary focus of the thesis is the estimation of bulk modulus.

2.1 Bulk Modulus

Bulk modulus is a measure of the compressibility of a fluid. The isentropic bulk modulus of a material can be calculated with the change in volume due to a change in pressure, which is defined as [7]:

\[
\beta_{isentropic} = -V \left( \frac{\partial p}{\partial V} \right)_{isentropic} \tag{2.1}
\]

If the system is isothermal, the same relation can be used as follows:

\[
\beta_{isothermal} = -V \left( \frac{\partial p}{\partial V} \right)_{isothermal} \tag{2.2}
\]

During pumping events, the change in volume of fluid is known, and by measuring the change in pressure in the rail, an estimate for the bulk modulus can be obtained.

It is important to know the dependence of bulk modulus on pressure and temperature for different diesel and biodiesel fuels for diesel engines in order to predict the behavior of fuel injection systems. A series of measurements taken by Payri, et al. [8] in common rail systems showed a general trend where the bulk modulus is observed
to increase with increasing pressure and decrease with increasing temperature. When comparing trends for conventional diesel, Rape Methyl Ester (biodiesel) and Arctic (Winter) fuel for a particular pressure range, the bulk modulus of conventional diesel is found to be $\sim 7\%$ lower than biodiesel. At low pressure the bulk modulus of conventional diesel is $\sim 3\%$ more than Arctic fuel and nearly the same at high pressures. The data obtained is fitted to linear coefficients (pressure varying from 0-200 MPa) and is shown in Figure 2.1.

![Figure 2.1. Bulk modulus - isothermal curves [8].](image)

A slight variation in this trend was observed in the bulk modulus measurements conducted by Nikolic, et al. [9] on these fuels: conventional diesel duel, pure rapeseed oil (PRO) and rapeseed methyl ester (RME - biodiesel), shown in Figure 2.2. The bulk modulus values are calculated from experimental values of density and sonic speed.
The data obtained from this experiment can be modeled with the following polynomial expression:

\[ \beta = \beta_0 + \beta_1 p + \beta_2 p^2. \]  

(2.3)

In both [8] and [9], the differences in the bulk modulus values is not significantly affected by the pressure rise. Also, the bulk modulus of biodiesel is higher than that of diesel. A fuel with higher biodiesel content would have a higher bulk modulus. Thus bulk modulus may be a suitable fuel characteristic to differentiate between various diesel and biodiesel blends.

![Bulk modulus for tested fuels (PRO, RME and diesel) calculated by experimental values of density and sonic speed [9].](image)

Figure 2.2. Bulk modulus for tested fuels (PRO, RME and diesel) calculated by experimental values of density and sonic speed [9].

Nikolic et al. [9] report that having a greater bulk modulus (and subsequent lower compressibility), when biodiesel is injected, the pump produces a faster pressure rise. Consequently, due to a higher sonic speed, this pressure rise propagates faster towards the injectors. Compared to unit injectors operating with diesel fuel, biodiesel produces an earlier (and faster) needle opening.
2.2 Density

Density is the mass of an object divided by its volume.

\[ \rho = \frac{m}{V} \]  \hspace{1cm} (2.4)

For the same injection conditions, there is a somewhat higher mass flow rate for fuels with higher density. Hence, the quantity of fuel in the combustion chamber after injection would be higher [8]. Density has significant effects on the performance characteristics of the engine. Other performance characteristics, such as cetane number and heating value can be related to the density making it an important fuel property [10–12]. Therefore, changes in the fuel density will affect the mass of fuel injected and would thereby influence engine output power [13].

It is important to know the dependence of density on pressure, temperature and fuel types for diesel engines in order to predict the behavior of fuel injection systems. The density values are observed to increase with pressure and decrease with temperature. When comparing trends for conventional diesel, Rape Methyl Ester (biodiesel) and Arctic (Winter) fuel for a particular pressure and temperature range, the density of biodiesel fuel is \(~5\%\) higher than the conventional diesel fuel density. Besides, Arctic fuel density is \(~2\%\) lower than reference fuel density, making the difference insignificant [14]. This can be seen in Figure 2.3.

This agrees with the results obtained by Nikolic, et al. [9] for experimental density measurements on these fuels: conventional diesel duel, pure rapeseed oil (PRO) and rapeseed methyl ester (RME - biodiesel), shown in Figure 2.4. Density \((\rho)\) can be expressed as a second-order polynomial function of pressure \((p)\) [14]:

\[ \rho = R_0 + R_1 p + R_2 p^2. \]  \hspace{1cm} (2.5)
Figure 2.3. Density values for isothermal curves [14].

Figure 2.4. The density for tested fuels (PRO, RME and diesel) from experimental values [9].
Biodiesel fuels have higher densities than that of diesel fuel, and with the increase of biodiesel concentration, the density of the blend would increase.\cite{15}. The fluid density in a chamber is a function of pressure and temperature. Shirsiar\cite{16} has found data points of rail pressure and corresponding fuel density inside the body volume. The temperature inside the injector body volume during all the injection processes is largely constant. Thus, the density is defined as a function of just the pressure.

### 2.3 Sonic Speed

The speed of sound or the sonic velocity in a fluid is the defined velocity at which an infinitesimal disturbance would propagate through a fluid. Sonic speed in the fuel is related to the compressibility of isentropic bulk modulus and density as follows:

\[
    c = \sqrt{\frac{\beta}{\rho}}
\]  

(2.6)

Whenever a pumping event or an injection occurs, pressure fluctuations are set up in the rail that can be measured via the pressure transducer on the rail. The frequency of these pressure fluctuations is directly related to the sonic speed, and therefore can be used to estimate sonic speed.

Results from measurements conducted by Payri, et al.\cite{8} suggested that the speed of sound is observed to increase with increasing pressure and decrease with increasing temperature. When comparing trends for conventional diesel and other biodiesel blends at a particular pressure range, the speed of sound is higher for fuels with higher biodiesel content and thus the speed of sound in biodiesel blends is greater than conventional diesel. This can be seen in Figure 2.5. Similar results were obtained in the measurements conducted by Nikolic, et al.\cite{9}, with these fuels: diesel fuel, pure rapeseed oil, rapeseed methylester (for the operating range of atmospheric pressure to 160 MPa) (shown in Figure 2.6).
The measured speeds of sound obtained in the experiment (c) can be accurately modelled with second-order polynomial expressions of pressure:
\[ c = A_0 + A_1 p + A_2 p^2. \] \hfill (2.7)

With an increase in pressure, the value of the sonic speed increases. Additionally, with increasing operating pressure, the difference in the sonic speed between the tested fuels decreases.

It has been verified [17] [18] that the isentropic bulk modulus can be calculated as a function of the density and sonic speed using the following relation:

\[ \beta = c^2 \times \rho. \] \hfill (2.8)

Thus, either sonic speed or bulk modulus can be used to independently estimate the other quantity, knowing the density.

### 2.4 Viscosity

Viscosity is a measure of a fluid’s internal resistance to flow. The viscosity of a fluid affects its ability to be squeezed through an orifice. Viscosity affects the size of fuel droplet, atomization quality and jet penetration, all of which affect the quality of combustion [19]. The more viscous a fluid is, the longer it will take to be pushed through an orifice.

A higher viscosity causes poorer atomization during the fuel spray. This would require more energy to pump the fuel and would wear out fuel pump components and injectors. As viscosity increases with decreasing temperature, fuels with higher viscosity also cause more issues in cold weather [15]. The density and viscosity affect the engine performance and consequently the exhaust emissions as both fuel properties have an effect not only on the fuel spray characteristic and the start of injection but also on the injection pressure [20].

Viscosity is a suitable characteristic that can be used to differentiate between diesel and biodiesel blends. The viscosities of biodiesel fuels are higher than those
of diesel fuel. The viscosity of biodiesel blends increase with the increasing biodiesel concentration [15].

The existing literature that explores effects of viscosity is mostly for the unit injector and not the common rail fuel system. As diesel and biodiesel blends have different viscosities, it may take different amounts of time to pass the plunger in a unit injector. This can be used in the fuel system by measuring the delay between commanded on time and start of injection (as measured at the onset of the pressure drop in the rail). A longer delay will be associated with a more viscous fuel. Thus viscosity would potentially be a suitable fuel characteristic, which would aid us to differentiate diesel and biodiesel blends for a unit injector fuel system.

However, in a common rail system, the effects of viscosity may not be noticeable. To confirm this, preliminary tests conducted in GT Power for diesel and biodiesel showed very small variations in injection timing between the two test cases. This difference is not significant enough to be used practically to differentiate between D2 and B100.

2.5 Biodiesel Fuel Property Variation

This research is based on using critical fuel parameters that have unique values for different fuels to help identify the fuel type. The variability of the values of each fuel parameter between different fuels is thus important to analyze. In order to analyze the variability, we gathered data of fuel parameters from various sources for different fuels. A reference paper by Tat and Van Gerper [21] consisted of an analysis of 36 different fuels. The paper characterized all the fuels on the basis of fuel parameters like density, isentropic bulk modulus and sonic speed. These fuel parameters were modeled by a common polynomial expression shown below:

$$\text{Density (g/cm}^3) = C_1T^2 + C_2TP + C_3T + C_4P + C_5,$$  \hspace{1cm} (2.9)
Speed of Sound (m/s) = \( C_1 T^2 + C_2 TP + C_3 T + C_4 P + C_5 \), \hspace{1cm} (2.10)

Bulk Modulus (MPa) = \( C_1 T^2 + C_2 TP + C_3 T + C_4 P + C_5 \), \hspace{1cm} (2.11)

where: \( T \) is the temperature in °C and \( P \) is the pressure in MPa.

To visualize the variation in the fuel parameters, the coefficients given in the reference paper were substituted into the respective polynomial expressions at a fixed temperature of 100 °C and range of pressures from 120 MPa to 240 MPa. The 36 fuels were sub-categorized into methyl biodiesel fuels, ethyl biodiesel fuels and conventional diesel fuels. Then the calculated fuel parameter data of all the fuels within each category was averaged and graphed. The density of the three categories of fuels has been plotted in Figure 2.7. The graphs show a strong similarity in the density between the ethyl and methyl biodiesels but a significant difference between the biodiesels and conventional diesel. The isentropic bulk modulus of the three categories of fuels has been plotted in Figure 2.8. The graphs show a fine difference of about 3000 bar in the bulk modulus at 2400 bar pressure between the ethyl biodiesel and methyl biodiesel, and the conventional diesel. The sonic speed of the three categories of fuels has been plotted in Figure 2.9. The graphs show a significant difference in the sonic speed between the biodiesels and conventional diesel.
Figure 2.7. Variation of density with pressure [22].

Figure 2.8. Variation of bulk modulus with pressure [22].
In order to get a clearer understanding of the variability of each parameter between different fuels it would be helpful to understand the likely individual spread of possible values of each fuel parameter for similar kinds of fuels. Having such a spread of data would help in seeing where there is an overlap between the fuel parameter values and where there is a significant difference between the values for different fuels. This would help in identifying the fuel.

Figure 2.9. Variation of sonic speed with pressure [22].

Figures 2.10, 2.11 and 2.12 show fitted normal distributions based on the estimated mean and standard deviation of the measurements for a group of methyl biodiesel fuels at a fixed temperature and pressure of 50°C and 200 MPa. Also shown are the normalized histograms derived from the measurements (normalized by the number of measurements and the bin width to given the estimated probability density).
Figure 2.10. Estimated probability density functions of the density of a group of methyls. Red: a Gaussian distribution with a mean and standard deviation set to the estimated values from the data. Purple: estimated probability density function from the measurements. Number of observations of density \( (N) = 1000 \) and bin width \( = 0.2 \text{ kg/m}^3 \).

Figure 2.11. Estimated probability density functions of the isentropic bulk modulus of a group of methyls. Red: a Gaussian distribution with a mean and standard deviation set to the estimated values from the data. Purple: estimated probability density function from the measurements. Number of observations of density \( (N) = 1000 \) and bin width \( = 50 \text{ MPa} \).
Figure 2.12. Estimated probability density functions of the speed of sound of a group of methyls. Red: a Gaussian distribution with a mean and standard deviation set to the estimated values from the data. Purple: estimated probability density function from the measurements. Number of observations of density (N) = 1000 and bin width = 3 m/s.

These plots provide the range of values for each fuel parameter that can be used to characterize or identify methyl biodiesels. Similarly, the normal distributions for other fuels would give a range of fuel parameter values that could help characterize the variability for other fuels as well.

2.6 Concluding Comments

In conclusion, the bulk modulus, sonic speed and density are properties that have the potential to distinguish biodiesel from diesel. The difference in viscosity of the two fuels does not affect the injection timing in the common rail, as it does in the unit injector, and therefore is not a fuel property that can be measured to determine the fuel type. There is inherent variability in the bulk modulus values of biodiesel due to different feedstocks. This variability sets a lower limit on expected variation of the biodiesel bulk modulus, and our ability to determine fuel type.
3. SIMULATION MODEL AND DATA COLLECTION

In this chapter, the simulation models built in GT-Suite are described. These models are helpful in modeling the fuel system studied in this thesis. The test rig data, provided by Cummins, Inc., and the simulation models are helpful in anticipating the rail conditions, and signals that will be observed.

3.1 Need for a Simulation Model

Most of the results discussed in this thesis are obtained using rail pressure data that was shared by Cummins, Inc. The data was obtained at steady-state conditions from a test rig that used Viscor (a diesel substitute) for several different rail pressures: 2400 bar, 1800 bar, 1600 bar, 1200 bar and 800 bar. The engine speed was maintained at a constant 1000 RPM, and the commanded fueling was a constant 100 mg. The fuel-on time (with respect to BTDC) was varied from 0 to 10 and 15 ms start-of-injection (SOI). The pumping quantities are, however, unknown.

Although this data is of value, it is limited to a specific set of conditions. Also, this data is available only for diesel fuel and not biodiesel. Therefore, it proved important to design a simulation model that can be used to generate data using both diesel (equivalent) and biodiesel fuel for a variety of conditions to study factors that may affect our estimation and the estimation technique. The data provided by Cummins, Inc. was vital in the modeling of the Cummins Fuel System as a check to see if the results were consistent with the data from the test rig in terms of rail and system dynamics.

GT-Suite is engine performance simulation software, from Gamma Technologies. There are two main applications in GT-Suite 7.4.0: GT-ISE (Integrated Simulation Environment) and GT-POST. The main interface is GT-ISE, where models are built,
simulation settings are declared and launched. There are pre-defined components in
the Library that can be mapped together and modified to create a system model.
Simulation results can be viewed in GT-POST, which allows you to view and process
the results of the simulation graphically.

3.2 Fuel Modeling in GT-ISE

In the GT library, several fluids are defined, which can be used as fuels, such
as refrigerants, gases, compressible and incompressible liquids. However, there is
no biodiesel fuel pre-defined in GT. GT allows its users to custom-define the fuel,
by giving the object input fluid properties like sonic speed, bulk modulus, viscosity,
density and enthalpy as a function of temperature and pressure.

3.2.1 GT-ISE Library Fuel (Diesel)

The template library for the ‘FluidLiqCompressible’ object in GT contains pre-
defined liquids including coolants, fuels, hydraulic oils, and lubricating oils. Several
diesel fuels are predefined in this template. Standard diesel D2 was used in the
simulations conducted for this thesis.

Figures 3.1, 3.2 and 3.3 show the variation with temperature and pressure of
density, isentropic and isothermal bulk modulus, respectively, for diesel fuel. These
values of bulk modulus at high pressures (1000 bar to 2400 bar) provide a reference
to compare the estimated values of bulk modulus.
Figure 3.1. Variation of density with temperature and pressure for the GT diesel D2 object ‘FluidLiqCompressible’.

Figure 3.2. Variation of isentropic bulk modulus with temperature and pressure for the GT ‘FluidLiqCompressible’ diesel D2 object.
3.2.2 Defining Custom Fuel (Biodiesel)

In order to create a custom fuel using the ‘FluidLiqCompressible’ object in GT, the biodiesel fuel property inputs for varying temperature and pressure are required. For high pressure simulations (greater than 100 bar), GT ‘FluidLiqCompressible’ object uses the following expression for density [23]:

$$\rho(P, T) = a_0 + \left[ p + a_1 T + a_2 \right]^{a_3+1} \frac{\left(a_3 + 1\right)}{a_3 + 1} + p[a_4 T^{a_5} + a_6] + a_7 \sqrt{T} \quad (3.1)$$

where: $T$ is the temperature (in K), $P$ is the pressure (in bar), $a_0$ to $a_7$ are coefficients of the density equation.

Equation (3.1), when extrapolated to a range outside the measured data range, has shown to yield more accurate density predictions [24]. The methyl biodiesel speed of sound and bulk modulus values [21] were obtained for a range of temperatures from $20^\circ C$ to $100^\circ C$ and pressure varying from 0 to 320 bar. This was extrapolated to the
engine operating range, that is, temperature varying from 50°C to 150°C and pressure varying from 800 to 2600 bar.

The coefficients $a_0$ to $a_7$ in Equation (3.1) can be generated by using a GT object called ‘FitPropDataLiqComp’. The object requires inputs of measured data, for different temperatures and pressures for any of the following: density, speed of sound, isothermal bulk modulus, or isentropic (adiabatic) bulk modulus [23]. This utility uses a nonlinear optimization routine to determine the set of coefficients $a_0$ to $a_7$ that will provide the best fit of the equation of state to the measured data. The set which yields the smallest error (square root of chi-squared error) between the input data and the prediction from the equation of state coefficients will be the final set of coefficients reported by the object. The user can also input an initial guess to the object, as a starting point for the optimization routine.

The sonic speed and bulk modulus are then computed by the ‘FluidLiqCompressible’ object, solving the Navier-Stokes equations, as a function of the density, using the following relations [23]:

$$c = \sqrt{\frac{\gamma}{\left(\frac{\partial \rho}{\partial p}\right)_T}}$$  \hspace{1cm} (3.2)

where $\gamma$ : ratio of specific heats

$$\beta_{\text{isothermal}} = \frac{\rho}{\left(\frac{\partial \rho}{\partial p}\right)_T}$$  \hspace{1cm} (3.3)

$$\beta_{\text{isentropic}} = \gamma \beta_{\text{isothermal}}$$  \hspace{1cm} (3.4)
Figure 3.4. Variation of density with temperature and pressure for the custom-built methyl ester GT ‘FluidLiqCompressible’ object.

Figure 3.5. Variation of isothermal bulk modulus with temperature and pressure for the custom-built methyl ester GT ‘FluidLiqCompressible’ object.
Using data generated from Equations (2.9), (2.10) and (2.11) for methyl biodiesels, the coefficients were found and fluid property plots for each were generated. Figures 3.4, 3.5 and 3.6 show the variation with temperature and pressure of density, isothermal and isentropic bulk modulus, respectively, for the custom biodiesel fuel. These values of bulk modulus at high pressures (1000 bar to 2400 bar) provide a reference for us to compare our estimated values of bulk modulus for biodiesel.

3.3 GT Models

To estimate the bulk modulus, the pressure rise in the rail due to pumping events is the most important part of the model. The fuel system model consists of the pump, the common rail and injectors. This section describes the different simulation models built in GT to model the Cummins Fuel System.
3.3.1 Pressure Rise and Bulk Modulus

To demonstrate the bulk modulus estimation from pressure rise, a GT model analogous to a pump-orifice-rail model was built at Purdue. Consider a GT model as shown in Figure 3.7, which consists of a ‘Chamber’ held at a high pressure, an ‘Orifice’ and an accumulator ‘Acc’. The chamber and orifice act like a pump. When the orifice opens, fuel is pumped into the accumulator. This ramps up the pressure in the accumulator element ‘Acc’ as seen in Figure 3.8. The simulation was run with diesel fuel for 1.5 cycles, and the orifice opens once every cycle. The initial accumulator pressure was set to 2400 bar. As pressure rises, volume flow rate of fuel pumped in decreases slightly, as seen in Figure 3.9. The volume pumped in every event can be found by finding the integral of the volume flow rate.

![Figure 3.7. GT model to verify bulk modulus calculation.](image)

![Figure 3.8. Pressure rise observed in accumulator element (‘Acc’ in GT model).](image)
This is an adiabatic system for which we can use Equation (2.1) to find bulk modulus. Consider the first orifice opening event:

\[
\text{‘Acc’ volume } V_s = 4.90874 \times 10^{-5} l, \\
\Delta P = 91.50 \text{ bar}, \\
\Delta V = 1.11 \times 10^{-7} l, \\
\beta_{\text{isentropic}} = V_s \frac{\Delta P}{\Delta V}, \\
= 40,556 \text{ bar.}
\]

Cross-checking with the isentropic bulk modulus plot from GT in Figure 3.10, the bulk modulus after the first pumping event is \( \sim 40,490 \text{ bar} \). Therefore, our estimate of bulk modulus for this case is accurate (with an error of +0.1630%).
Figure 3.10. Observed changes in isentropic bulk modulus with time in the GT model.

This simulation was conducted in order to check whether we can correctly estimate the bulk modulus from pressure rise without any additional effects of pump, rail or injection dynamics.

### 3.3.2 Pump Model

Figure 3.11 shows the different blocks that constitute the pump. The pump model is based on a high-pressure positive-displacement pump and was provided by Cummins Fuel Systems. There are two pumping chambers within the high-pressure pump. The main constituents of the model are listed below:

1. **Low-Pressure Pump** - This is the supply pressure for the high-pressure pump. The pressure in the low-pressure inlet volume can be changed to change pump output. In an actual engine, the pump works the same way, but with the inlet metering valve (IMV) throttling the LPP pressure down (an element not included in our model).
2. Inlet Head, Inlet Control Valve and Pump Chamber - The inlet valves are opened when the outlet valves are closed and fuel fills the pump chamber.

3. Outlet Control Valve and Outlet Head - The outlet valves open to discharge the fuel through the rail orifice.

4. Rail Orifice - The rail can be merged to the outlet end of the rail orifice. The orifice will pump fluid that is controlled by the opening and closing of the Outlet Control valves.

The ratio between pump RPM and engine RPM could be smaller, equal or greater than 1.0, depending on application requirements. For steady-state applications, it is recommended that pump speed be kept the same as the engine speed. A constant or variable low-pressure source can be used in the simulations. Although this pressure is kept constant in the simulations conducted, in actual operations, this pressure will not be constant.

Although the pressures and volumes cannot be observed in an actual pump, in the GT model, the pressures in fluid volumes and pipe elements are readily available in outputs as pressures in Inlet Head, Outlet Head and Pump Chambers. The pumped fuel volume is the accumulative volume from Outlet Control Valves 1 and 2.
Figure 3.11. Block diagram of high pressure pump.
Figure 3.12. Observed poppet lift of outlet control valve (OCV1) for a target pressure of 2400 bar in GT high pressure pump model.

Figure 3.13. Observed poppet lift of outlet control valve (OCV2) for a target pressure of 2400 bar in GT high pressure pump model.
Figures 3.12 and 3.13 show the poppet lift over one cycle (720° of crank shaft) of Outlet Control Valves OCV1 and OCV2, respectively. The target pressure that the pump must maintain is 2400 bar for this simulation and the engine speed is 1000 RPM. Each pump chamber fires every 180°, with a phase shift of 90°. So, effectively, the pump pumps every 90°. Figure 3.14 shows the volume flow rate over 720° at the rail orifice. The integral of volume flow rate for one pumping event gives us the volume of fuel pumped by that pumping event. This volume from GT is useful for calculating the bulk modulus from pressure rise.

Figure 3.14. Observed volume flow rate out of pump-to-rail orifice for a target pressure of 2400 bar in GT high pressure pump model. The rail flow volume is a combination of flows out of outlet valves OCV1 and OCV2.
3.3.3 Pump Cutout

The fuel system that this research focuses on is the Cummins Common Rail, which uses Cummins XPI injectors. The XPI (eXtreme High Pressure Injection) injectors are solenoid-actuated injectors that operate at very high pressures (of up to 3500 bar). XPI injectors have the capability to fire up to 16 pulses (although it is limited to 7 due to software) in a single injection event. Use of pilot pulses and post-injection pulses, along with the main injection pulse, can reduce noise and emissions. These fuel systems are primarily used for heavy-duty and mid-range applications.

The fuel system with the rail and six injectors, but without the pump, is called a ‘pump cutout’ model. Figure 3.15 shows a block diagram of the pump cutout model. There is a pressure sensor on the common rail that provides information about the rail pressure. The injection quantities can be controlled by commanding the fueling.

![Figure 3.15. Block diagram of the pump cutout model. The pump cutout contains the common rail and injectors.](image)

A full cycle is 720° of crank angle rotation and an injector fires every 120°. The firing sequence follows the order in which the cylinders reach the combustion stroke. If Injector 1 fires at 0°, the firing order of other injectors relative to Injector 1 is given in Table 3.1.
Table 3.1.
Injector firing order for the 6-cylinder Cummins Common Rail.

<table>
<thead>
<tr>
<th>Crank Angle</th>
<th>Injector Firing</th>
</tr>
</thead>
<tbody>
<tr>
<td>0°</td>
<td>Injector 1</td>
</tr>
<tr>
<td>120°</td>
<td>Injector 5</td>
</tr>
<tr>
<td>240°</td>
<td>Injector 3</td>
</tr>
<tr>
<td>360°</td>
<td>Injector 6</td>
</tr>
<tr>
<td>480°</td>
<td>Injector 2</td>
</tr>
<tr>
<td>600°</td>
<td>Injector 4</td>
</tr>
</tbody>
</table>

The rail pressure for the 2400 bar, 1000 RPM case of the pump cutout model is shown in Figure 3.16. The pressure drop associated with each injection event is ~40 bar. The corresponding injector flow volumes are indicated in Figure 3.17, the injectors firing in the sequence 1-5-3-6-2-4. The injected volume decreases slightly as the pressure decreases in the rail. This can be observed in Figure 3.17.
Figure 3.16. Rail pressure variation with time in the GT pump cutout model. Because there is no pump in this model, with each injection event there is a drop in rail pressure.

Figure 3.17. The volume flow rate through injector nozzles in the GT pump cutout model.
3.3.4 Combined Model

The Cummins Common Rail XPI Fuel System is shown in Figure 3.18. The three main components in this system are the low-pressure system, high-pressure pump and injection system. The low pressure pump is just represented using a low-pressure source in the combined GT model.

![Figure 3.18. Schematic of the Cummins Common Rail XPI fuel system](image)

It is important to note here that between the low-pressure pump and the high-pressure pump, there is an inlet metering valve (IMV) present. The IMV meters the amount of fuel that is taken into the pump. A fully-open IMV corresponds to a “full” pumping event, that is, maximum fuel delivery from the pump to the rail. However, in practice, for the fuel system that we are working with, there is no way of knowing how much the IMV is open. The test rig that was used to get rail pressure data has an IMV, the GT model does not. Therefore, all the pumping events in GT are “full” pumping events.
The combined fuel system model in GT consists of the high-pressure pump, common rail and six injectors. The blocks that constitute this model are shown in Figure 3.19. This model is consistent with the specifications of the Cummins Common Rail XPI Fuel System.

Figure 3.19. Block diagram of the combined model with pump, rail and six injectors.

There is no direct phasing relationship between injection and pumping events, although pumping events may be controlled by fueling requirements. Injection pulses are determined by combustion recipe requirements (e.g., single or multiple pulses, when to inject fuel and for how long, etc.). However, relative to the crank angle, there is a pumping event every $90^\circ$ and an injection event every $120^\circ$.

3.3.4.1 Combined Model with Pumping and Injection

Figure 3.20 shows the rail pressure signal during the operation of the GT combined model. The pumping quantities, being “full”, are large as compared to the injection quantities and this leads to an overall rise in the rail pressure, until the rail pressure
hits saturation. At this point, the drain in the injector opens to allow excess fuel to drain out of the rail pressure, thus maintaining saturation.

Figure 3.20. Pressure signal when combined model is run with pumping and injection.

Figure 3.21. The transient portion of the pressure signal when the combined model is run with pumping and injection.
Figure 3.21 shows the rail pressure signal in its transient state. The pressure rise for a single pumping event around a rail pressure of 2400 bar is 159 bar. Figure 3.22 shows the rail pressure signal at saturation, along with the injection and pumping events.

Figure 3.22. The saturation portion of the pressure signal when the combined model is run with pumping and injection, along with injector needle lift to indicate the injection events and pumping mass flow rates to indicate pumping events.

The pressure rise due to pumping in the drain phase (at saturation) is about 131 bar. Because of the drain flow, the pressure rise due to pumping does not reach the full capacity.
3.3.4.2 Pumping Only

A special case of the combined model is the “pumping only” case, where the injectors are turned off, i.e., they do not fire. Therefore, “full” pumping events can be observed without the injection dynamics.

Figure 3.23. Pressure signal when combined model is run in a pumping only condition.

Figure 3.23 shows the rail pressure signal in its transient state when only pumping events occur, with injections turned off. The pressure rise around a rail pressure of 2400 bar is 159 bar. Figure 3.24 shows the corresponding volume flow rate, which integrates to a volume change $\Delta V = 0.000470$ l.
Using Equation (2.1) for a system volume of 0.1181 l, then the estimate for bulk modulus $\beta_{\text{isentropic}}$ is:

$$\beta_{\text{isentropic}} = 0.1181 \times \frac{159}{0.000470} = 39,946 \text{ bar}.$$ 

From GT, the isentropic bulk modulus at 2400 bar is 39,770 bar and, therefore, this estimation is accurate (with an error of +0.443%).

3.4 Consistency of GT Model with Test Rig Data

As described in Section 3.1, the data provided to us by Cummins, Inc., although limited, is a good representation of how the fuel system behaves. In this section, the consistency of the data obtained from the test rig and the data obtained from the GT combined model will be discussed. In the subsections that follow, comparisons of the rail pressure waveforms and the frequency spectra of the two sets of data are provided.
3.4.1 Test Rig Data

Reiterating the description of the test data from Section 3.1, the test data was provided by Cummins, Inc. Viscor, a diesel substitute, was used as fuel. The rig was run at steady state for the following rail pressures: 2400 bar, 1800 bar, 1600 bar, 1200 bar and 800 bar. The engine speed was maintained at a constant 1000 RPM, and the commanded fueling was a constant 100 mg. The SOI (with respect to BTDC) was varied from 0 to 10 and 15 ms. The rail pressure sensor was the only signal that was monitored, mimicking an actual fuel system. Figure 3.25 shows two cycles of the 2400 bar, 1000 RPM test data case.

Figure 3.25. Rail pressure signal for rig data case: 2400 bar, 0 SOI, 1000 RPM. Red - indicates starts of pumping events. Dotted green - indicates starts of injection events.

Comparing Figure 3.25 with the rail pressure waveform from the GT Model simulation, as shown in Figure 3.22, we can see that pressure rise due to pumping events is
of the order of 24 bar. The GT simulation pumping events give a higher pressure rise because they are “full” pumping events. The test rig was operated at steady-state conditions where full pumping may not occur. The pump pumps enough to maintain the rail pressure. The pumping volume in each event may not be constant. In reality, “full” pumping events occur in transient conditions. For example, if there is a sudden change to high load, high pressure from a low load, the pump will pump fully to satisfy the commanded rail pressure.

3.4.2 Frequency Spectrum

The sampling rate for test rig data collection was set at 10,000 kHz. The power spectral density for the rig data was found for the entire length of the rig data, giving a frequency resolution $\Delta f = 0.61035$ Hz.

![Frequency spectrum of test rig data for case: 2400 bar, 0 SOI, 1000 RPM.](image)

Figure 3.26. Frequency spectrum of test rig data for case: 2400 bar, 0 SOI, 1000 RPM.
For comparison, a similar frequency analysis was conducted for the GT data (in the saturation region for a data length of about 0.4 s), giving a power spectral density with a frequency resolution $\Delta f = 2.4414$ Hz.

![Frequency spectrum of the GT data (saturation region) for case: 2400 bar, 1000 RPM.](image)

Figures 3.26 and 3.27 show the frequencies present in the rail pressure signal of rig data and GT data, respectively. Rig data has dominant frequencies at 50 Hz and 66.66 Hz. And GT has two main dominant frequencies at 49 Hz and 68 Hz. More detail about the methods of frequency analysis and its implications is covered in Chapter 4.

There are two important observations to be made here. First, in the GT data, the higher frequency 68 Hz has a higher amplitude, whereas in the rig data, 50 Hz has a higher amplitude. This is because the pumping pressure rises are higher in GT than they are in the rig data. Second, the GT data has less noise at higher frequencies, and only harmonics are present. The effects of noise make the rig data more challenging.
to work with, but this is more likely to be the rail pressure signal that is seen in practice and, therefore, we will be applying filtering techniques on the test rig data.
4. BULK MODULUS ESTIMATION

Fuel type determination using the bulk modulus is the main focus of this thesis. This chapter outlines the strategy and methods to obtain the bulk modulus from the rail pressure signal. The isentropic (adiabatic) bulk modulus, given by Equation (2.2), is expressed as:

$$\beta_{\text{isentropic}} = V_s \frac{\Delta P}{\Delta V}.$$  

By estimating the pressure rise due to pumping ($\Delta P$) and volume of fuel pumped in the event ($\Delta V$), the bulk modulus can be estimated. In the sections that follow, methods to obtain the pressure rise from the rail pressure signal will be discussed.

4.1 Overview of the Strategy

There is a continuous flow of data coming to the engine ECM from the rail pressure sensor. This must be processed in real time along with various other operations that the engine ECM must perform, such as controlling the fueling, injection timing, and engine diagnostics. Therefore, computational complexity, and speed of the algorithm are the most important criteria in selecting a method for finding the bulk modulus.

Fig 4.1 illustrates the strategy for finding the bulk modulus from the rail pressure signal:

1. Filter - signal smoothening, to remove oscillations and noise
2. Identify clean pumping events
3. Calculate the pressure rise ($\Delta P$) for these events
4. Calculate the bulk modulus
4.2 Assumptions

Before we begin to demonstrate the bulk modulus estimation strategy, it is important to state the assumptions made in the process:

1. All pumping events pump the same volume into the rail.
2. There is no leakage in pumping or injection.
3. The partial pumping events from the test data can be scaled to “full” pumping events using data from GT simulations.
4. The pumping volume $\Delta V$ is a known quantity. In the GT simulations, this is found by integrating the volume flow rate through the pump-to-rail orifice.

4.3 Filtering

The rail pressure signal obtained from the rail pressure sensor is noisy and contains high-frequency oscillations due to rail dynamics. In order to extract useful information from this signal in a form that enables easy measurement of $\Delta P$, all the undesirable frequencies need to be removed. Signal filtering is the process of removing or suppressing unwanted signal components.

In simple terms, a filter is a device that discriminates what passes through it according to some attribute of the objects applied at its input [26]. Based on the duration of the impulse response of the digital filter, there are two main types -
Finite Impulse Response (FIR) filters and Infinite Impulse Response (IIR) filters. FIR filters are always stable and have linear phase. Although a larger filter order may be required to achieve the same performance as an IIR filter, FIR filter transients have finite duration, and the design methods are linear and can be realized in hardware [27].

4.3.1 Signal Frequency Analysis

Before analysing the rail pressure signal, let us have a look at the expected dominant frequencies. In order to make this expectation more generic, we can relate the frequencies to a known quantity, such as the engine speed (in RPM). In the most ideal rail conditions, there is a pumping event every 90° and an injection event every 120°. Figure 4.2 shows pumping and injection events occurring relative to crank angle over 720°.

![Figure 4.2. Pumping and injection events over 720°. Blue upward arrows indicate pumping events. Red downward arrows indicate injection events.](image)

One revolution of the crank shaft is 360°. In every half-cycle (i.e., one revolution of the crankshaft), there are four pumping events and three injection events. If the engine RPM is known, the injection and pumping frequencies can be computed. Hence, the cut-off frequencies will be dependent only on the engine RPM. Therefore, the pumping and injection frequencies can be found as follows:
The engine speed in all the test rig data cases was constant at 1000 RPM, and so, $F_{\text{inj}} = 1000 \times \frac{3}{60} = 50 \text{ Hz}$, and $F_{\text{pump}} = 1000 \times \frac{4}{60} = 66 \text{ Hz}$. This can be verified in Figure 4.3, which shows the signal frequency analysis for 2400 bar, 1000 RPM. For 1000 RPM, we note that there are dominant frequencies at 50 Hz and 66 Hz (injection and pumping frequencies, respectively).

Figure 4.3. Power spectral density showing frequencies in rail pressure signal at 1000 RPM. The injection and pumping frequencies are at 50 Hz and 66 Hz respectively.
To obtain an estimate of what frequencies are present in the signal, a frequency analysis of the pressure signal needs to be performed. The frequency analysis will also provide an idea of the dominant frequencies and will help determine the cut-off frequencies for filtering. The signal frequencies can be found in several ways, the most popular being the Fast Fourier Transform (FFT). A ‘periodogram’ power spectral density distribution was used in MATLAB to obtain the spectrum of the time-series signal, i.e., the rail pressure signal.

4.3.2 Low-Pass Filtering

The frequencies of interest, the injection frequency \( F_{inj} \) and pumping frequency \( F_{pump} \), lie on the lower side of the frequency spectrum. The oscillations and noise have higher frequency values. So in order to remove the high frequencies, a low-pass filter can be used. The cut-off frequency of the low-pass filter needs to be greater than the highest frequency that is to be retained - the pumping frequency.

Choosing

\[ F_c = 2.5 \times F_{inj}, \]

(4.1)

in our low-pass filter implementations, we chose the window-based filtering method and used a Blackman window, as the Blackman window has a smoother response [27]. The ‘fir1’ function in MATLAB was used to implement the window-based filter design.

4.3.2.1 Effect of Filter Order

The magnitude responses of low-pass filters of different orders are compared in Figure 4.4. This was plotted using the ‘fvtool’ (Filter Visualization Tool) in MATLAB. As the filter order increases, the response becomes sharper, i.e., a low-pass filter of higher order more effectively removes frequencies above the cut-off frequency \( F_c \).
Figure 4.4. Magnitude response for filter orders $N = 20$ (blue), 50 (red) and 75 (magenta). The green dashed line indicates the filter cut-off at -3 dB down (indicated by the orange dashed line).

Figures 4.5(a), 4.5(b) and 4.5(c) show the filtered rail pressure signals for different filter orders. The rail pressure target is 2400 bar, and engine speed is 1000 RPM. The low-pass filters were designed with cut-off frequency at 125 Hz.
Figure 4.5. Low-pass filtered rail pressure signals for filter orders $N = 20$, 50 and 75. As the filter order increases, the higher frequencies are better filtered.
The post-filtered frequencies for these rail pressure signals can be found in Figures 4.6, 4.7 and 4.8. The sampling rate for test rig data collection was set at 10,000 kHz. The power spectral density for the rig data was found for the entire length of each case of rig data, giving a frequency resolution $\Delta f = 0.61035$ Hz.

**Figure 4.6.** Power spectral density showing frequencies present in the low-pass filtered signal for case 1: filter order $N = 20$.

**Figure 4.7.** Power spectral density showing frequencies present in the low-pass filtered signal for case 2: filter order $N = 50$. 
Figure 4.8. Power spectral density showing frequencies present in the low-pass filtered signal for case 3: filter order $N = 75$.

Note that although the filter of order $N=75$ does a better job of filtering above cut-off, it has a greater computational cost. Therefore, it is essential to ensure that the most optimum filter order is chosen.

4.3.3 Moving Average Filter

As the name implies, the moving average filter operates by averaging a number of points from the input signal to produce the corresponding point in the output signal. The output as a function of the inputs is given as [28]:

$$y[i] = \frac{1}{L} \sum_{j=1}^{L} x[i + j]$$  \hspace{1cm} (4.2)

The moving average filter is optimal for reducing random noise while retaining a sharp step response [28].
4.3.3.1 Effect of filter length

The test rig data was recorded at a sampling frequency of 10 kHz. This means we have 10,000 digitized data points for 1 second of rail pressure.

We can find the crank angle equivalent of a length of data points by using the relation:

\[ \text{crank angle} = \text{engine speed} \times \text{length} \times \frac{360}{60} \times \frac{1}{10k} \]  \hspace{1cm} (4.3)

Figures 4.9(a) and 4.9(b) show the implementation of Moving Average Filters of different lengths. Though higher averaging lengths give smoother waveforms, there is a noticeable change in the signal.

(a) Moving average filtered rail pressure signal for case 4: filter length L = 49.

(b) Moving average filtered rail pressure signal for case 5: filter length L = 99.

Figure 4.9. Moving average filtered rail pressure signals for filter orders L = 49 and 99. The moving average filter distorts the original signal.
Figure 4.10. Power spectral density showing frequencies present in the moving average filtered signal for case 4: filter order $L=49$.

Figure 4.11. Power spectral density showing frequencies present in the moving average filtered signal for case 5: filter order $L=99$. 
Looking at this closer in the frequency spectra in Figures 4.10 and 4.11, it is evident that, as the averaging window length increases, the higher frequencies are attenuated further.

Note that although the filter of order L=99 appears to do a better job of filtering, it reduces the magnitude of the frequencies of interest. This can be seen in Figure 4.9(b) as “over-smoothening” of the signal and loss of some valuable information about the pressure rise. Therefore, it is essential to ensure that the most optimum filter order is chosen. In general, the moving average filter is a very poor low-pass filter, due to its slow roll-off and poor stopband attenuation.

4.3.4 Parks-McClellan Low-Pass Filter

The Parks-McClellan algorithm is used to design optimal FIR filters. The algorithm minimizes the error in the pass-bands and stop-bands by using the Chebyshev approximation. The ‘firpmord’ function in MATLAB estimates the Parks-McClellan optimal FIR filter order and ‘firpm’ function implements a Parks-McClellan filter.

For a low-pass filter with pass-band edge frequency of 100 Hz and stop-band frequency of 300 Hz, pass-band ripple of 0.01, stop-band ripple of 0.1, and a sampling frequency of 10,000 Hz, the optimum filter order found by this algorithm is N = 68. Figure 4.12 shows the magnitude response of the frequency response of this filter.
Figure 4.12. Magnitude response of a Parks-McClellan low-pass filter for case 6: filter order $N = 68$.

Figure 4.13. The filtered waveform for an optimum low-pass filter of order $N = 68$. 
Figures 4.13 and 4.14 show the filtered waveform and post-filtered frequencies for the rig data case: 2400 bar, 0 SOI and 1000 RPM. Since this is the most optimum filter, all the results in the sections that follow and in Chapter 5 use this filter.

4.4 Pressure Rise Estimation

There are eight pumping events and six injection events in 720°. Referring back to Figure 4.2, which depicts graphically the timeline of simultaneous injection and pumping in the rail, there is only one clean or “full” pumping event in every revolution of the crankshaft, i.e., there is only one pumping event uninterrupted by an injection event. The pressure rise estimate corresponding to this full pumping event is the only estimate that can be used.

After obtaining the filtered rail pressure signal, we have a clean enough waveform to find the pressure rise estimates. In Figure 4.15 is shown the filtered rail pressure
signal. The markers indicate the start of pumping events. An accurate estimate of the start of the injection and pumping is required to get an accurate estimate of pressure rise.

Figure 4.15. Low-pass filtered rail pressure signal for case 2: 2400 bar, 1000 RPM, 0 SOI, N=68. The red markers indicate start of pumping events.

Two approaches to find pressure rise $\Delta P$ were explored: the one moving window approach and the two moving window approach.
4.4.1 One Moving Window Approach

In this approach, there is one window that moves along the length of the signal. The window size is about 60 degrees. This is illustrated in Figure 4.16.

Figure 4.16. Illustrated implementation of the one moving window approach on filtered rail pressure signal.

The window calculates the difference between the first and last points on the pressure signal. The maximum difference corresponds to the peak pressure rise:

\[ \Delta P = \text{Max}(\text{Last} - \text{First}) \]  \hspace{1cm} (4.4)

A graphical demonstration of this algorithm on the test rig data for 2400 bar, 1000 RPM is shown in Figure 4.17.
Figure 4.17. Demonstration of one window approach. The filtered rail pressure signal is shown along with the plot of pressure rise estimates obtained per window. The maximum of these pressure rises per window corresponds to the maximum $\Delta P$.

A statistical comparison of the effects of window size is discussed in Chapter 5.
4.4.2 Two Moving Windows Approach

In this approach, the window size is smaller, 30 degrees. This is illustrated in Figure 4.18.

![Illustrated implementation of the two moving windows approach on filtered rail pressure signal](image)

Figure 4.18. Illustrated implementation of the two moving windows approach on filtered rail pressure signal

This window moves along the length of the pressure signal and finds the difference between the first and last points. This difference gives us an idea of the nature of the gradient in the window. This is then compared to the value obtained from the previous window. A minimum is found when this gradient changes from negative to positive. Once the minimum is found, the window moves ahead to look for the maximum where the gradient changes from positive to negative. This is graphically depicted in Figures 4.19 and 4.20.
Figure 4.19. Minimum point is found as a change of gradient from negative to positive [18].

Figure 4.20. Maximum point is found as a change of gradient from positive to negative [18].

When the minimum and maximum values are found, the pressure rise is found as the difference between maximum and minimum:

\[ \Delta P = Maximum - Minimum \] (4.5)

A graphical demonstration of this algorithm on the test rig data for 2400 bar, 1000 RPM is shown in Figure 4.21.
Figure 4.21. Demonstration of two moving window approach. The filtered rail pressure signal is shown along with the plot of pressure rise estimates obtained per window. The maximum $\Delta P$ is the pressure rise estimate with the highest magnitude.

A statistical comparison of the effects of window size is discussed in Chapter 5.
5. STATISTICAL ANALYSIS

The ability to differentiate between fuel types using bulk modulus relies heavily on the difference between the estimated values of bulk modulus of the two fuels. The smaller this difference, the larger is the ambiguity in determining fuel type. There are several sources of variability associated with pressure rise estimation and the corresponding bulk modulus estimation from pressure rise. The variability due to the estimation technique, that is, impact of windowing techniques and window lengths is explored in this chapter. The effect that this variability has on the fuel type determination is also discussed in the sections that follow.

5.1 Pressure Rise Estimation

In order to get a better sense of which windowing approach and window size gives better pressure rise estimates, histograms of the pressure rise estimate were generated for the test rig data. The $\Delta P$ estimates over the entire length of the data were calculated, and their average and standard deviation were found. Figures 5.1 to 5.6 show a few cases of varying window lengths that were analyzed for test rig data of 2400 bar, 0 SOI and 1000 RPM. The bin width was selected as one standard deviation.
Figure 5.1. Histogram of pressure rise estimates for one moving window case 1: $M = 48^\circ$. This case has the lowest variability. The red marker indicates the mean.

Figure 5.2. Histogram of pressure rise estimates for one moving window case 2: $M = 60^\circ$. The red marker indicates the mean.
Figure 5.3. Histogram of pressure rise estimates for one moving window case 3: $M = 72^\circ$. The red marker indicates the mean.

Figure 5.4. Histogram of pressure rise estimates for two moving windows case 4: $M = 10^\circ$. The red marker indicates the mean.
Figure 5.5. Histogram of pressure rise estimates for two moving windows case 5: $M = 20^\circ$. The red marker indicates the mean. This case represents the maximum variability in pressure rise estimates.

Figure 5.6. Histogram of pressure rise estimates for two moving windows case 6: $M = 30^\circ$. The red marker indicates the mean.

The same results are summarized in Table 5.1. The one-window approach shows a smaller standard deviation than the two-window approach. The $60^\circ$ window gives
the smallest spread amongst the one-window cases and both the $10^\circ$ and $30^\circ$ window gives the smallest spread amongst the two-window cases.

Table 5.1.
Pressure rise estimation mean and standard deviation results for test rig data of 2400 bar, 1000 RPM, 0 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average $\Delta P$ [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>24.805</td>
<td>0.6565</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>25.953</td>
<td>1.1233</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>26.16</td>
<td>0.7025</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>25.069</td>
<td>1.2087</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>24.754</td>
<td>1.2337</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>25.069</td>
<td>1.2087</td>
</tr>
</tbody>
</table>

Appendix A has more details of results of $\Delta P$ for other cases of rig data.

5.2 Bulk Modulus Estimation

A full pumping event (from GT) corresponds to 159 bar (Section 3.3.3). The average of pressure rise estimate ($\Delta P$) from the rig data for the case of one window of length $M = 48$ deg is 24.805 bar. Therefore,

$$\text{Percentage of IMV open} = \frac{24.805}{159} = 15.60\%$$

Therefore, the highest pressure rise is associated with 15.60% of fully open IMV. Scaling the pressure rise estimates to a full pumping event, we have $\Delta P = 24.805 \times 6.41$ bar = 159 bar. The system volume $V_s = 181.10$ cc = 0.1181 l, and the volume of fuel pumped into the rail (found by integrating the volume flow rate from the GT model) is 0.000470 l.
The isentropic bulk modulus can be calculated as follows:

\[
\beta_{\text{isentropic}} = V \frac{\Delta P}{\Delta V} = 0.1181 \times \frac{159}{0.000470} = 39,946 \text{ bar}
\]

The isentropic bulk modulus at 2400 bar for D2 is 39770 bar. This implies that our bulk modulus estimate is accurate with a +0.4430% error.

5.3 Fuel Type Determination

Section 2.5 discusses the variability in methyl biodiesel values. This data represents the inherent variability in biodiesel due to different feedstocks. Using the best-case results from the one-window approach, we have found the bulk modulus corresponding to each pressure rise using the method shown in Section 5.2. The \(\Delta P\) error propagates to the \(\beta\) value as we calculate the bulk modulus from its corresponding pressure rise estimate. Table 5.2 shows the best and worst case pressure estimates and the corresponding bulk modulus estimates.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Best Case</th>
<th>Worst Case</th>
</tr>
</thead>
<tbody>
<tr>
<td>Window Type</td>
<td>One</td>
<td>Two</td>
</tr>
<tr>
<td>Window Length</td>
<td>48°</td>
<td>20°</td>
</tr>
<tr>
<td>(\Delta P) average [bar]</td>
<td>24.805</td>
<td>24.754</td>
</tr>
<tr>
<td>(\Delta P) standard deviation [bar]</td>
<td>0.6565</td>
<td>1.2337</td>
</tr>
<tr>
<td>(\beta) average [bar]</td>
<td>39,946</td>
<td>39,946</td>
</tr>
<tr>
<td>(\beta) standard deviation [bar]</td>
<td>686.97</td>
<td>1954.40</td>
</tr>
</tbody>
</table>
Figures 5.7(a) and 5.8(a) show the histogram distributions of estimated bulk modulus of D2 with the theoretical value of bulk modulus for biodiesel B100. The variability in B100 represented here is the inherent variability in biodiesel due to different feedstocks as shown in Section 2.5. Figures 5.7(b) and 5.8(b) show the histogram distributions of estimated bulk modulus of D2 with the estimated values of bulk modulus for biodiesel B100, using the mean of B100 and standard deviation associated with the diesel estimation method. Table 5.3 lists the mean and standard deviation found for the best and worst case bulk modulus histograms for diesel and biodiesel.

Table 5.3.
Comparison of bulk modulus estimates for D2 and B100.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Best Case</th>
<th>Worst Case</th>
</tr>
</thead>
<tbody>
<tr>
<td>Window Type</td>
<td>One</td>
<td>Two</td>
</tr>
<tr>
<td>Window Length</td>
<td>48°</td>
<td>20°</td>
</tr>
<tr>
<td>D2 (\beta) average [bar]</td>
<td>39,946</td>
<td>39,946</td>
</tr>
<tr>
<td>D2 (\beta) standard deviation [bar]</td>
<td>1057.5</td>
<td>1991.30</td>
</tr>
<tr>
<td>B100 (\beta) average [bar]</td>
<td>44672.53</td>
<td>44672.53</td>
</tr>
<tr>
<td>B100 (\beta) standard deviation [bar]</td>
<td>1092</td>
<td>1845.60</td>
</tr>
</tbody>
</table>
(a) Histogram distribution of bulk modulus estimate for diesel D2 and theoretical biodiesel B100 for case: one moving window approach $M = 48^\circ$.

(b) Histogram distribution of bulk modulus estimates for diesel D2 and biodiesel B100 for case: one moving window approach $M = 48^\circ$.

Figure 5.7. Fuel type determination using bulk modulus approach for best case: one moving window $M = 48^\circ$. 
(a) Histogram distribution of bulk modulus estimate for diesel D2 and theoretical biodiesel B100 for case: two moving windows approach $M = 20^\circ$.

(b) Histogram distribution of bulk modulus estimates for diesel D2 and biodiesel B100 for case: two moving windows approach $M = 20^\circ$.

Figure 5.8. Fuel type determination using bulk modulus approach for worst case: two moving windows $M = 20^\circ$. 
The separation between the estimated values of diesel D2 and biodiesel B100, as seen in the best-case results of Figure 5.7, would mean we may be able to distinguish between the two fuels based on the calculated values of bulk modulus. Biodiesel blends will lie in between the estimates for D2 and B100. This would, however, not be true if our estimates were closer to the actual values, as shown in the worst-case results of Figure 5.8, where the D2 and B100 estimates are overlapping. In addition to this, there is variability associated with the pumping volume $\Delta V$ variation. There will also be variability in estimating this volume pumped. As the variability increases, the width of the histograms will increase. This may lead to the overlap of the diesel and biodiesel estimates and cause ambiguity in fuel type determination.
6. CONCLUSIONS AND PROPOSED FUTURE WORK

The work presented in this thesis can be categorized into three parts: fuel system modeling and simulation, pressure rise and bulk modulus estimation techniques, and fuel type determination using the bulk modulus. These categories are summarized below, along with contributions and recommendations for future work.

6.1 Summary

Fuel type determination is the first step to complete adaptive injection control. Fueling parameters including injection quantity, injection duration and timing can be controlled by knowing the fuel type. This results in better performance of the fuel system.

Biodiesel (B100) comes from different feedstocks and has different fuel properties from diesel. Therefore, the injection, combustion and emission characteristics are different for biodiesel. The key fuel properties of interest are the bulk modulus, sonic speed, density and viscosity. In this thesis, the main focus was the estimation of the bulk modulus. The bulk modulus is given by:

\[ \beta = \frac{V_s \Delta P}{\Delta V} \]

Therefore, by finding the pressure rise due to pumping, and the volume of fuel in a pumping event, the bulk modulus estimate can be calculated. In order to do this, a fuel system model in GT-Suite comprising of a high-pressure pump, common rail and six injectors was developed. Along with this, there was test rig data for different rail pressures at 1000 RPM. We also modelled biodiesel fuel in GT. We conducted a frequency analysis on the rig data and the rail pressure from the GT model to ensure their consistency.
In order to determine the bulk modulus, a strategy was developed in which the rail pressure signal is filtered to remove oscillations and noise and then the pressure rise is estimated. Signal frequency analysis helped us determine the injection ($F_{inj}$) and pumping ($F_{pump}$) frequencies in terms of engine speed. This was useful in determining which frequencies are to be retained, and which ones need to be filtered out.

After analyzing the performance of low-pass filters of different orders and moving average filters of different lengths, it was found that the Parks McClellan low-pass filter is the most effective. Therefore, a low-pass filter with an order $N=68$ was chosen.

For estimating the pressure rise, two techniques were proposed. The first, one-window approach was basically a large window that moves along the length of the rail pressure signal and finds the difference between the first and last points. The maximum difference gives the maximum pressure rise. The second, two-window approach uses a smaller window that first finds the minimum and then the maximum. The difference of the maximum and minimum gives the pressure rise.

The pressure rise estimates obtained by varying window lengths were statistically analyzed for both the one-window and the two-window approach. Overall, the two-window approach gave a larger standard deviation.

Using the best-case and worst-case estimates for 2400 bar, 1000 RPM, we found the bulk modulus estimates for diesel and compared it with estimated values of biodiesel as shown in Figures 5.7 and 5.8. The separation between biodiesel and diesel dictates our ability to differentiate the fuel type, using the bulk modulus metric.

### 6.2 Contributions

The contributions of this thesis towards fuel type determination include:

1. Characterizing fuel system parameters and determining the fuel properties that affect injection characteristics.

2. Studying the inherent variability in biodiesel fuel characteristics due to different feedstocks, for both methyl and ethyl esters, and quantifying this variability and
finding an optimum mean value for isentropic bulk modulus, density and sonic speed.

3. Integrating the existing GT models and building a complete system model comprised of the high pressure pump, common rail and six injectors, and also building a fuel model for methyl biodiesel.

4. Demonstrating the technique of calculating the isentropic bulk modulus for the combined simulation model with pumping only, and a simplified simulation model with pumping only.

5. Developing a strategy for finding the bulk modulus by filtering the pressure signal and finding the pressure rise.

6. Analyzing statistically the performance of different windowing techniques in finding pressure rise $\Delta P$.

7. Demonstrating the performance of the algorithm developed to determine fuel type by estimating the isentropic bulk modulus for diesel D2 and methyl biodiesel B100.

6.3 Recommendations for Future Work

The research relating to the bulk modulus method for determining the fuel type is one of the first steps to fuel type determination using fuel system parameters. Though the results in this thesis do not indicate that the bulk modulus is a promising metric, there are several aspects of the estimation technique that need to be explored further.

1. The existing combined GT model is assembled by combining different Cummins models. It is very complex and takes several hours to execute one run. This needs to be simplified in order to reduce computation time and to conduct more tests on the effectiveness of the bulk modulus estimation strategy. With a simpler model, we can also conduct tests for different biodiesel blends.
2. Other fuel system parameters like sonic speed and density also look promising. A suitable estimation strategy for estimating these parameters would be useful to determine fuel type. There is also potential to combine information from several different fuel property estimates to determine the fuel type.
LIST OF REFERENCES
LIST OF REFERENCES


[23] Gamma Technologies. GT-Suite 7.4.0 Help Documentation.


A. PRESSURE RISE ESTIMATES - RIG DATA

In this section are pressure rise estimation results tabulated for different windowing techniques and window sizes for the test rig data. In general, the two-window approach gives a higher estimate for pressure rise. As for the variability, it is inconclusive as to which method performs better. Also with higher rail pressure, the pressure rise is also higher.

A.1 Rail Pressure = 2400 bar

Tables A.1 and A.2 are $\Delta P$ estimates for 2400 bar case, with 5 ms and 10 ms SOI delay, respectively.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average $\Delta P$ [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>24.421</td>
<td>1.0434</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>24.889</td>
<td>0.90337</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>24.146</td>
<td>1.3753</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>25.261</td>
<td>0.70663</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>25.282</td>
<td>0.70179</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>25.286</td>
<td>0.69729</td>
</tr>
</tbody>
</table>
Table A.2.
Pressure rise estimation results for case: 2400 bar, 1000 RPM, 10 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average $\Delta P$ [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>23.944</td>
<td>0.87182</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>23.573</td>
<td>1.1805</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>23.847</td>
<td>1.2315</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>24.403</td>
<td>1.0109</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>24.451</td>
<td>0.96185</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>24.562</td>
<td>0.97874</td>
</tr>
</tbody>
</table>

A.2 Rail Pressure = 1800 bar

Tables A.3, A.4 and A.5 are $\Delta P$ estimates for 1800 bar case, with 0 ms, 5 ms and 10 ms SOI delay, respectively.

Table A.3.
Pressure rise estimation results for case: 1800 bar, 1000 RPM, 0 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average $\Delta P$ [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>22.747</td>
<td>1.0047</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>23.256</td>
<td>0.96371</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>23.043</td>
<td>1.0066</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>23.629</td>
<td>0.82316</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>23.703</td>
<td>0.83775</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>23.703</td>
<td>0.83775</td>
</tr>
</tbody>
</table>
Table A.4.
Pressure rise estimation results for case: 1800 bar, 1000 RPM, 5 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average ΔP [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>21.974</td>
<td>0.5843</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>22.270</td>
<td>0.63547</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>22.467</td>
<td>0.91245</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>22.578</td>
<td>0.7849</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>22.610</td>
<td>0.81435</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>22.61</td>
<td>0.81435</td>
</tr>
</tbody>
</table>

Table A.5.
Pressure rise estimation results for case: 1800 bar, 1000 RPM, 10 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average ΔP [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>22.080</td>
<td>0.87785</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>22.212</td>
<td>0.8361</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>22.176</td>
<td>0.83822</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>22.584</td>
<td>0.86327</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>22.516</td>
<td>0.90875</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>22.533</td>
<td>0.91074</td>
</tr>
</tbody>
</table>

A.3 Rail Pressure = 1600 bar

Tables A.6, A.7 and A.8 are ΔP estimates for 1600 bar case, with 0 ms, 5 ms and 10 ms SOI delay, respectively.
Table A.6.
Pressure rise estimation results for case: 1600 bar, 1000 RPM, 0 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average $\Delta P$ [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>22.378</td>
<td>0.65777</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>22.553</td>
<td>0.68639</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>22.514</td>
<td>0.79266</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>22.598</td>
<td>0.75593</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>22.609</td>
<td>0.80277</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>22.609</td>
<td>0.80277</td>
</tr>
</tbody>
</table>

Table A.7.
Pressure rise estimation results for case: 1600 bar, 1000 RPM, 5 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average $\Delta P$ [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>21.416</td>
<td>0.54263</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>21.592</td>
<td>0.62019</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>21.649</td>
<td>0.78700</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>21.745</td>
<td>0.64174</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>21.773</td>
<td>0.62161</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>21.773</td>
<td>0.62161</td>
</tr>
</tbody>
</table>
Table A.8.
Pressure rise estimation results for case: 1600 bar, 1000 RPM, 10 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average ΔP [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>22.042</td>
<td>0.72281</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>22.024</td>
<td>0.90963</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>22.300</td>
<td>0.92828</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>22.302</td>
<td>0.90976</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>22.307</td>
<td>0.87507</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>22.314</td>
<td>0.87337</td>
</tr>
</tbody>
</table>

A.4 Rail Pressure = 1200 bar

Tables A.9 and A.10 are ΔP estimates for 1200 bar case, with 0 ms and 5 ms SOI delay, respectively.

Table A.9.
Pressure rise estimation results for case: 1200 bar, 1000 RPM, 0 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average ΔP [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>18.553</td>
<td>0.53484</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>18.743</td>
<td>0.61131</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>18.553</td>
<td>0.53484</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>18.661</td>
<td>0.65699</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>18.717</td>
<td>0.65525</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>18.717</td>
<td>0.65525</td>
</tr>
</tbody>
</table>
Table A.10.
Pressure rise estimation results for case: 1200 bar, 1000 RPM, 5 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average $\Delta P$ [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>18.871</td>
<td>0.56968</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>18.840</td>
<td>0.53513</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>18.890</td>
<td>0.44630</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>19.076</td>
<td>0.98239</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>19.126</td>
<td>0.96288</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>19.126</td>
<td>0.96288</td>
</tr>
</tbody>
</table>

A.5 Rail Pressure = 800 bar

Tables A.11, A.12 and A.13 are $\Delta P$ estimates for 800 bar case, with 0 ms, 5 ms and 10 ms SOI delay, respectively.

Table A.11.
Pressure rise estimation results for case: 800 bar, 1000 RPM, 0 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average $\Delta P$ [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>16.646</td>
<td>0.90617</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>17.109</td>
<td>0.90735</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>16.646</td>
<td>0.90617</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>16.992</td>
<td>0.80647</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>17.361</td>
<td>0.89429</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>17.302</td>
<td>0.91223</td>
</tr>
</tbody>
</table>
Table A.12.
Pressure rise estimation results for case: 800 bar, 1000 RPM, 5 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average ΔP [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
<td>48</td>
<td>18.255</td>
<td>0.45248</td>
</tr>
<tr>
<td>One</td>
<td>60</td>
<td>18.226</td>
<td>0.48787</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>19.723</td>
<td>1.7777</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>18.410</td>
<td>0.77729</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>17.584</td>
<td>0.76509</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>17.614</td>
<td>0.74289</td>
</tr>
</tbody>
</table>

Table A.13.
Pressure rise estimation results for case: 800 bar, 1000 RPM, 10 SOI.

<table>
<thead>
<tr>
<th>Window Type</th>
<th>Window Size [deg]</th>
<th>Average ΔP [bar]</th>
<th>Standard Deviation [bar]</th>
</tr>
</thead>
<tbody>
<tr>
<td>One</td>
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<td>17.265</td>
<td>0.67597</td>
</tr>
<tr>
<td>One</td>
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<td>0.67827</td>
</tr>
<tr>
<td>One</td>
<td>72</td>
<td>17.265</td>
<td>0.67597</td>
</tr>
<tr>
<td>Two</td>
<td>10</td>
<td>17.507</td>
<td>0.77023</td>
</tr>
<tr>
<td>Two</td>
<td>20</td>
<td>18.588</td>
<td>0.74084</td>
</tr>
<tr>
<td>Two</td>
<td>30</td>
<td>18.594</td>
<td>0.74084</td>
</tr>
</tbody>
</table>