Landfill Gas as a Fuel for IC Engine Operations

Guruprasath Narayanan
Western Michigan University

Follow this and additional works at: https://scholarworks.wmich.edu/masters_theses

Part of the Mechanical Engineering Commons

Recommended Citation
https://scholarworks.wmich.edu/masters_theses/1428

This Masters Thesis-Open Access is brought to you for free and open access
by the Graduate College at ScholarWorks at WMU. It has been accepted for
inclusion in Master's Theses by an authorized administrator of
ScholarWorks at WMU. For more information, please contact
maira.bundza@wmich.edu.
LANDFILL GAS AS A FUEL FOR IC ENGINE OPERATIONS

by

Guruprasath Narayanan

Thesis
Submitted to the
Faculty of The Graduate College
in partial fulfillment of the
requirements for the
Degree of Master of Science in Engineering (Mechanical)
Department of Mechanical and Aeronautical Engineering

Western Michigan University
Kalamazoo, Michigan
July 2006
LANDFILL GAS AS A FUEL FOR IC ENGINE OPERATIONS

Guruprasath Narayanan, M.S.
Western Michigan University, July 2006

An alternative fuel research facility was developed to study different alternative gaseous fuels like Landfill gases and Biogases and their mixtures with hydrogen as a fuel in a single cylinder spark ignition engine. Landfill gas was used as a fuel and the optimum process parameters such as compression ratio, spark timing and equivalence ratio were identified and reported.

Hydrogen enhanced combustion of landfill gas was studied and it was found that the addition of small amounts of hydrogen in the land fill gas has improved the combustion characteristics of the fuel and the performance of the spark ignition engine in general. Some interesting results while adding very small quantities of hydrogen in the landfill gas were found to improve the performance and thermal efficiency to significant levels which will be useful for the land fill gas to electricity (LFGTE) industry projects.

A computer model was also developed for the prediction of the combustion process of the landfill gas as a fuel for a spark ignition engine.
ACKNOWLEDGMENTS

I would like to express my thanks and appreciation to Dr. Bade Shrestha for providing me the opportunity to work on this research project. Without him I would have missed this great research experience. His guidance was very important for the completion of thesis and the graduate work.

I would like to express my sincere thanks to Glenn Hall who helped throughout the experimental setup design and fabrication phase. I would like to thank Pete Thanhauser for his able guidance and help in designing and implementing the data acquisition hardware and software. I would like to thank Dr. Richard Hathaway who helped me in Engine troubleshooting tasks. I would like to thank Jim Vandepolder who guided me with engine testing methods and safety concerns.

I would like to thank Dr. Norali Pernalete for helping me with the basics of Labview software and providing me the necessary information about data acquisition. I would like to express my heartfelt thanks to all my friends who have helped me in thousands of ways to reach this far and complete my thesis.

Guruprasath Narayanan
# TABLE OF CONTENTS

ACKNOWLEDGMENTS ........................................................................................................... ii
LIST OF TABLES ................................................................................................................... v
LIST OF FIGURES .................................................................................................................. vi

CHAPTER

1. INTRODUCTION .................................................................................................................. 1
   1.1 Importance of alternative fuels .................................................................................. 1
   1.2 Objective of study ....................................................................................................... 3

2. LITERATURE SURVEY ..................................................................................................... 5
   2.1 Landfill gas basics ..................................................................................................... 5
   2.2 Hydrogen enhanced combustion ............................................................................. 16
   2.3 Landfill gas studies in IC engines ........................................................................... 18

3. EXPERIMENTAL SETUP ................................................................................................. 21
   3.1 Laboratory setup ....................................................................................................... 21
   3.2 Setup validation ......................................................................................................... 44
   3.3 Experimental procedure ............................................................................................ 51
   3.4 Summary .................................................................................................................... 55

4. COMBUSTION MODELING ............................................................................................... 56
   4.1 Background ................................................................................................................ 56
   4.2 Fuel air residual gas properties ............................................................................... 57
   4.3 Equilibrium combustion products .......................................................................... 61
   4.4 Heat release formulations ......................................................................................... 65
Table of Contents -- continued

CHAPTER

4.5 Validation of the model and testing of the software ........................................... 74
4.6 Combustion duration analysis and estimation ...................................................... 79
4.7 Summary ................................................................................................................. 94

5. RESULTS AND DISCUSSIONS ................................................................................. 95
5.1 Methane experiments ............................................................................................ 95
5.2 Landfill gas experiments ....................................................................................... 100
5.3 Comparison of landfill gas to methane performances ........................................... 107
5.4 Summary ................................................................................................................. 120

6. THE EFFECTS OF ADDING SMALL AMOUNTS OF HYDROGEN TO LANDFILL GAS FUEL ................................................................. 121
6.1 Data analysis of landfill gas with hydrogen mixtures ............................................ 121
6.2 Economics of hydrogen usage as an additive for landfill gas ................................ 157
6.3 Summary ................................................................................................................. 161

7. CONCLUSIONS AND RECOMMENDATIONS ...................................................... 163

BIBLIOGRAPHY ............................................................................................................. 166

APPENDICES

A. Partial differential equations used in the equilibrium combustion products module ................................................................. 169
B. Matlab programs developed for combustion modeling ........................................ 179
C. Labview VIs developed for data acquisition and analysis ..................................... 204
D. Alternative fuels research lab .................................................................................. 210
### LIST OF TABLES

2.1 General composition of landfill gas ................................................................. 8

2.2 Heating value of fuels ..................................................................................... 17

3.1 CFR engine details ......................................................................................... 22

3.2 Flowmeters calibration range ........................................................................ 31

4.1 Low temperature combustion products of landfill gas (moles/mole of air) ................................................................. 60

4.2 Equilibrium reactions, equations and constants ............................................ 64

4.3 Percentage deviation of the estimated results from the experimental results ......................................................................................... 87

4.4 Percentage deviations of estimated combustion durations ....................... 91

4.5 Percentage deviations of predicted indicated power output .................... 91

5.1 Landfill gas compositions .............................................................................. 101
# LIST OF FIGURES

2.1 The schematic of a typical landfill gas collection system ....................... 6

3.1 The schematic of the modified air intake system .................................. 24

3.2 A schematic of the fuel control panel and the engine setup ..................... 26

3.3 Fuel control panel (rear view) .......................................................... 27

3.4 Fuel control panel and the CFR engine ............................................. 27

3.5 The schematic of the data acquisition design ....................................... 33

3.6 Front panel of flow data acquiring VI ............................................. 37

3.7 Front panel of pressure data recording VI ......................................... 38

3.8 Front panel of the composition equation solver VI ............................... 43

3.9 The variation of the compression pressure curve at 0.352 micrometer reading for the CFR engine during motoring at intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Setup validation for compression ratio setting ............................................. 45

3.10 The variation of pressure vs. crank angle in the CFR engine during motoring at compression ratio of 8.5, spark timing 20 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Setup validation for spark timing setting ............................................. 47

3.11 The variation of pressure vs. crank angle in the CFR engine during motoring at compression ratio of 8.5, spark timing 36 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Setup validation for spark timing setting ............................................. 47

3.12 The variation of dP/dθ vs. crank angle in the CFR engine during motoring at compression ratio of 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Error in Top Dead Center (TDC) detection ............................................. 49
3.13 The variation of $\frac{dP}{d\theta}$ vs. crank angle in the CFR engine during motoring at compression ratio of 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Corrected Top Dead Center (TDC) detection

3.14 The variation of pressure vs. equivalence ratio in the CFR engine with methane as fuel at compression ratio of 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Setup validation for the air and fuel flow system for leak proof

4.1 Wiebe function fitted for mass fraction burned from the experimental pressure data in the CFR engine with methane as fuel at compression ratio 8.5

4.2 Comparison of the mole fractions of equilibrium combustion species with the published data [5] at a pressure of 30 atm. and temperature 1750 K for Isooctane as fuel. (symbols are from [5] and lines are from program results)

4.3 Program results of equilibrium combustion products of methane showing the mole fractions of the species formed at a pressure of 30 atm. and temperature 1750 K

4.4 Program results of equilibrium combustion products of landfill gas showing the mole fractions of the species formed at a pressure of 30 atm. and temperature 2250 K

4.5 Program results of equilibrium combustion products of landfill gas with hydrogen showing the mole fractions of the species formed at a pressure of 30 atm. and temperature 2750 K

4.6 Comparison of experimental pressure and modeled pressure in the CFR engine with landfill gas as fuel at compression ratio 8.5, spark timing 25 BTDC, equivalence ratio 0.8, intake temperature 303 k, intake pressure 98 kPa and 600 rpm
4.7 Comparison of experimental pressure and modeled pressure in the CFR engine with landfill gas as fuel at compression ratio 8.5, spark timing 25 BTDC, equivalence ratio 1.0, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ........................................ 78

4.8 Typical pressure curve analysis to determine combustion duration from the experimental data .......................................................... 80

4.9 Typical variations of polytropic index with crank angle. The combustion duration $\Delta \theta_c$ is shown ......................................................... 81

4.10 Typical variations of experimentally derived combustion duration vs. equivalence ratio in the CFR engine with methane as fuel at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........ 82

4.11 A comparison between estimated combustion duration vs. equivalence ratio and experimental data for two spark timings in the CFR engine with methane as fuel at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.............. 88

4.12 A comparison between estimated ignition lag vs. equivalence ratio and experimental data for two spark timings in the CFR engine with methane as fuel at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........ 88

4.13 A comparison between estimated combustion duration vs. equivalence ratio and experimental data for landfill gas operation in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm............................................................... 92

4.14 A comparison between estimated combustion duration vs. equivalence ratio and experimental data for landfill gas with 20% hydrogen operation in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ............................................................... 92
4.15 A comparison between predicted indicated power output vs. equivalence ratio and experimental data for methane operation in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................................................................................. 93

4.16 A comparison between predicted indicated power output vs. equivalence ratio and experimental data for landfill gas operation and its 3% hydrogen mixture in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................................................................................. 93

5.1 The indicated power vs. equivalence ratio for various spark timings in the CFR engine with methane operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................................................................................. 96

5.2 The indicated power vs. equivalence ratio for various compression ratios in the CFR engine with methane operation at spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................................................................................. 97

5.3 The indicated power vs. spark timing for various equivalence ratios in the CFR engine with methane operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................................................................................. 97

5.4 The indicated power vs. spark timing for various equivalence ratio 1.0 in the CFR engine with methane operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................................................................................. 99

5.5 The indicated power vs. compression ratio for various equivalence ratios in the CFR engine with methane operation at spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................................................................................. 99
List of Figures—continued

5.6 The indicated power vs. equivalence ratio for various spark timings in the CFR engine with landfill gas operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 102

5.7 The indicated power vs. equivalence ratio for various spark timings in the CFR engine with landfill gas operation at compression ratio 12, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 103

5.8 The indicated power vs. equivalence ratio for various compression ratios in the CFR engine with landfill gas operation at spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 104

5.9 The indicated power vs. compression ratio for various equivalence ratios in the CFR engine with landfill gas operation at spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 105

5.10 The indicated power vs. spark timing BTDC for various equivalence ratios in the CFR engine with landfill gas operation at compression ratio 12, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 105

5.11 The comparison of indicated power vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratios 8.5 and 12, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 110

5.12 The comparison of indicated power vs. spark timing BTDC for equivalence ratio 1.0 in the CFR engine with methane and landfill gas operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 110

5.13 The comparison of indicated power vs. equivalence ratio in the CFR engine with various landfill gas compositions’ operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 112
5.14 The comparison of indicated power vs. equivalence ratio in the CFR engine with various landfill gas compositions’ operation at compression ratio 12, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 112

5.15 The comparison of indicated power vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm......................... 114

5.16 The comparison of COV of power vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........................................ 114

5.17 The comparison of exhaust gas temperature vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm............................................. 115

5.18 The comparison of indicated thermal efficiency vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm............................................. 115

5.19 The comparison of ignition lag vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................. 117

5.20 The comparison of COV of ignition lag vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm............................. 117

5.21 The comparison of combustion duration vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm............................................. 118
List of Figures—continued

5.22 The comparison of COV of combustion duration vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................. 118

5.23 The comparison of peak pressure vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........................................ 119

5.24 The comparison of COV of peak pressure vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........................................ 119

6.1 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........................................................................ 123

6.2 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........ 123

6.3 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........................................................................ 124

6.4 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........ 124
List of Figures—continued

6.5 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 125

6.6 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........ 125

6.7 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 126

6.8 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........ 126

6.9 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 127

6.10 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........ 127

6.11 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 128
List of Figures—continued

6.12 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm........ 128

6.13 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 133

6.14 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 133

6.15 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 134

6.16 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 134

6.17 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 135
List of Figures—continued

6.18 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 135

6.19 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 136

6.20 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm....................................................................................... 136

6.21 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm....................................................................................... 137

6.22 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm....................................................................................... 137

6.23 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm....................................................................................... 138
6.24 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.................................................................................................................. 138

6.25 The indicated power vs. spark timing BTDC for various equivalence ratios at a hydrogen percentage of 5% in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.................................................................................................................. 141

6.26 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.................................................................................................................. 142

6.27 The COV of average peak cylinder pressure vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm .................................................................................................................. 142

6.28 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.................................................................................................................. 143

6.29 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.................................................................................................................. 143
List of Figures—continued

6.30 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.......................................................... 144

6.31 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.......................................................... 144

6.32 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.......................................................... 145

6.33 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.......................................................... 145

6.34 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.......................................................... 146

6.35 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.......................................................... 146
List of Figures—continued

6.36 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 147

6.37 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................................. 147

6.38 The average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm...................................................................................... 150

6.39 The COV of average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm...................................................................................... 150

6.40 The average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm...................................................................................... 151

6.41 The COV of average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm...................................................................................... 151
List of Figures—continued

6.42 The average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ........................................................................................................... 152

6.43 The COV of average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ........................................................................................................... 152

6.44 The average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ........................................................................................................... 153

6.45 The COV of average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ........................................................................................................... 153

6.46 The average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ........................................................................................................... 154

6.47 The COV of average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ........................................................................................................... 154
6.48 The average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................. 155

6.49 The COV of average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................. 155

6.50 The pressure vs. crank angle showing the cyclic variations of using landfill gas as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm...................... 156

6.51 The pressure vs. crank angle showing the cyclic variations of using landfill gas with 3% hydrogen additive as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm............................................. 156

6.52 The increase in indicated power by adding 3% of hydrogen to the landfill gas as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm................................................. 160

6.53 The increase in indicated thermal efficiency by 3% of hydrogen to the landfill gas as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm..................................................... 160

6.54 The energy gain by adding 3% of hydrogen to the landfill gas as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm ................................................................. 161
1.1 Importance of alternative fuels

The evolution of transport technology has been dominated by an abundant supply of cheap petroleum fuels from abroad and domestic resources. The gradual increase in the price of the crude oil over a period in the international market has put an unbearable strain on the economy of most countries. The use of the non-fossil domestic energy resources to make an alternative or supplementary fuel that would lessen the need for imported petroleum is under investigation.

In fact most of the countries are heavily depended upon the oil producing countries for their energy production and transportation needs. In order to reduce this energy dependence most of the countries are involved in alternative energy research for their future energy needs.

Most of the conventional fuels used for energy production emit obnoxious gases like carbon monoxide and nitrous oxides that are very harmful for the environment. On the other hand most of the alternative energy technologies are environment friendly. This because the conventional fossil fuels mainly consists of hydrocarbons causing undesirable emissions. On contrary alternative energy technologies like solar energy and wind
energy don’t cause such emissions. For example hydrogen is considered to be the fuel of the future which when combusted produces only steam/or water and zero hydrocarbon emissions.

Solar energy, wind energy, biofuels, landfill gas and biogases are some of the most important alternative energy resources. Each alternative energy resource has its own advantages and disadvantages. Solar energy is freely available but the solar panels and equipment are costly and require a large space to capture enough sunlight. Wind energy is another successful alternative energy which contributes to the worlds energy needs. Landfill gases and biogases are low Btu gases which were, until recently, underutilized. In recent years the landfill gases and biogases have been utilized for the energy production.

The main portion of the landfill gas is comprised of Methane and carbon dioxide. If Methane is released directly into the atmosphere, it is a potent greenhouse gas. In fact its global warming potential is 21 times greater than that of carbon dioxide. Using it to generate energy encourages more efficient collection and thereby reduces emissions into the atmosphere. For this reason, energy recovery from landfill gas, where economically viable, is of considerable benefit to the environment. Thus the use of alternative fuel resources is beneficial to the environment as well. However because of some disadvantages involved in the landfill gas to electricity projects, landfill gases
are sometimes not considered to an attractive option. Thus it becomes necessary to solve the disadvantages involved by studying the landfill gases in a technological perspective and motivate the utilization of the landfill gas for the future energy needs and environmental protection. This research study has been done to study landfill gas as a fuel for spark ignition engine as a fuel to produce power in an effective way which can then be converted to electricity.

1.2 Objective of study

The goal of this work was to study the usage of low Btu gaseous fuels like landfill gases and biogases as an alternative fuel for the spark ignition engine. For this purpose an alternative fuels research facility had to be developed. Alternative energy resources like landfill gases, biogases vary widely in their compositions which makes it necessary to have a setup where the compositions can be varied easily and experimented. In addition it was designed to test alternative fuels with other diluents like nitrogen, carbon dioxide and hydrogen.

This was achieved by fabricating a fuel control panel and doing modifications in a CFR (Co-operative Fuel Research) gasoline run spark ignition engine. A data acquisition setup was designed and developed to get
the real time data from the engine. The LabVIEW software was used for data acquisition and analysis.

It was also desired to test landfill gas in the alternative fuels research setup and study various parameters like compression ratio, spark timing and equivalence ratio to develop the optimum parameters at which the engine can be run economically. Further it was designed to study the effects of adding small amounts of hydrogen with landfill gas as a combustion enhancer.

The data acquired were analyzed and discussed for performance, efficiency and combustion aspects of the landfill gas utilization in the internal combustion engine.

A computer model of the landfill gas combustion process was developed to diagnose and predict the engine performances using thermodynamic and heat transfer principles.
CHAPTER 2

LITERATURE SURVEY

2.1 Landfill gas basics

2.1.1 Landfill sites

Today’s landfills are not just dumping areas. They are carefully engineered fields constructed in different layers to contain the waste. Any landfill consists of five major parts. They are a liner system, a storage space for the waste, a leachate collection system, a gas collection system and a covering. The liner system prevents the waste contaminating the ground water and the leachate collection pipe collects the waste liquid generated inside the waste, separately so that it doesn’t mixes with ground water. The anaerobic decomposition of the organic materials in the waste produces a gas which is mainly composed of methane and carbon dioxide, very little amounts of nitrogen and trace amounts of oxygen and non-methane organic compounds (NMOCs). This gas produced in the landfills is called landfill gas. This gas gets collected by the gas collection system and is taken to the surface of the landfill for further use. The covering prevents the landfill gas getting vented to the atmosphere and it also prevents the atmospheric air from getting into the landfill waste.
Figure 2.1 The schematic of a typical landfill gas collection system.

Generally the life spans of landfills are from 20 to 30 years to accept the solid waste. One such landfill in Middlesex County, New Jersey [23] accepts 325 trucks equivalent to 2,200 tons of garbage a day, which amounts to 690,000 tons of garbage a year. It has a life expectancy of 15 to 20 years of operation (number of years it can accept solid waste), with an estimated
acceptance of 20 million cubic yards of solid waste. The wastes are compacted as they arrive at the landfill site and then dumped into the waste storage space. At the end of each day, the dumped waste is covered by a layer of soil 3 to 6 inches forming a cell. When the whole area is covered after several days the final capping is built which consists of a layer of plastic, a layer of soil, a protective blanket and finally a layer of sand. Above this shallow vegetation like grass is grown. The schematic of a typical landfill gas collection system is shown in Figure 2.1.

2.1.2 Landfill gas

Landfill gas is produced because of the anaerobic bacterial decomposition of the organic waste material contained in the landfill. Anaerobic refers to absence of oxygen and only such condition will produce methane. Depending upon the landfill site, type of waste, moisture content and the age of the waste, the composition of the gas vary widely. The general composition of the landfill gas [17] is shown in Table 2.1.
Table 2.1

General composition of landfill gas [17].

<table>
<thead>
<tr>
<th>No.</th>
<th>Gas</th>
<th>% in volume</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Methane</td>
<td>45-60%</td>
</tr>
<tr>
<td>2</td>
<td>Carbon dioxide</td>
<td>40-60%</td>
</tr>
<tr>
<td>3</td>
<td>Nitrogen</td>
<td>2-5%</td>
</tr>
<tr>
<td>4</td>
<td>Oxygen and other NMOCs</td>
<td>&lt; 1%</td>
</tr>
</tbody>
</table>

The gases are produced at four different phases of bacterial decomposition. The first phase is an aerobic decomposition phase, where the aerobic bacteria which lives only with oxygen consumes all the available oxygen and breaks down the longer molecules of complex carbohydrates, proteins and lipids that form organic wastes. This phase extends from few days to several months depending upon the amount of oxygen present along with the waste during disposal. After the first phase, the remaining bacterial decomposition happens in an anaerobic environment which enables the establishment of methane producing bacteria and at the end of all the phases, the landfill gas mainly consists of methane and carbon dioxide and other gases in trace amounts.
It takes almost 30-50 years for a landfill to stabilize producing landfill gas continuously. But shredding of the waste and adding water to the waste brings down the stabilization time from 30-50 years to 5-10 years [22]. One kg of waste generates around 0.18 m³ of carbon dioxide and 0.25 m³ of methane over several years of decomposition.

Landfill gas should be properly taken care of as it is hazardous to vent to atmosphere. Methane is a potent green house gas and leaking it to the atmosphere will cause global warming. It may also cause severe fire hazard. Such landfill fires in the past has injured and killed many people [1]. Because of these concerns and government regulations, landfill gas is flared in controlled conditions. Flaring is burning the landfill gas without any usage so that methane in the landfill gas is converted to carbon dioxide, reducing the dangers of fire hazard and reducing the green house gas effect. Rather than flaring the landfill gas without any usage, it is much better to utilize the landfill gas as a fuel.

### 2.1.3 Landfill gas utilization

Landfill gas can be used as a fuel because of the methane present in it. Since landfill gas consists of only around 50% methane and remaining 50% are incombustible gases like carbon dioxide, the heating value of the landfill gas is 16,785 to 20,495 kJ/m³ (450 to 550 Btu/ft³) as compared to that of
natural gas is 35,406 kJ/m$^3$ (950 Btu/ft$^3$). Because of the low heating value of the landfill gas, it is often overlooked and not utilized properly.

The landfill gas can be utilized with boilers, industrial heaters, space heating, to run engines and produce electricity. It can be processed to separate methane to get pipeline quality gas. There are more than 6000 landfills in US, out of which approximately 300 landfills have landfill gas to energy (LFGTE) projects implemented. One of the estimation shows that 0.5 m$^3$ of landfill gas is consumed to produce 1 kW-hr of electricity [1]. In that same project 42 million m$^3$ of landfill gas is consumed to generate 66 million kW-hr of electricity. Thus better usage of landfill gas not only produces energy but also avoids green house gas getting into atmosphere, reduces pollution in the coal fired power plants and conserves the fast depleting fossil fuel reserves of the world.

Direct landfill gas use refers to using the landfill gas directly as a fuel for boiler, space heating and cooling, industrial heating and co-firing. Very little gas clean up is required for these applications. Boiler and other systems already run by natural gas can be used with landfill gas with very little modifications in the equipment. Industrial use of landfill gas is advantageous because the requirement will be continuous for 24 hours a day. Whereas it has got limitations to be used as space heating because of the high piping costs and the load tend to be changing over time, both with day and with season.
These applications are very limited as the consumer has to be located nearer to the landfill site, less than 2 miles.

Landfill gas is also used with internal combustion engines and gas turbines to generate electricity. Both stoichiometric and lean burn internal combustion engines are used with landfill gas. Lean burn engines are used when the reduction of nitrogen oxides (NOx) and carbon monoxide (CO) is of concern. When there is a higher volume of landfill gas available for energy production then gas turbines are used instead of internal combustion engines. The general rule is that if the gas at the site is capable of producing 1-3 MW electricity then internal combustion engines are used and the site capable of producing more than 3 MW, a gas turbine is used. Because of the corrosive elements in the landfill gas, it is required to have spark ignition engines specifically designed for landfill gas these engines comprise little modifications from the usual natural gas engines. These kinds of engines are manufactured by major engine manufacturers like Waukesha, Caterpillar and Cooper-Superior. One of the statistics [1] compares that there are 61 landfill gas to electricity (LFGTE) projects with internal combustion engines as against 24 projects with turbine engines. But the amounts of electricity produced are compared as 49% from the internal combustion engines and 51% from the gas turbine engines [1]. It further claims that the parasitic
energy losses are 17% in the turbines as compared to only 7% in the internal combustion engines.

One other way of utilizing the landfill gas is the production of hydrogen. One of the popular ways of producing hydrogen is natural gas steam reforming process. The methane in the landfill gas can be used to produce hydrogen which can be either sold as hydrogen or it can be further used in fuel cells along with air or oxygen to produce electricity. Hydrogen combines with oxygen in a fuel cell to produce electricity and steam. Fuel cells are considered to be the best option for electricity generation as the efficiency is higher, lower maintenance cost because of no moving parts and lesser emissions than other conventional methods of producing electricity. Fuel cells are the best option for places where there are stringent emission norms.

2.1.4 Landfill gas to electricity demonstration project

A demonstration project [2] was conducted by the Energy research and development administration of New York state, to study the feasibility of producing electricity from the landfill gas along with the objectives to study the continuous supply of landfill gas from the site, efficiency of the engine with the minimum treatment of the landfill gas (water separation and particulate filtration), examine engine emissions and landfill
gas composition and to determine the corrosive effects if any because of the untreated gas.

The guidelines laid out [2] for landfill gas to electricity project (LFGTE) said that the major components required for implementing a LFGTE were:

1. Piping from the gas wells and into the engine generator.
2. A separator to remove water particles which otherwise will reduce the heating value of the gas and interfere with the combustion.
3. A compressor to create vacuum at the wells to extract the gas and to provide positive pressure for the gas to flow through all the systems until it reaches the engine.
4. A generator to convert landfill gas to electricity.
5. Gas engine generator building to protect all the equipment.

A gas analyzer was used to monitor the percentage of oxygen in the landfill gas, so that if the volume percentage reaches 2% then it was set to shut down the engine and the blower for safety reasons.

There were some interesting outcomes from this demonstration project which will motivate landfill gas to electricity projects. Very little modifications in the natural gas engine can help to run the engine more efficiently with landfill gas. The emissions from the engine using landfill gas were found to be better than that using natural gas. The engine head was
checked for corrosion after a long period of running it with landfill gas and it was concluded that the corrosion was very nominal. The thermal efficiency was found to be as low as 10% but increased to 20% after doing some modifications in the engine by advancing the spark timing.

2.1.5 Benefits and limitations of landfill gas utilization

Methane is a potent green house gas which traps the heat from the sun causing global warming. This damages the entire earth’s environment causing the polar glaciers to melt and sea levels are expected to rise flooding the shore cities. Methane is 21 times destructive than Carbon dioxide as a greenhouse gas. Though the methane is again converted to carbon dioxide by utilizing landfill gas, it is better than venting the methane directly to the atmosphere. Moreover by utilizing the landfill gas the pollution caused at the coal power plants are significantly reduced and the world’s oil reserves are conserved. It further reduces the dependence on foreign oil.

Landfill emissions also cause cancer and non-cancer health hazards [1] which will be avoided by utilizing the landfill gas properly. Each person in US is estimated to generate a ton of waste every year. If this waste is used to generate electricity through landfill gas then it will be a renewable source of energy.
In spite of more than 6000 landfills in US only less than 300 sites have implemented landfill gas to energy projects. More than 30 projects were closed during 1980s because of the competitive market conditions. Fluctuations in demand and the gas production cause some limitations in implementing landfill gas to energy projects. Economics of producing the landfill gas plays a vital role in determining the success of the projects. It becomes necessary for the landfill gas to energy projects to be cost competitive with the available energy projects such as natural gas, fossil fuels etc.
2.2 Hydrogen enhanced combustion

Hydrogen is believed to be the fuel of the future. Hydrogen has got some extra ordinary properties than any other fuel which makes it a popular fuel. Some of them are:

- Hydrogen has got a very high laminar flame speed of 1.6 m/s as compared to that of methane which is 0.38 m/s.
- Hydrogen is the best known clean burning fuel. On combustion hydrogen combines with oxygen to produce water. So the exhaust will be just steam and there is no carbon or nitrous compound emissions.
- Hydrogen has got a very high flammability limits. It burns in air from 4% to 75% on volume basis as compared to that of methane it is only 5% to 15% on volume basis.
- Hydrogen is highly ignitable.

These properties make hydrogen a special fuel. In spite of these great benefits hydrogen has got some disadvantages. They are:

- Though hydrogen has the highest heating value on mass basis, on volume basis its heating value is very low compared to other fuels. (See Table 2.2). This makes it difficult for hydrogen to be used in many applications like transportation.
• Hydrogen is not freely available and the cost producing hydrogen is very high. In order to extract hydrogen from water, a significant amount of electrical energy has to be spent.

Table 2.2

Heating value of fuels [24].

<table>
<thead>
<tr>
<th>Fuel</th>
<th>Mass basis</th>
<th>Volume basis</th>
</tr>
</thead>
<tbody>
<tr>
<td>Hydrogen</td>
<td>119.3 kJ/g (51,500 Btu/lb)</td>
<td>10,050 kJ/m³ (270 Btu/ft³)</td>
</tr>
<tr>
<td>Methane</td>
<td>50.02 kJ/g (21,500 Btu/lb)</td>
<td>32,560 kJ/m³ (875 Btu/ft³)</td>
</tr>
<tr>
<td>Gasoline</td>
<td>44.5 kJ/g (19,000 Btu/lb)</td>
<td>31,150,000kJ/m³ (836,000 Btu/ft³)</td>
</tr>
</tbody>
</table>

Thus hydrogen, despite its superior fuel qualities it is not being utilized fully because of the disadvantages involved. Another way of utilizing the benefits of hydrogen is instead of using hydrogen as 100% fuel it can be used as an additive for other fuels as combustion enhancer. This kind of hydrogen enhanced combustion is being studied with fuels like methane and gasoline [29].

Landfill gas, because of the presence of incombustible diluents like carbon dioxide and nitrogen in it, may not be performing to the level of 100% methane. If hydrogen is used as a combustion enhancer for the landfill gas, it
may improve the efficiency and performance of the landfill gas combustion. Any improvement in efficiency and performance will be extremely beneficial for the better utilization of the landfill gas. This will further encourage the landfill gas to energy projects, which helps the betterment of environment, reduces energy dependence and generate revenue.

2.3 Landfill gas studies in IC engines

As the fossil fuels are fast depleting there is a great awareness among the researchers to tryout alternative fuels in the internal combustion engines. Biogases and landfill gases because of their easy availability and renewable supply are considered for research with the internal combustion engine. Particularly countries like India and China are showing great interest for this kind of biogas energy as they have a higher rural environment capable of producing a large amount of biogas.

Most of the research studies have simulated the biogases and landfill gases compositions to study them in engines. One of the study conducted by Huang and Crookes [25] show that increasing the compression ratio will substantially increase the performance of the biogas fuelled engine. It was found that compression ratio of 13 was giving the maximum power in a variable compression ratio Ricardo E6 single cylinder engine. It was found
that at higher compression ratios beyond 13, engine was knocking which will deteriorate the power and durability of the engine. It was found that the increasing percentage of carbon dioxide in the biogas reduced the NO\textsubscript{x} emissions significantly which was mainly attributed to the reduced combustion temperatures. As the carbon dioxide percentage increased in the fuel though NO\textsubscript{x} levels reduced the unburned HC levels increased. A 3% reduction in brake power and brake thermal efficiency was observed as the carbon dioxide increased to 40% in the fuel. Significant decrease in the peak cylinder pressure was also observed as the carbon dioxide percentage increased. It was concluded that though a compromise can be achieved by optimum operating conditions for improved performance and reduced emissions, it is required to have lean mixture, fast burning engine design technology for better utilization of biogas in internal combustion engines.

Methane, carbon dioxide and nitrogen mixtures were studied as fuels for spark ignition engines by Karim and Wierzba [26]. The adverse effects caused by the diluents like carbon dioxide and nitrogen on the main fuel methane were studied and some guidelines to alleviate those adversities were given. It was observed that the presence of diluents reduces the flame temperatures significantly. The apparent flame speed of the mixture was drastically reduced with the increased presence of diluents. It was observed that the slowing down of the combustion reaction and the energy release will
depend upon the diluent involved, the temperature level and equivalence ratio. It was found that the type of diluent involved made a big difference for the combustion characteristics observed. The carbon dioxide was having a greater deteriorating effect than the nitrogen. It was suggested that preheating of the mixtures would permit higher concentrations of diluents in the fuel. There was a shift in the maximum indicated power towards the lean side of the stoichiometric operation which was attributed to the slow burning of the mixtures. As the diluents increased the spark advance had to be increased to produce maximum power. It was suggested that the spark timing was the easiest parameter that can be adjusted to run a spark ignition engine efficiently with fuels like biogases and landfill gases.
CHAPTER 3

EXPERIMENTAL SETUP

3.1 Laboratory setup

3.1.1 CFR engine

The engine used in this research was a CFR (co-operative fuel research) single cylinder spark ignition engine. This kind of engine is usually used for fuel research where the octane number of the fuel is to be determined.

This kind of engine is coupled with a synchronous motor. This motor acts as a starter motor to start the engine and power the engine until the engine is fuelled. This way of running the engine through synchronous motor without fuelling is called “Motoring”. When the engine is fuelled and starts producing power, this power is absorbed by the synchronous motor. The speed of the engine is constant at 600 rpm but during motoring its speed is slightly less which is around 590 rpm.

The most important features of this type of engine are:

- A single cylinder.
- Variable compression ratio.
- Variable spark timing.
- Constant speed.
It is convenient to work with a single cylinder engine rather than a four cylinder or a six cylinder engine which will make the entire setup more complicated. The head of the engine is a movable block with respect to the cylinder block which facilitates to vary the compression ratio from 4:1 to 16:1. This feature is not possible in other non-research engine which makes it a peculiar engine to do fuel research. The spark timing can also be varied over a wide range from 40° BTDC (before top dead center) to 40° ATDC (after top dead center). The engine geometric details are given in Table 3.1. Thus the fuel under research can be tested at various compression ratios and various spark timings.

Table 3.1

CFR engine details.

<table>
<thead>
<tr>
<th>Make</th>
<th>Waukesha</th>
</tr>
</thead>
<tbody>
<tr>
<td>Compression ratio</td>
<td>4:1 to 16:1</td>
</tr>
<tr>
<td>Cylinder bore</td>
<td>0.08255 m (3.25”)</td>
</tr>
<tr>
<td>Stroke</td>
<td>0.1143 m (4.5”)</td>
</tr>
<tr>
<td>Connecting rod length</td>
<td>0.254 m (10”)</td>
</tr>
<tr>
<td>Displacement volume</td>
<td>0.000611m³ (37.33 in³)</td>
</tr>
</tbody>
</table>
The engine is fitted with an air intake system and a carburetor for fuel intake. The air intake is fitted with a filter, air heater and an intake manifold. The exhaust manifold is cooled by a running water system. The combustion chamber has the provision for fitting a pressure sensor to measure the in-cylinder pressure of the combustion chamber. The engine also has an extended crankshaft on which a rotary encoder can be fitted in to measure the crank angle.

**Air Intake Modifications**

The CFR engine usually comes with air intake system which consists of an air filter, air heater and an intake manifold. Since it was required to use electronic flow meters to measure the flow rate of air and other gases so that the flow data can be taken to the computer on real time, the design of the intake manifold was modified. The available air filter, air heater and intake manifold were removed and was replaced such that the atmospheric air enters into the air filter, passes through the flow meter, a 20 gallon air tank and a PVC pipe manifold where it gets mixed up with the gaseous fuels. This premixed air and fuel mixture enters into the engine for combustion. Since the airflow to the engine is an intermittent process, it becomes very important to reduce the pulsations in the air flow which is possible by having a large air tank as a buffer. The size of the buffer tank should be significantly larger
compared to the displacement volume of the cylinder so that the air flow through the flow meter is almost continuous. The Figure 3.1 shows the schematic of the modified air intake system.

![Schematic of the modified air intake system](image)

**Figure 3.1** The schematic of the modified air intake system.

**Fuel control panel**

The carburetor which was available for metering the liquid fuels was removed and a new fuel control panel was designed and fabricated. The important aspect of the fuel control panel is to flow different gases with varying proportions from the corresponding gas cylinders like Methane, Carbon dioxide, Nitrogen and Hydrogen.
The schematic of the fuel control panel is shown in Figure 3.2. Basically the fuel control panel consists of four different layers, each one consisting of a fuel filter, gas flow meter, manual flow controllers and a pressure gauge. There are two different manual flow controllers, a coarse flow controller to control the flow in a coarse manner and a fine flow controller to adjust the flow in a precise manner. All these four gases flow from the pressure regulators of the gas cylinders and then flow through their corresponding set of flow meters and flow controllers and finally reach a gas mixing chamber where all the four gases can be mixed homogeneously. The gases have to be mixed well for proper combustion. This homogeneous gaseous mixture is then taken to the air and fuel mix manifold where the gaseous mixture gets mixed up with the air intake and enters into the engine. A solenoid operated safety valve was placed in the fuel line which shuts off the fuel supply if the CFR engine stops for some reasons. This was done as a safety initiative to avoid the fuel getting supplied when there is no sparking in the engine which may cause a fire hazard.

The composition of Landfill gas is not the same everywhere. It varies widely depending upon the landfill site and the kind of waste dumped. This kind of fuel control panel allows changing the gaseous mixture composition, which is essential in this kind of fuel research where the fuel compositions
have to be varied widely. Moreover when hydrogen is used as a diluent its percentage in the fuel has to be varied over a wide range.

Moreover this kind of fuel control panel facilitates to vary the fuel quantity with respect to the air intake so that the equivalence ratio can be easily varied. Thus the fuel can be tested under different equivalence ratios say from 0.6 to 1.2. The Figures 3.3 and 3.4 show the photographs of the fuel control panel.

Figure 3.2 A schematic of the fuel control panel and the engine setup.
Figure 3.3 Fuel control panel (rear view).

Figure 3.4 Fuel control panel and the CFR engine.
3.1.2 Data acquisition hardware

Pressure sensor

The type of pressure sensor used was 7061 B water-cooled ThermoCOMP pressure sensors from Kistler. These kinds of piezoelectric pressure sensors play an important role in engine pressure measurement. The working principle of the pressure sensor was that the pressure acts on the diaphragm, which is converted into a proportional force. This force is taken to a quartz packet and gets converted to an electrostatic charge under load. These charges are further fed to a charge amplifier which converts the negative charges from the pressure sensor to a positive voltage. The thermal shock in the pressure measurements is reduced by the double walled diaphragm and the load cycle drift is reduced by the inbuilt water cooling system. Thus the accuracy of the pressure measurement was increased. The pressure sensor was capable of measuring a wide range of pressure from 0 to 250 bars and the sensitivity was -80pC/bar (Pico-coulombs per bar).

The thread size in the pressure sensor was different from the thread size in the CFR engine. So a stainless steel adapter was machined and fitted in the engine according to the guidelines provided by the pressure sensor maker. Moreover the cooling water system of the CFR engine was used to supply cooling water for the pressure sensor. The connector cable which
carries the charges was connected to the PCB charge amplifier. This connector cable should be of high insulation low-noise type; otherwise external noises affect the accuracy of the pressure measurement.

**PCB charge amplifier**

The type of charge amplifier used was MOD 462A PCB charge amplifier. The PCB charge amplifier is basically a device which converts the negative charges received from the pressure sensor into positive voltages. This is because in any data acquisition setup it is the voltage which is easily measured. This charge amplifier was fitted with a 180 kHz low pass filter which means it will pass the signals that are lower than 180 kHz frequency and the signals above this frequency will be cutoff. This is important if we want to measure the knock in the combustion chamber as knocking are high frequency signals. Some of the knock frequencies are in the range of 40 kHz. If this factor of low pass filter in the charge amplifier is not taken care then we may get the pressure signals but may not be able to get the high frequency knock signals as they may get cutoff by the low pass filter.

The pressure sensor’s sensitivity which was -80pC/bar has to be set in this charge amplifier. Moreover the magnification factor was set to 5000. The voltage signals from the charge amplifier are taken to the terminal block of the data acquisition card.
Rotary encoder

The type of rotary encoder used was model HS35 incremental optical encoder from BEI industrial encoders. The rotary encoder is the device which is fitted in the crankshaft to measure the crank angle position. This measurement was very important as the in-cylinder pressure was measured along the crank angle and the signals from the rotary encoder acts as the external clock signal for the pressure measurement. These encoders produce certain number of pulses for each shaft revolution. This was achieved by a combination of an optical disk consisting of alternating transparent and opaque slits, a light emitting diode and photo diodes as receivers. This encoder provides three different signals. They are the pulse signal (A), another pulse signal with a phase shift of 90° (B) and a zero signal (Z) for each revolution. The phase shift signal can be used to find the direction of rotation and the zero signal provides a reference point for each revolution.

This encoder was capable of producing 4096 pulses per revolution of shaft which means the crank angle resolution will be 0.0879°. Thus the encoder will be producing 4096 pulses per revolution of crankshaft that makes 8192 pulses per engine cycle (one 4-stroke engine cycle=2 revolutions). Since these counter pulses act as an external clock for the pressure measurement, 8192 pressure readings will be got for one engine cycle. The
bore size on the encoder was 2.2225 cm (0.875”) which makes it easier to directly fit the encoder in the crankshaft.

Mass flowmeters

The mass flow meter used for air and methane was of the type 40241 TSI flow meter. The underlying principle of any thermal mass flow meter is that the heat absorption rate by a flow stream is directly proportional to its mass.

<table>
<thead>
<tr>
<th>Gas</th>
<th>Flow range calibrated for (SLPM)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air</td>
<td>300</td>
</tr>
<tr>
<td>Nitrogen</td>
<td>77</td>
</tr>
<tr>
<td>Hydrogen</td>
<td>100</td>
</tr>
<tr>
<td>Carbon dioxide</td>
<td>35</td>
</tr>
<tr>
<td>Methane</td>
<td>30</td>
</tr>
</tbody>
</table>

flow. These flow meters have an accuracy of ± 2% of reading and a 4 millisecond flow response which is highly desired in this kind of engine flows where the air flow is intermittent.
For other gases like nitrogen, carbon dioxide and hydrogen, the type of mass flow meter used was model GR111 from Fathom technologies. These mass flow meters use a capillary type thermal technology to directly measure the mass flow rate of the gas. The accuracy of this type of mass flow meter is ±1% of full scale. The response time was around 1 second. The flow of gases was continuous, as the gases pass through a long path to reach the air and fuel mix panel. Whereas in the case of air flow it was not so, as the air doesn't flow through a long path before reaching the engine and the intermittent flow was felt by the air flow meter. So the air flow meter should have a very low response time to measure the intermittent air flow. The air and other gas flow meters are calibrated for a particular flow range as required by the experiments and are shown in the Table 3.2.

In place of flow meters, electronic flow controllers can be used which not only measures the flow rate but also controls it. For cost effectiveness flow meters were used instead of flow controllers. But flow controllers are extremely advantageous while doing this kind of experiments where the flow rate of the gases have to be varied often to do experiments in different compositions and in different equivalence ratios. Flow controllers will reduce the experimental time drastically and improve the efficiency of the experimenter.
All these flow meters had to be calibrated in proper time intervals as recommended by the manufacturer, in order to maintain the accuracy of the experiments. All these flow meters had an analog output voltage proportional to the flow rate and this analog output was taken to the terminal block of the data acquisition card.

Thermocouples

The air and fuel mixture temperature was measured at the intake manifold and the exhaust gas temperature was measured in the exhaust manifold by thermocouples. The J type Omega thermocouples are used here. The basic principle of thermocouple is that a voltage is produced across a junction of dissimilar metals which is a function of temperature. These
thermocouples are quick-disconnect type which makes it easier to disconnect the probe from the rest of the connector. These voltage signals from the thermocouple probes are fed to a temperature display panel which displays the intake and exhaust temperatures.

**Terminal block**

The terminal block is where all the output signals from the sensors are connected to. From the terminal block these signals are taken to the data acquisition card. The type of terminal block used was SCB 68 from National instruments. This has got 68 connections in it and each connection is designated as analog signal, digital signal, and power source or counter input/output.

**Data acquisition card**

The type of data acquisition card used was PCI-MIO-16E-4 with 12 bit resolution from National instruments. Data acquisition card is basically an ADC (analog to digital converter) device which converts the analog signal from the sensors into computer recognizable digital signals. The sensitivity of a measured parameter mainly depends on the resolution of the ADC device used. For example the sensitivity of pressure measurement can be found by
following method which is 0.061 bar for the calibrated pressure range of 0-250 bar.

$$S := \frac{\Delta P}{2^r}$$  \hspace{1cm} (3.1)

where $S$ is the pressure sensitivity, $\Delta P$ is the calibrated pressure range and $r$ is the resolution of the ADC.

Data acquisition computer

The computer used for the real time engine data acquisition was Intel Pentium III, with Windows XP operating system with 256 MB RAM and 10 GB hard disk space. This computer was loaded with LabVIEW 7.1 and Measurement and Automation Explorer (MAX) for data acquisition and analysis.
3.1.3 Data acquisition software

LabVIEW from National instruments was the software used for acquiring the data and the same is used for analyzing and displaying the data. The LabVIEW programs are called as VIs (virtual instrument) instead of programs, as they are more likely to be a combination of instruments in a graphical programming environment. Every software is uniquely developed for a particular application. LabVIEW is one such software specially developed for data acquisition and analysis. Basically any LabVIEW VI consists of two basic parts namely the front panel and the block diagram. The front panel is the area where the user interacts with the VI such as giving inputs and looking at the outputs in numeric or graphical forms. The block diagram is the area where the programmer programs the VI to make it function as desired. The main VIs developed for this data acquisition application are:

1. Flow data acquiring VI.
2. Pressure data recording VI.

The flow data acquiring VI acquires and displays the flow of all gases and air flow. With this VI we can use different proportions of gases as fuel to simulate different composition of landfill gas and biogases. Moreover it displays the equivalence ratio of the air fuel mixture which is a very important parameter of study in these experiments. In addition to the gas
Figure 3.6 Front panel of flow data acquiring VI.

composition variation, the hydrogen percentage in the fuel is also shown by this VI. Thus the percentage of hydrogen in the fuel-hydrogen mixture can be studied as one of the parameter. All these gas compositions, equivalence ratio and hydrogen percentages are displayed on real time which makes it easier to bring the fuel-air-hydrogen mixture to required values. This is perfectly achieved by a composition equations solver as a subVI in this VI. At the end of the VI all the flow rates of all gases and air, and the equivalence ratio are exported to a LVM data file.
Figure 3.7 Front panel of pressure data recording VI.

While the flow VI helps to set the required flow of gases, the pressure recording VI is the one which collects the pressure data from the engine run. The primary function of this VI is to collect the in-cylinder pressure data of the engine against all crank angle positions. Before starting the VI to run, the atmospheric pressure in inches of mercury, compression ratio, spark timing, intake temperature and exhaust temperature are all entered in the VI. Once the VI is started to run, it records the in-cylinder pressure from the combustion chamber for 100 consecutive engine cycles. This is important for studying the cyclic variations in the engine. After recording the pressure
readings for 100 cycles the average pressure reading is calculated from them. Since the encoder produces 4096 pulses per revolution and 8192 pulses per engine cycle, the number of pressure readings recorded for one cycle is 8192.

The VI further uses the average pressure data to calculate the work done per cycle. This is further used to calculate the power generated per cycle. In addition to this the VI displays the PV diagram and the pressure vs. crank angle diagrams. Moreover this VI finally creates a LVM data file containing all the information of that particular run. This data file contains work, power, all input parameter details including the flow data and equivalence ratio details, average pressure data, motored pressure data, 100 cycle pressure data and combustion duration analysis details. The front panel of the flow data acquiring VI and the pressure data recording VI are shown in Figures 3.6 and 3.7 respectively. The motored pressure data was recorded using a similar VI as the pressure data recording VI whenever the compression ratio was changed.

**Composition equation solver**

The first thing to be achieved in each engine run is that the required landfill gas composition and the required fuel air ratio. This is achieved by the proper proportions of methane, carbon dioxide, nitrogen and hydrogen and the air flow. However achieving this is a difficult task, because
equivalence ratio and the fuel composition are inter related features where in adjusting the fuel composition also changes the equivalence ratio. This can not be achieved by trial and error method of adjusting the flows of the individual gases. A mathematical approach of finding the required flows of individual gases from the known variables was adopted.

When hydrogen is added to the landfill gas, both methane and hydrogen are combustibles and the way the equivalence ratio is calculated changes. Here it becomes a variable stoichiometric case as the proportions between the methane and hydrogen keeps continuously changing for different experiments.

Let us assume that h is the air flow rate, x is the methane flow rate, y is the carbon dioxide flow rate, z is the nitrogen flow rate, and p is the hydrogen flow rate.

\[
\frac{x+p}{h} \cdot \varphi := \Phi. 
\]

is the desired equivalence ratio and the stoichiometric air flow \( S_a \) is calculated from the variable stoichiometry equation given by,

\[
S_a = 9.52x + 2.38p 
\]
where 9.52 is the stoichiometric volume of air required for one volume of methane and 2.38 is the stoichiometric volume of air required for one volume of hydrogen.

In order to get the required landfill gas composition, the following equations were developed.

\[
\frac{x}{x + y + z} := mp. \tag{3.4}
\]

\[
\frac{y}{x + y + z} := co2p. \tag{3.5}
\]

\[
\frac{z}{x + y + z} := np. \tag{3.6}
\]

Where mp is the required methane percentage, co2p is the required carbon dioxide percentage and np is the required nitrogen percentage in the landfill gas composition. In order to get the required percentage of hydrogen in the fuel part the following equation was developed. Say for example 10% of hydrogen means the remaining 90% is landfill gas, within which the methane, carbon dioxide and nitrogen proportions are maintained as required.

\[
\frac{p}{x + y + z + p} := h2p. \tag{3.7}
\]

where h2p is the required hydrogen percentage in the landfill gas with hydrogen mixture. Further the sum of all the flow rates should be equal to the total flow rate \( T_{LPM} \), acquired during the engine run. This total flow rate
varies with different parameters and is not a constant. So it becomes necessary to get this data real time in the Labview VI during the engine run.

\[ h + x + y + z + p = T_{LPM}. \]  

(3.8)

Solving these six linear equations for \( x, y, z, p \) unknowns makes it possible to set the desired flowrate from the fuel control panel. The air flow rate cannot be controlled but can be verified to reach the value solved mathematically.

As an example if 53% methane, 42% carbon dioxide and 5% nitrogen is the desired landfill gas composition and say 10% hydrogen is desired in the fuel part, then using the above equations the linear equations formed will be,

\[
\begin{align*}
  h - 9.52x + 0y + 0z - 2.38p &= 0 \\
  0 + (0.53-1)x + 0.53y + 0.53z + 0p &= 0 \\
  0 + 0.42x + (0.42-1)y + 0.42z + 0p &= 0 \\
  0 + 0.05x + 0.05y + (0.05-1)z + 0p &= 0 \\
  0 + 0.1x + 0.1y +0.1z + (0.1-1)p &= 0 \\
  h + x + y + z + p &= T_{LPM}
\end{align*}
\]

(3.9)

Bringing the gas flow rates to the solved values automatically achieves the required equivalence ratio, the landfill gas composition and the percentage of hydrogen in the fuel part. This feature is incorporated in the flow data VI, through a sub VI called composition equation solver VI. The
built in linear equations solver is made use of in this VI to solve the equations in real time as the total LPM keeps varying significantly. If electronic flow controllers are used in place of electronic flowmeters and manual flow controllers, this part of achieving the required flow can be made much more easily, quickly and automatically rather than any human intervention. The front panel of the composition equation solver VI is shown in the Figure 3.8.

![Figure 3.8 Front panel of the composition equation solver VI.](image)

Flowing the reqd. LFG composition by trial and error is tedious and cannot be achieved at all. The logical way is to solve the linear equations got out of the known and reqd. composition values.

\[
\begin{align*}
\text{Flowing the reqd. LFG composition by trial and error is tedious and cannot be achieved at all. The logical way is to solve the linear equations got out of the known and reqd. composition values.}
\end{align*}
\]
3.2 Setup validation

Once the engine setup was ready with fuel control panel, air intake system and data acquisition modules, it was validated for various parameters. The setup validation was the most time consuming process in the entire work. As this part determines the accuracy of the results and the correctness of the experiments, it was given the utmost care. The setup validation was done for the following parameters.

1. To ensure the correct relationship of the micrometer reading with respect to the compression ratio.

2. To ensure the spark timing set is correct.

3. To ensure the top dead center detected by the rotary encoder is correct.

As the CFR engine has a variable compression ratio because of the adjustable clearance volume, it is necessary to ensure the correct relationship of the micrometer reading with respect to the compression ratio. The procedure given in the ASTM manual for “Rating motor fuels” was carried out for this checking. The 1.587 cm (⅝ in.) plug gage was connected in the place of the pressure sensor on top of the combustion chamber. This gage is equipped with a check valve assembly fitted with TFE-fluorocarbon wafer seals. With the engine at normal operating temperature, the cylinder height was adjusted to get the compression pressure shown in the manual. For
Figure 3.9 The variation of the compression pressure curve at 0.352 micrometer reading for the CFR engine during motoring at intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Setup validation for compression ratio setting.

76.14 cm (29.28 in.) of mercury the compression pressure is 1465 kPa (Absolute pressure). Then without changing the cylinder height the micrometer reading was adjusted to 0.352. This ensures the correct relationship between the micrometer reading and the compression ratio. The compression pressure was further verified by the peak pressure reading from the pressure sensor shown in Figure 3.9.

The spark advance is set by looking at a flashing neon light on the spark timing scale of the CFR engine. Because of the difficulties with this method, a computerized tachometer/advance timing light from Snap-On Tools Corporation was used to set the spark timing. The type used was model number MT1261A. In order to verify the correctness of this method to set the
spark timing, the pressure reading from the pressure sensor was scrutinized to see the voltage spikes during the sparking and this was with perfect match with the set spark timing. This is done for two different spark timings, 20° BTDC and 36° BTDC shown in Figure 3.10 and Figure 3.11.

The data acquisition starts acquiring the pressure signals after a trigger point given by the rotary encoder. This trigger point is supposed to be the exact top dead center of the piston motion. If this is not correctly set then all the analysis done from the pressure reading will be erroneous. The maximum pressure for the motoring has to occur at the top dead center. To verify this, the differentiation of the pressure curve with respect to the crank angle was plotted shown in Figure 3.12. The differentiation at the top dead center has to be zero. This showed that the signal top dead center was offset from the actual top dead center. This offset was corrected by offsetting the crank angle values in the pressure recording VI. The corrected top dead center is shown in Figure 3.13.

When the equivalence ratio is varied from the lean to rich limits, the maximum power occurs at near stoichiometric equivalence ratio. If this is not the case then the air intake or the fuel control panel is most likely to have leaks in it. If the highest power is on the far leaner side of the stoichiometric
Figure 3.10 The variation of pressure vs. crank angle in the CFR engine during motoring at compression ratio of 8.5, spark timing 20 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Setup validation for spark timing setting.

Figure 3.11 The variation of pressure vs. crank angle in the CFR engine during motoring at compression ratio of 8.5, spark timing 36 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Setup validation for spark timing setting.
then possibly the leak is in the fuel control panel. If the highest power is on the far richer side of stoichiometric then possibly the leak is in the air intake system. To detect leaks soap water was applied on the fuel and air lines of the intake system and the fuel control panel. The leaks have to be carefully detected and arrested, so that the flow of gases and the air intake to the engine are properly measured and taken into account of the equivalence ratio calculation.

Other than leaks the most common reason for the shift in maximum power can be the flow meters calibration error. Whenever the flow meters were doubted, a rotameter was connected in line with the doubtful flow meters and checked for its correctness. If the variation from the electronic flow meter and the rotameter is significantly different then the flow meter was sent for recalibration. All these flow meters were recommended to be recalibrated for once in a year by the respective flow meter manufacturer. After eliminating all the leaks in the system and recalibration of flow meter, the maximum power occurred at near stoichiometric equivalence ratio shown in Figure 3.14.
Figure 3.12 The variation of $dP/d\theta$ vs. crank angle in the CFR engine during motoring at compression ratio of 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Error in Top Dead Center (TDC) detection.

Figure 3.13 The variation of $dP/d\theta$ vs. crank angle in the CFR engine during motoring at compression ratio of 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Corrected Top Dead Center (TDC) detection.
Figure 3.14 The variation of pressure vs. equivalence ratio in the CFR engine with methane as fuel at compression ratio of 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm. Setup validation for the air and fuel flow system for leak proof.

The volumetric efficiency of the engine had to be checked prior to conducting the experiments. Because of the air intake system design and the restrictions in the air passage causes the volumetric efficiency to decrease. The same reasons for the shift in the highest power from the stoichiometric can be causing this decrease in volumetric efficiency. So it is necessary to check the leaks in the air intake system and also the correctness of air flowmeter calibration. Prior to doing these checks the erroneous volumetric efficiency was only 65%. After thorough leak check of the air intake system and recalibration of air flowmeter, the volumetric efficiency increased to 90%. Even the pulsated flow of air intake can cause a problem in measuring the air
flow accurately. Care must be taken in air flowmeter selection for the response time to be small enough to handle the pulsating flow. Placement of a 20 gallon air tank also helped to substantially reduce the pulsations in the air flow.

3.3 Experimental procedure

After the setup was validated the actual experiments were conducted with three different kinds of fuels. Since the major portion of landfill gas consists of methane, experiments were conducted with 100% methane which will serve as a baseline for landfill gas. All the landfill gas experiments were conducted with the average landfill gas composition (explained in section 5.2) with 53% methane, 42% carbon dioxide and 5% nitrogen. A minimum number of experiments were conducted in the extreme landfill gas compositions. Finally hydrogen was added in small amounts with landfill gas. Instead of using a direct landfill gas acquired form landfill sites, the landfill gas composition was simulated by having methane, carbon dioxide and nitrogen gas cylinders separately and mixing them in proper proportions in the gas mixing chamber in the fuel control panel. This homogeneous gaseous mixture which represents the average landfill gas composition was tested in the engine.

The parameters which were varied for doing these experiments were:
• Compression ratio.
• Spark timing.
• Equivalence ratio.
• Hydrogen percentage in fuel.

Since the CFR engine is a variable compression ratio engine, the compression ratio can be varied from 4 to 16. In these experiments in order to get data for a wide range of compression ratios, it was varied from 7 to 14. Most of the experiments were done at compression ratio 8.5 and for higher compression ratios 12 and 14 were used. The pressure data recording VI was developed with a micrometer setting calculation based on the compression ratio desired. Thus by setting the micrometer to the VI calculated value the desired compression ratio can be achieved. Whenever the compression ratio is changed it becomes necessary to check the spark timing again to set it in the required value. Moreover each time the compression ratio was changed the motoring pressure had to be recorded which was used in the combustion duration calculation. This was done to avoid the variations in the atmospheric pressure, reducing the accuracy of combustion duration analysis.

The spark timing was varied over a wide range of values from 5° BTDC to 40° BTDC (Before top dead center). Depending upon the fuel used different spark timings produce different performances and it becomes necessary to study the MBT spark timing which produces the maximum
brake torque with minimum spark advance. So the experiments were conducted at 5°, 12°, 20°, 25°, 30° and 40° BTDCs. A computerized tachometer/advance timing light device was used in addition to the neon flash light device available in the CFR engine for better accuracy of the spark advance setting.

The equivalence ratio was primarily varied between a wide range of lean to rich limits from 0.6 to 1.2. The equivalence ratio is controlled by controlling the fuel flow from the fuel control panel. The air flow is by the negative pressure in the engine and it is not controlled. By looking at the data flow VI front panel the equivalence ratio can be varied by varying the manual flow controller knobs in the fuel control panel. The experiments were conducted at 0.6, 0.8, 1.0 and 1.2 equivalence ratios. Often when there is poor burning in the lean or rich limits, the nearest equivalence ratios like 0.7 or 1.1 were tried for better burning of the fuel.

For the experiments which involved hydrogen addition with landfill gas, the percentage of hydrogen was varied in the range of 3% to 30% in the fuel. Experiments were conducted at 3%, 5%, 10%, 20% and 30% hydrogen in the fuel. Controlling this percentage of hydrogen was achieved by the fuel control panel and the flow data VI which uses a composition equation solver to instruct exactly what is the flow to be achieved to get the hydrogen percentage in the fuel.
Since the settings change over a period of time, it was necessary to repeat the setup validation tests often to ensure the correctness of the setup. If there were any problems then it had to be corrected before proceeding with the experiments.

The engine was made to run with natural gas for at least 20 minutes before starting the experiments on a day. This helps to warm the engine and improves the stabilization of the combustion process in the combustion chamber.

To conduct an experiment, first the compression ratio is set to the desired value using the micrometer reading. Then the motoring pressure is recorded for that particular compression ratio using the motoring pressure data recording VI. Then the spark timing is set for the desired value. The flow data VI is opened and the desired landfill gas composition is given. The desired equivalence ratio and the desired hydrogen percentage are given. Then the manual flow controllers in the fuel control panel are adjusted one by one to get the desired flow of each gas. Once the desired flow of gases and the desired equivalence ratio is achieved, a 10 minute timer was switched on to allow the combustion in the engine to stabilize. Then the pressure data recording VI is opened and after entering the required input values like Inches of mercury (for atmospheric pressure data), compression ratio, spark timing, intake temperature and exhaust temperature (from the temperature
display panel) the VI is run to record 100 pressure cycles. Then all these data is recorded in a data file specified by the user. Care should be taken for the pressure recording to start from the intake TDC and not to start from compression TDC of the engine cycle. This can be checked by verifying the pressure curve recorded in the pressure data recording VI.

While closing down to end the experiments care was taken not to leave any fuel in the fuel lines. This was achieved by shutting down the gas cylinders while the engine was still on depleting the fuel in the fuel control panel and the entire fuel lines.

3.4 Summary

The alternative fuels research facility was built as desired to do the required landfill gas and biogas experiments. The CFR engine air intake system was modified as desired. The fuel control panel was built to induct four different gases in to the engine at the various mixture proportion and various air fuel ratios. The data acquisition system was also built to collect the in cylinder pressure data and other engine data required for data analysis. Labview VIs were developed for the data acquisition purposes. The entire setup was validated for its correctness and accuracy.
CHAPTER 4

COMBUSTION MODELING

4.1 Background

In order to achieve better fuel economy and lesser emissions from the engine, it becomes necessary to study the performance characteristics of the engine at varying parameters. But performing practical engine experiments is time consuming and costly process. So it becomes necessary to build computer model to predict the combustion process using thermodynamic and heat transfer principles. If such models are built and validated, a large amount of experiments can be avoided, saving time and money. A two zone model was developed to simulate the combustion process in internal combustion engine with a hydrocarbon fuel like Methane along with the diluents like Carbon dioxide and Nitrogen. This was done by studying the combustion model developed by Ferguson [4] and then additional aspects were added for having carbon dioxide and nitrogen as diluents. The equilibrium combustion products were increased to 13 species instead of 10 species developed by Ferguson [4].

The modeling consists of two main modules namely the fuel-air and residual gas properties calculation and equilibrium combustion products’ properties calculation. The fuel-air and residual gas module calculates the gas
properties required for compression phase and the unburned zone in the combustion phase. The equilibrium combustion products module calculates the gas properties for the burned zone in the combustion phase and the expansion phase. These two modules are made use of, in the thermodynamic and heat transfer models, which mainly consists of calculating the pressure rise during compression, combustion and expansion phases.

4.2 Fuel air residual gas properties

As the fuel air mixture enters the engine, it mixes with the residual gas left behind in the cylinder from the previous cycle. This charge is compressed during the compression phase. Calculation of the mole fractions and its properties becomes necessary. The following combustion reaction of a common hydrocarbon consisting of carbon, hydrogen, oxygen and nitrogen along with diluents like carbon dioxide and nitrogen is considered. This is because of the interest for landfill gas and biogas as the fuels. But the same formulations are very well applicable for a hydrocarbon fuel without any diluents by making mp=1, co2p=0 and np=0 where mp is the methane percentage, co2p is the carbon dioxide percentage and np is the nitrogen percentage in the fuel. At low temperatures it is assumed that only six species are formed. In the hydrocarbon fuel α is the no. of carbon atoms, β is the no. of hydrogen atoms, γ is the no. of oxygen atoms and δ is the no. of nitrogen
atoms. \( \varepsilon \) is the stoichiometric fuel for one mole of air and \( \Phi \) is the equivalence ratio. \( n_1 \) to \( n_6 \) are the corresponding no. of moles in the products.

\[
\phi \varepsilon \left[ \text{mp} \left( \alpha \cdot H_\beta \cdot O_\gamma \cdot N_\delta \right) + \text{co2p} \cdot \text{CO}_2 + \text{np} \cdot N_2 \right] + \left( 0.21 \cdot \text{O}_2 + 0.79 \cdot N_2 \right) \rightarrow n_1 \cdot \text{CO}_2 + n_2 \cdot \text{H}_2 \text{O} + n_3 \cdot N_2 + n_4 \cdot \text{O}_2 + n_5 \cdot \text{CO} + n_6 \cdot \text{H}_2
\]

\((4.1)\)

Atom balance equations:

- C \( \phi \varepsilon (\alpha \cdot \text{mp} + \text{co2p}) := n_1 + n_5 \)
- H \( \beta \phi \varepsilon \text{mp} := 2 \cdot n_2 + 2 \cdot n_6 \)
- O \( 0.42 + \phi \varepsilon \text{mp} + 2 \cdot \phi \varepsilon \cdot \text{co2p} := 2n_1 + n_2 + n_5 + 2n_4 \)
- N \( 1.58 + \delta \phi \varepsilon \text{mp} + 2 \cdot \phi \varepsilon \cdot \text{np} := 2n_3 \)

Equilibrium equation:

\[
\text{CO}_2 + \text{H}_2 \rightarrow \text{CO} + \text{H}_2 \text{O}
\]

\((4.2)\)

The equilibrium constant \( K \) for this reaction is given by,

\[
K := \frac{n_2 \cdot n_5}{n_1 - n_6}
\]

\((4.3)\)

For lean and stoichiometric mixtures it is assumed that \( n_5 = 0 \) and \( n_6 = 0 \), so the atom balance equations are sufficient to determine the other unknowns. In the case of rich mixtures \( n_4 = 0 \), and in addition to the four atom balance equations, an equilibrium equation is used to determine the unknowns. The solved results for lean/stoichiometric and rich cases are
shown in the Table 4.1. \( n_5 \) is solved using the atom balance and the equilibrium constant equations, and this was done numerically in the computer.

For the fuel-air and residual gas mixture the equation is modified as follows. The reactant side gases represent the fuel and air mixture. The product side gases represent the residual gases from the previous engine cycle.

\[
\begin{align*}
n_0'\left(C_\alpha \cdot H_\beta \cdot O_\gamma \cdot N_\delta \right) + n_1'CO_2 + n_4'O_2 + n_3'N_2 & \rightarrow \\
n_1''CO_2 + n_2''H_2O + n_3''N_2 + n_4''O_2 + n_5''CO + n_6''H_2
\end{align*}
\]

(4.4)

\( n_0' \) Reactant coefficient.

\( n_0'' \) Product coefficient.

According to Ferguson [4] the mole fractions \( y_i \) and the residual mole fraction \( y_r \) are given as,

\[
y_i := (1 - y_r)y_i' + y_r \cdot y_i''
\]

(4.5)

\[
y_r := \left[1 + \frac{M''}{M'} \cdot \left(\frac{1}{f} - 1\right)\right]^{-1}
\]

(4.6)

where \( f \) is the residual mass fraction.
Table 4.1

Low temperature combustion products of landfill gas (moles/mole of air).

<table>
<thead>
<tr>
<th>i</th>
<th>species</th>
<th>( \Phi \leq 1 )</th>
<th>( \Phi &gt; 1 )</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>CO(_2)</td>
<td>( \phi \alpha \cdot m_p + co2p )</td>
<td>( \phi \alpha \cdot m_p + co2p - n5 )</td>
</tr>
<tr>
<td>2</td>
<td>H(_2)O</td>
<td>( \frac{\beta \phi \alpha \cdot m_p}{2} )</td>
<td>( 0.42 + \phi \alpha \cdot m_p \gamma - 2 \phi \alpha \cdot m_p + n5 )</td>
</tr>
<tr>
<td>3</td>
<td>N(_2)</td>
<td>( 0.79 + \frac{\delta \phi \alpha \cdot m_p}{2} + \phi \alpha \cdot n_p )</td>
<td>( 0.79 + \frac{\delta \phi \alpha \cdot m_p}{2} + \phi \alpha \cdot n_p )</td>
</tr>
<tr>
<td>4</td>
<td>O(_2)</td>
<td>( 0.21(1 - \phi) )</td>
<td>0</td>
</tr>
<tr>
<td>5</td>
<td>CO</td>
<td>0</td>
<td>n5</td>
</tr>
<tr>
<td>6</td>
<td>H(_2)</td>
<td>0</td>
<td>( \frac{\beta \phi \alpha \cdot m_p}{2} - (0.42 + \phi \alpha \cdot m_p \gamma - 2 \phi \alpha \cdot m_p + n5) )</td>
</tr>
</tbody>
</table>

From these mole fraction calculations the mixture properties like enthalpy, internal energy, specific volume, entropy, and specific heat at constant pressure, derivative of log volume with respect to log temperature at constant pressure, and derivative of log volume with respect to log pressure at constant temperature are calculated. For thermodynamic properties at varying temperatures, the interpolated values from the JANAF tables [6] were used, rather than using the polynomial functions curve fitted by minimizing the least squares error.
4.3 Equilibrium combustion products

During combustion the temperatures reach very high levels and it becomes necessary to study a wide set of equilibrium combustion products. In this study 13 species were considered to be the equilibrium combustion products.

\[
\phi \varepsilon \left[ mp \left( C_\alpha \cdot H_\beta \cdot O_\gamma \cdot N_\delta \right) + co2p \cdot CO_2 + np \cdot N_2 \right] + \left( 0.21 \cdot O_2 + 0.79 \cdot N_2 \right) \rightarrow \\
\begin{align*}
n1 \cdot CO_2 + n2 \cdot H_2O + n3 \cdot N_2 + n4 \cdot O_2 + n5 \cdot CO + n6 \cdot H_2 + n7 \cdot H + n8 \cdot O + . \\
n9 \cdot OH + n10 \cdot NO + n11 \cdot N + n12 \cdot NO_2 + n13 \cdot CH_4
\end{align*}
\]

(4.7)

Atom balance equations:

\[
\begin{align*}
C & \quad \phi \varepsilon (\alpha \cdot mp + co2p) := (y_1 + y_5 + y_{13}) \cdot n \cdot . \\
H & \quad \beta \phi \varepsilon mp := (2 \cdot y_2 + 2 \cdot y_6 + y_7 + y_9 + 4 \cdot y_{13}) \cdot n \\
O & \quad \phi \varepsilon \gamma mp + \phi \varepsilon co2p \cdot 2 + 0.42 := (2 \cdot y_1 + y_2 + 2 \cdot y_4 + y_5 + y_8 + y_9 + y_{10} + 2 \cdot y_{12}) \cdot n \\
N & \quad 1.58 + \delta \phi \varepsilon mp + \phi \varepsilon np \cdot 2 := (2 \cdot y_3 + y_{10} + y_{11} + y_{12}) \cdot n \\
\sum_{i=1}^{13} (y_i - 1) := 0
\end{align*}
\]

(4.8)
n is the total no. of moles and \( y_i \) is the corresponding mole fractions. In order to eliminate the \( n \) from the atom balance equations, they are written as follows,

\[
\frac{\beta \cdot mp}{(\alpha \cdot mp + co2p)} = \frac{2 \cdot y_2 + 2 \cdot y_6 + y_7 + y_9 + 4 \cdot y_{13}}{y_1 + y_5 + y_{13}} \quad (4.9)
\]

\[
\frac{\phi \varepsilon \gamma \cdot mp + \phi \varepsilon \cdot co2p \cdot 2 + 0.42}{\phi \delta (\alpha \cdot mp + co2p)} = \frac{2 \cdot y_1 + y_2 + 2 \cdot y_4 + y_5 + y_8 + y_9 + y_{10} + 2 \cdot y_{12}}{y_1 + y_5 + y_{13}} \quad (4.10)
\]

\[
\frac{1.58 + \delta \phi \varepsilon \cdot mp + \phi \varepsilon \cdot np \cdot 2}{\phi \delta (\alpha \cdot mp + co2p)} = \frac{2 \cdot y_3 + y_{10} + y_{11} + y_{12}}{y_1 + y_5 + y_{13}} \quad (4.11)
\]

\[y_1 + y_2 + y_3 + y_4 + y_5 + y_6 + y_7 + y_8 + y_9 + y_{10} + y_{11} + y_{12} + y_{13} - 1 = 0 \quad (4.12)\]

These four equations along with the 9 equilibrium equations given in Table 4.2 are solved to determine the mole fractions. The reference pressure in these equilibrium equations is taken as \( P_o = 1 \) atmospheres. Each species is referred by the number shown in the combustion equation. The equilibrium constant is calculated from the JANAF tables [6]. These equations are non-linear in nature and because of the large no. of equations it is highly impossible to solve it manually and the use computer programs becomes inevitable. These non-linear equations are solved by Newton-Raphson iteration method and the linear equations got at the end of this iterative
process are further solved by Gauss-Jordan method. The further formulations of the equations used in the Matlab software are given in the Appendix A.

Once the mole fractions are found as in the case of fuel-air residual gas section, the thermodynamic properties such as enthalpy, internal energy, specific volume, entropy, specific heat at constant pressure, derivative of log volume with respect to log temperature at constant pressure, and derivative of log volume with respect to log pressure at constant temperature are calculated.

According to Ferguson [4] the thermodynamic properties are given by,

\[
C_p := \left( \frac{\partial h}{\partial T} \right)_p := \frac{1}{M} \left[ \sum_{i=1}^{13} \left( y_i C_{p_i} + h_i \frac{d}{dT} y_i \right) - \frac{h-M_T}{M} \right]
\]

(4.13)

\[
\left( \frac{\partial \ln(v)}{\partial \ln(T)} \right)_p := 1 - \frac{TM}{M} \quad M_T := \frac{\partial M}{\partial T} \quad := \sum_{i=1}^{13} M_i \frac{\partial}{\partial T} y_i
\]

(4.14)

\[
\left( \frac{\partial \ln(v)}{\partial \ln(P)} \right)_T := 1 - \frac{PM}{M} \quad M_P := \frac{\partial M}{\partial P} \quad := \sum_{i=1}^{13} M_i \frac{\partial}{\partial P} y_i
\]

(4.15)
### Table 4.2

Equilibrium reactions, equations and constants.

<table>
<thead>
<tr>
<th>No.</th>
<th>Equilibrium reaction</th>
<th>Equilibrium equation</th>
<th>Equilibrium constant calculation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>$\text{CO} + \frac{1}{2} \text{O}_2 \rightleftharpoons \text{CO}_2$</td>
<td>$k_1(T) = \frac{y_1 \cdot P^{0.5}}{y_5 \cdot (y_4)^{0.5}}$</td>
<td>$k_1(T) := \frac{K(1, T)}{K(5, T) \cdot K(4, T)^{0.5}}$</td>
</tr>
<tr>
<td>2</td>
<td>$\text{H}_2 + \frac{1}{2} \text{O}_2 \rightleftharpoons \text{H}_2\text{O}$</td>
<td>$k_2(T) = \frac{y_2 \cdot P^{0.5}}{y_6 \cdot (y_4)^{0.5}}$</td>
<td>$k_2(T) := \frac{K(2, T)}{K(6, T) \cdot K(4, T)^{0.5}}$</td>
</tr>
<tr>
<td>3</td>
<td>$\frac{1}{2} \text{H}_2 \rightleftharpoons \text{H}$</td>
<td>$k_7(T) = \frac{y_7 \cdot P^{0.5}}{(y_6)^{0.5}}$</td>
<td>$k_7(T) := \frac{K(7, T)}{K(6, T)^{0.5}}$</td>
</tr>
<tr>
<td>4</td>
<td>$\frac{1}{2} \text{O}_2 \rightleftharpoons \text{O}$</td>
<td>$k_8(T) = \frac{y_8 \cdot P^{0.5}}{(y_4)^{0.5}}$</td>
<td>$k_8(T) := \frac{K(8, T)}{K(4, T)^{0.5}}$</td>
</tr>
<tr>
<td>5</td>
<td>$\frac{1}{2} \text{H}_2 + \frac{1}{2} \text{O}_2 \rightleftharpoons \text{OH}$</td>
<td>$k_9(T) = \frac{y_9}{(y_6)^{0.5} \cdot (y_4)^{0.5}}$</td>
<td>$k_9(T) := \frac{K(9, T)}{K(6, T)^{0.5} \cdot K(4, T)^{0.5}}$</td>
</tr>
<tr>
<td>6</td>
<td>$\frac{1}{2} \text{O}_2 + \frac{1}{2} \text{N}_2 \rightleftharpoons \text{NO}$</td>
<td>$k_{10}(T) = \frac{y_{10}}{(y_4)^{0.5} \cdot (y_3)^{0.5}}$</td>
<td>$k_{10}(T) := \frac{K(10, T)}{K(4, T)^{0.5} \cdot K(3, T)^{0.5}}$</td>
</tr>
<tr>
<td>7</td>
<td>$\frac{1}{2} \text{N}_2 \rightleftharpoons \text{N}$</td>
<td>$k_{11}(T) = \frac{y_{11} \cdot P^{0.5}}{(y_3)^{0.5}}$</td>
<td>$k_{11}(T) := \frac{K(11, T)}{K(3, T)^{0.5}}$</td>
</tr>
<tr>
<td>8</td>
<td>$\frac{1}{2} \text{N}_2 + \text{O}_2 \rightleftharpoons \text{NO}_2$</td>
<td>$k_{12}(T) = \frac{y_{12} \cdot P^{0.5}}{y_{10} \cdot (y_4)^{0.5}}$</td>
<td>$k_{12}(T) := \frac{K(12, T)}{K(10, T) \cdot K(4, T)^{0.5}}$</td>
</tr>
<tr>
<td>9</td>
<td>$4 \text{H}_2 + \text{CO}_2 \rightleftharpoons \text{CH}_4 + 2 \text{H}_2$</td>
<td>$k_{13}(T) = \frac{y_{13} \cdot (y_2)^2 \cdot P^{2.5}}{(y_6)^4 \cdot y_1}$</td>
<td>$k_{13}(T) := \frac{K(13, T) \cdot K(2, T)^2}{K(6, T)^4 \cdot K(1, T)}$</td>
</tr>
</tbody>
</table>
4.4 Heat release formulations

The heat release model given by Ferguson is made use in the combustion modeling software which acts as the main program to calculate the pressure in the cylinder using a two zone model in the combustion chamber. The combustion volume is considered to be of two zones namely burned zone and unburned zone. At the start of the combustion the entire volume is assumed to be of unburned zone and at the end of combustion the entire volume is assumed to be of burned zone. During the combustion as the flame propagates through the chamber, the volume ahead of the flame is considered to be the unburned zone and the volume consumed by the flame is considered to the burned zone.

The energy equation of the cylinder contents is given by

$$m \frac{du}{d\theta} + u \frac{dm}{d\theta} := \frac{dQ}{d\theta} - P \frac{dV}{d\theta} - \frac{m_1' h_1}{\omega} \tag{4.16}$$

The internal energy $u$ can be written in terms of the internal energy of unburned zone $u_u$ and burned zone $u_b$. $x$ is the mass fraction burned.

$$u := \frac{U}{m} := xu_b + (1 - x)u_u \tag{4.17}$$

Similarly the specific volume $v$ can be written in terms of specific volume of the unburned zone $v_u$ and burned zone $v_b$. 
\[ \nu := \frac{V}{m} = \nu_b + (1 - \chi) \nu_u \]  

(4.18)

Differentiating these internal energy and specific volume terms with respect to crank angle and replacing the partial derivative terms with the logarithmic forms yields,

\[
\frac{du}{d\theta} := \left[ C_{p_b} - \frac{P \nu_b}{T_b} \left( \frac{\partial}{\partial \ln(T_b)} \ln(\nu_b) \right) \right] \frac{dT_b}{d\theta} - \nu_b \left( \frac{\partial}{\partial \ln(T_b)} \ln(\nu_b) + \frac{\partial}{\partial \ln(P)} \ln(\nu_b) \right) \frac{dP}{d\theta}
\]

(4.21)

\[
\frac{du_u}{d\theta} := \left[ C_{p_u} - \frac{P \nu_u}{T_u} \left( \frac{\partial}{\partial \ln(T_u)} \ln(\nu_u) \right) \right] \frac{dT_u}{d\theta} - \nu_u \left( \frac{\partial}{\partial \ln(T_u)} \ln(\nu_u) + \frac{\partial}{\partial \ln(P)} \ln(\nu_u) \right) \frac{dP}{d\theta}
\]

(4.22)

This is because the fuel-air residual gas and equilibrium combustion products modules are developed so as to give the logarithmic forms as the output. The first term in Equation 4.16 can be further written as

\[
m \frac{du}{d\theta} := \left[ \frac{du_b}{d\theta} + \frac{du_u}{d\theta} \right] = x \frac{du_b}{d\theta} + (1 - x) \frac{du_u}{d\theta} + \left( u_b - u_u \right) \frac{dx}{d\theta} \cdot m
\]

(4.23)

and the second term in this Equation can be written as follows,
\[
\frac{dm}{d\theta} = \frac{m_1'}{\omega} \quad := -\frac{Cm}{\omega}
\]  

(4.24)

where \( C \) is a blowby constant and \( \omega \) is the engine frequency in radians per second. This mass leakage depends upon the ring design. The heat loss from the chamber can also be considered to be happening from two different zones, the burned and unburned.

\[
\frac{dQ}{d\theta} = \frac{Q_b'}{\omega} \quad := \frac{-Q_b' - Q_u'}{\omega}
\]  

(4.25)

\[Q_b' := h \cdot A_b \cdot (T_b - T_w)\]  

(4.26)

\[Q_u' := h \cdot A_u \cdot (T_u - T_w)\]  

(4.27)

The surface areas of the two zones are given by \( A_u \) and \( A_b \). These areas can be related to the mass fraction burned \( x \), using an empirical formulation [4],

\[
A_b := \left( \frac{\pi b^2}{2} + \frac{4V}{b} \right) \cdot \frac{1}{x^2}
\]  

(4.28)

\[
A_u := \left( \frac{\pi b^2}{2} + \frac{4V}{b} \right) \cdot \left( 1 - \frac{1}{x^2} \right)
\]  

(4.29)

where \( b \) is the bore of the cylinder. The convective heat transfer coefficient \( h \), is not a constant and is given by Woschni’s correlation,

\[
h := 3.26 b^{-0.2} \cdot p^{0.8} \cdot T^{-0.55} \cdot w^{0.8}
\]  

(4.30)
where \( b \) the cylinder bore is in meters, instantaneous pressure \( P \) is in kPa, temperature \( T \) is in K and the average cylinder gas velocity \( w \) is in m/s. The average gas velocity is given by,

\[
w := C_1 \cdot S_p \cdot + C_2 \cdot \frac{V_d \cdot T_r}{P_r \cdot V_r} \left( P - P_m \right)
\]

(4.31)

where \( S_p \) is the mean piston speed given by,

\[
S_p = 2LN
\]

(4.32)

where \( L \) is the stroke and \( N \) is the rotational speed. The constants \( C_1 \) and \( C_2 \) are given by,

\[
C_1 := 2.28 \quad C_2 := 0
\]

for compression period and

\[
C_1 := 2.28 \quad C_2 := 3.24 \times 10^{-3}
\]

for combustion and expansion period. \( V_d \) is the displacement volume and \( P_r \), \( T_r \) and \( V_r \) are the pressure, temperature and volume at a reference point which is usually the start of compression. \( P_m \) is the motored pressure at the corresponding crank angle.

The enthalpy loss can be given as,

\[
h_1 := \left( 1 - x^2 \right) h_u + x^2 \cdot h_b
\]

(4.33)

The instantaneous mass is given by,
where $m_1$ is the mass at the start of compression.

The instantaneous volume is given by following equations.

\[ V := V_c + \frac{\pi b^2}{4} (1 + a - s) \]  \hspace{1cm} (4.35)

\[ V_c := \frac{V_d}{CR - 1} \]  \hspace{1cm} (4.36)

\[ s := a \cdot \cos(\theta) + \sqrt{l^2 - a^2 \cdot \sin^2(\theta)} \]  \hspace{1cm} (4.37)

where $V_c$ is the clearance volume which depends upon the compression ratio $CR$. $l$ is the connecting rod length and $a$ is the crank radius which is also given by half of the stroke. The mass fraction burned is given by the Wiebe function.

\[ x := 1 - \exp \left[ -g \cdot \left( \frac{\theta - \theta_s}{\theta_b} \right)^{j+1} \right] \]  \hspace{1cm} (4.38)

where $\theta_s$ is the start of combustion and $\theta_b$ is the combustion duration or the burn angle. $g$ and $j$ are the constants which depend upon a given engine. So it becomes necessary for us to fit the Wiebe function curve with the experimentally derived mass fraction burned curve and find out the values of $g$ and $j$. Once $g$ and $j$ are found then this result can be used to model the
combustion with different parameters in the same engine. The g and j
determination for this study is further discussed in the validation of this
model section.

The entropy terms are introduced into the heat loss function as follows,

\[-Q_u' := \omega \cdot m \cdot (1 - \chi) \cdot T_u \cdot \frac{d s_u}{d \theta} \tag{4.39}\]

\[\frac{d s_u}{d \theta} := \frac{C_p u}{T_u} \cdot \frac{dT_u}{d \theta} - \frac{v_u}{T_u} \left( \frac{\partial}{\partial \ln(T_u)} \ln(v_u) \right) \cdot \frac{d P}{d \theta} \tag{4.40}\]

All these foregoing discussions are summarized in the A, B, C', D and
E equations as follows which are further utilized in the pressure, burned
temperature and unburned temperature derivatives. These derivatives are
utilized in the Matlab program as the integration equations to find out the
instantaneous pressure, burned temperature and unburned temperature. The
Matlab codes developed for this combustion model is given in the Appendix
B.

\[A := \frac{1}{m} \left( \frac{d V}{d \theta} + \frac{V C}{\omega} \right) \tag{4.41}\]

\[B := h \left( \frac{\pi b^2}{2} + \frac{4 V}{b} \right) \left[ \frac{v_b}{C_p b} \left( \frac{d}{d \ln(T_b)} \ln(v_b) \right) \cdot \left( \frac{1}{2} \right) \frac{T_b - T_w}{T_b} + \frac{v_u}{C_p u} \left( \frac{d}{d \ln(T_u)} \ln(v_u) \right) \cdot \left( \frac{1}{2} \right) \frac{T_u - T_w}{T_u} \right] \tag{4.42}\]
\[ C' := \left( \nu_b - \nu_u \right) \frac{dx}{d\theta} - \nu_b \left( \frac{d}{d\ln(T_b)} \ln(\nu_b) \right) \frac{h_u - h_b}{C_{p}\cdot T_b} \left[ \frac{dx}{d\theta} - \frac{(x - x^2)C}{\omega} \right] \]  
\[ (4.43) \]

\[ D := x \left[ \frac{(\nu_b)^2}{C_{p}\cdot T_b} \left( \frac{d}{d\ln(T_b)} \ln(\nu_b) \right)^2 + \nu_b \left( \frac{\partial}{\partial \ln(P)} \ln(\nu_b) \right) \right] \]  
\[ (4.44) \]

\[ E := (1 - x) \left[ \frac{(\nu_u)^2}{C_{p}\cdot T_u} \left( \frac{d}{d\ln(T_u)} \ln(\nu_u) \right)^2 + \nu_u \left( \frac{\partial}{\partial \ln(P)} \ln(\nu_u) \right) \right] \]  
\[ (4.45) \]

Integration equations:

\[ \frac{dP}{d\theta} = \frac{A + B + C'}{D + E} \]  
\[ (4.46) \]

\[ \frac{dT_b}{d\theta} := \frac{\pi b^2}{4} \frac{1}{b^2} \frac{x^2}{\omega \cdot m \cdot C_{p} \cdot x} \left( T_b - T_w \right) + \left[ \frac{\nu_b}{C_{p}\cdot T_b} \left( \frac{d}{d\ln(T_b)} \ln(\nu_b) \right) \right] \left( \frac{A + B + C'}{D + E} \right) + \frac{h_u - h_b}{C_{p}\cdot T_b} \left[ \frac{dx}{d\theta} - \frac{(x - x^2)C}{\omega} \right] \]  
\[ (4.47) \]

\[ \frac{dT_u}{d\theta} := \frac{\pi b^2}{4} \frac{1}{b^2} \frac{x^2}{\omega \cdot m \cdot C_{p} \cdot (1 - x)} \left( T_u - T_w \right) + \left[ \frac{\nu_u}{C_{p}\cdot T_u} \left( \frac{d}{d\ln(T_u)} \ln(\nu_u) \right) \right] \left( \frac{A + B + C'}{D + E} \right) \]  
\[ (4.48) \]

4.4.1 Wiebe function fitting

The functional form of the mass fraction burned is given by the Wiebe function. In order to get the constants involved in the Wiebe function which is
engine dependent, it becomes necessary to fit the Wiebe function for the mass fraction burned curve derived from experimental pressure data. The mass fraction burned from experimental pressure data is given by,

$$x_b := \frac{1}{P^n \cdot V - \left(\frac{P_o}{P_o}\right)^n \cdot V_o} \frac{1}{1 - \left(\frac{P_f}{P_o}\right)^n \cdot V_o}$$

(4.49)

where the o subscript refers to the start of combustion and f subscript refers to the end of combustion. The polytropic index n is assumed to be a constant of 1.3. This is an approximation as the polytropic index is not a constant during the combustion process.

The Wiebe function in Equation 4.38 contains the engine dependent constants g and j. Figure 4.1 shows the mass fraction burned curve derived from experimental pressure data and the Wiebe function. The values of g and j were determined to be g=5 and j=4.5 for the engine used in this study. Once the constants are determined the mass fraction burned can be determined from the Wiebe function for the model. The maximum percentage of deviation of the Wiebe function from the experimental values was found to be 30.7%.
Figure 4.1 Wiebe function fitted for mass fraction burned from the experimental pressure data in the CFR engine with methane as fuel at compression ratio 8.5.
4.5 Validation of the model and testing of the software

4.5.1 Equilibrium combustion products module

As the equilibrium combustion products module consists of a large no. of complicated non-linear equation solving tasks it becomes necessary to validate this model separately. The mole fractions of the equilibrium combustion species for Isooctane fuel at a pressure of 30 atm. and a temperature of 1750° K was compared with the published data [5] as shown in Figure 4.2. The same procedure was carried out for other temperatures and was found to agree well. The Figure 4.3 shows the mole fractions of the equilibrium combustion products of methane and it can be seen the trend of the mole fractions of the species is same as found in isoctane as fuel. Further the module was checked for landfill gas and its hydrogen mixtures as fuel and the results are shown in the Figures 4.4 and 4.5 respectively. It can be observed that at low temperatures the significant products are CO₂, H₂O, N₂, O₂, CO and H₂. As the temperature rises it can be seen that other species like OH, NO, O etc. becomes very significant.
Figure 4.2 Comparison of the mole fractions of equilibrium combustion species with the published data [5] at a pressure of 30 atm. and temperature 1750° K for Isooctane as fuel. (symbols are from [5] and lines are from program results).

Figure 4.3 Program results of equilibrium combustion products of methane showing the mole fractions of the species formed at a pressure of 30 atm. and temperature 1750° K.
Figure 4.4  Program results of equilibrium combustion products of landfill gas showing the mole fractions of the species formed at a pressure of 30 atm. and temperature 2250° K.

Figure 4.5  Program results of equilibrium combustion products of landfill gas with hydrogen showing the mole fractions of the species formed at a pressure of 30 atm. and temperature 2750° K.
4.5.2 Results of heat release module

The model was tested for landfill gas as the fuel. For landfill gas the composition was 53% methane, 42% carbon dioxide and 5% nitrogen. So the model was tested with $mp=0.53$, $co2p=0.42$ and $np=0.05$. When methane was considered to be the fuel, the model was tested with $mp=1$, $co2p=0$ and $np=0$. The results were compared with the experimental pressure curves got from the experiments.

![Graph showing experimental and modeled pressure comparison.](image)

**Figure 4.6** Comparison of experimental pressure and modeled pressure in the CFR engine with landfill gas as fuel at compression ratio 8.5, spark timing 25° BTDC, equivalence ratio 0.8, intake temperature 303° k, intake pressure 98kPa and 600 rpm.
Figure 4.7 Comparison of experimental pressure and modeled pressure in the CFR engine with landfill gas as fuel at compression ratio 8.5, spark timing 25° BTDC, equivalence ratio 1.0, intake temperature 303° k, intake pressure 98kPa and 600 rpm.

The constant terms involved in the model such as the residual fraction $f$, mass blowby constant $C$, the wall temperature $T_w$, volumetric efficiency, combustion start angle, combustion duration angle had a significant effect in the pressure curve obtained.

The Figures 4.6 and 4.7 shows the comparison of the modeled pressure trace diagram to the corresponding experimental data. It was observed that the modeled pressure trace was in good agreement with the experimental pressure. The combustion phase pressure curve mainly depends upon the ignition lag, combustion start angle and combustion duration. This is particularly a result got from the experimental data. Variations in these
values cause a significant deviation in the combustion phase. One of the observation was the peak pressure is inversely related to the combustion duration. So it becomes necessary to input the appropriate combustion duration details for the modeled pressure to agree with the experimental results.

This combustion modeling software can be used as a diagnostic tool for verifying the experimental results. Since the model is incorporated with diluents in the fuel, it can be a very good tool to analyze the low Btu fuels like landfill gas, biogas and other gases which have some hydrocarbon fuel as its main content and other gases like carbon dioxide and nitrogen as the diluents.

4.6 Combustion duration analysis and estimation

As the model described above required combustion duration as an input, it becomes necessary to do the combustion duration analysis from the experimental data. Further a method to estimate the combustion duration without conducting the experiments becomes necessary in order to predict the engine performances at different input parameters.
4.6.1 Combustion duration analysis

Figure 4.8 Typical pressure curve analysis to determine combustion duration from the experimental data.

a – Ignition spark timing.

b – Start combustion. (The point where the combustion pressure curve separates from the motoring curve. (Figure 4.8))

c – End of combustion. (The point at which the polytropic index reaches a value close to 1.3)

Ignition lag = b-a

Combustion duration = c-b

(4.50)
Polytropic index is given by, \( n \) which is calculated for a small crank angle increment say 0.1°.

\[
\log \left( \frac{p_1}{p_2} \right) = \log \left( \frac{v_2}{v_1} \right) \]

During combustion polytropic index \( n \) is not a constant. Once the combustion is ended the polytropic index reaches a constant value close to 1.3. The typical variation of polytropic index with crank angle is shown in Figure 4.9.

![Figure 4.9](image-url)  
Figure 4.9 Typical variations of polytropic index with crank angle. The combustion duration \( \Delta \theta_c \) is shown.
4.6.2 Combustion duration estimation

The combustion duration in a spark ignition engine can be estimated using the empirical relations given by Shrestha and Karim [15]. The typical variations in the combustion duration are shown in Figure 4.10. The minimum combustion duration occurs somewhere near stoichiometric operation.

![Figure 4.10](image)

Figure 4.10 Typical variations of experimentally derived combustion duration vs. equivalence ratio in the CFR engine with methane as fuel at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Using the experimental combustion duration curve, the combustion duration $\Delta \theta_c$ is given by,

$$\Delta \theta_c := A \cdot e^x + B \cdot e^z$$

(4.52)
Where

\[ z := \frac{- \left( \phi_{\text{min}} - \phi \right)}{\sqrt{\phi_r - \phi}} \]

and

\[ x := \frac{\left( \phi_{\text{min}} - \phi \right)}{\sqrt{\phi - \phi_l}} \]

\( \phi_{\text{min}} \) is the equivalence ratio at which minimum combustion duration occurs, \( \phi_l \) is the lean limit equivalence ratio and \( \phi_r \) is the rich limit equivalence ratio.

It has been suggested [15] that the lean limit equivalence ratio is a function of temperature at the spark timing. Hence it can be formulated as follows with \( A1 \) and \( B1 \) as constants.

\[ \phi_l := A1 \cdot Tst + B1 \]

\( \phi_{l} \) := \( A1 \cdot Tst + B1 \)  
(4.55)

Similarly the rich limit equivalence ratio is a function of the difference in temperature at spark timing and the intake temperature with \( A2 \) and \( B2 \) as constants.

\[ \phi_r := A2 \cdot (Tst - To) + B2 \]

\( \phi_r \) := \( A2 \cdot (Tst - To) + B2 \)  
(4.56)
The lean and rich limit equivalence ratios can be estimated by conducting minimum no. of experiments to find the constants which are engine dependent.

The A and B in the Equation 4.52 can be given in terms of minimum combustion duration as follows.

$$A := \Delta \theta_{cmin} \frac{\sqrt{\phi_{min} - \phi_l}}{\sqrt{\phi_r - \phi_{min}}} \frac{\phi_{min} - \phi_l}{1 + \sqrt{\phi_{min} - \phi_l}}$$

$$B := \left( \frac{\Delta \theta_{cmin}}{1 + \sqrt{\phi_{min} - \phi_l}} \right)$$

(4.57)

(4.58)

The minimum combustion duration can be got by conducting minimum no. of experiments. However this minimum combustion duration can be formulated as a function of flame propagation distance $d_c$ and maximum flame speed $S_f$.

$$\Delta \theta_{cmin} := f\left(d_c, S_f\right)$$

(4.59)

The flame propagation distance can be replaced with the volume at spark timing and the compression ratio as an approximation as follows.
\[ \Delta \theta_{\text{cmin}} := C \cdot \frac{1}{\left( V_{\text{st}} \right)^3} \cdot f(S_f) \cdot \frac{1}{\text{CR}^2} \]  

(4.60)

The maximum flame speed which was observed [15] to occur near stoichiometric conditions for methane operation was empirically correlated by,

\[ S_{f, \text{max}} := P^{-0.457} \cdot \exp \left( \frac{-746.8}{T} + 6.193 \right) \]  

(4.61)

where \( P \) and \( T \) are the instantaneous pressure and temperature during the combustion. Since the turbulence affects the combustion duration it can be approximated as a function of the mean piston speed and incorporated in the above formulations as follows.

\[ \Delta \theta_{\text{cmin}} := C \cdot \left( V_{\text{st}} \right)^3 \cdot P^{0.457} \cdot e \left( \frac{764.8}{T} \right) \cdot \frac{1}{\left( S_p \right)^3} \cdot \frac{1}{\text{CR}^2} \]  

(4.62)

Based upon the above formulations the ignition lag can also be calculated.

The minimum ignition lag is given by,
\[ \Delta \theta_{\text{igmin}} = \frac{1}{C''} \left( \frac{1}{T_{\text{st}}} \right)^2 \left( \frac{1}{S_p} \right)^3 \left( \frac{1}{CR^2} \right) \]  

(4.63)

The constants \( C' \) and \( C'' \) are the empirical constants that can be calculated by conducting minimum no. of experiments.

Based upon the above method combustion duration and ignition lag for methane operation were estimated for compression ratio 8.5 and spark timings 30 BTDC and 20 BTDC. The results were compared with the experimental data and are shown in Figures 4.11 and 4.12. The constant values involved in these calculations are \( A_1=0.001756 \), \( B_1=-0.319 \), \( A_2=0.000878 \), \( B_2=1.027 \), \( C'=0.375 \) and \( C''=0.89 \). It can be seen that the estimated combustion duration and ignition lag are in reasonable agreement with the experimentally derived data. However there are some deviations observed in the very lean and rich limit operation. The corresponding percentage deviations are shown in Table 4.3 for the spark timings of 30 BTDC and 20 BTDC.
4.6.3 Estimation of combustion duration for landfill gas and hydrogen mixtures

When fuel mixtures are used as fuel then in addition to the above method it becomes necessary to include the mole fractions of the fuels involved. One of the method suggested is [16] by knowing the combustion duration when individual fuels are used in the same engine and apply the following formulation to estimate the combustion duration of the fuel mixture.

\[
\frac{1}{\Delta \theta_{c,m}} := \frac{y_1}{\Delta \theta_{c1}} + \frac{y_2}{\Delta \theta_{c2}} + \frac{y_3}{\Delta \theta_{c3}} + \ldots
\]  
\tag{4.64}

Table 4.3

Percentage deviation of the estimated results from the experimental results.

<table>
<thead>
<tr>
<th></th>
<th>Combustion Duration</th>
<th>Ignition Lag</th>
</tr>
</thead>
<tbody>
<tr>
<td>EQR</td>
<td>ST30</td>
<td>ST20</td>
</tr>
<tr>
<td>0.6</td>
<td>8.5</td>
<td>16.6</td>
</tr>
<tr>
<td>0.8</td>
<td>2.5</td>
<td>1.2</td>
</tr>
<tr>
<td>1</td>
<td>0.1</td>
<td>0.2</td>
</tr>
<tr>
<td>1.2</td>
<td>0.2</td>
<td>17.2</td>
</tr>
</tbody>
</table>
Figure 4.11 A comparison between estimated combustion duration vs. equivalence ratio and experimental data for two spark timings in the CFR engine with methane as fuel at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 4.12 A comparison between estimated ignition lag vs. equivalence ratio and experimental data for two spark timings in the CFR engine with methane as fuel at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
\( \Delta \theta_{c,m} \) is the combustion duration of the fuel mixture, \( \Delta \theta_{c,i} \) is the combustion duration of the individual fuels and \( y_i \) is the mole fraction of the individual fuel.

In the case of landfill gas methane is present along with gases like carbon dioxide and nitrogen. Hydrogen is used as an additive to landfill gas in the present study. So it becomes necessary to estimate the combustion duration as a function of the diluents present in the landfill gas.

An empirical correlation was developed for the combustion duration results derived from the landfill gas experiments.

\[
\frac{100}{\Delta \theta_{c,\text{LFG}}} := \frac{mp}{\Delta \theta_c} + b \cdot \text{co2p} + c \cdot \text{np} \tag{4.65}
\]

where \( \Delta \theta_{c,\text{LFG}} \) is the estimated combustion duration of the landfill gas operation and \( \Delta \theta_c \) is the corresponding combustion duration of the methane operation which is known by the estimation method described. \( mp \) is the methane percentage, \( \text{co2p} \) is the carbon dioxide percentage and \( \text{np} \) is the nitrogen percentage in the landfill gas. \( b \) and \( c \) are the empirical constants which are found by conducting minimum no. of experiments with three different compositions of landfill gas. The values for these constants were found to be \( b=0.016 \) and \( c=-0.045 \) for the CFR engine used.
Similar to the above procedure an empirical correlation was developed for the landfill gas with hydrogen mixtures.

\[
\frac{100}{\Delta \theta_{c,m}} := \frac{100 - h2p}{\Delta \theta_{c,LFG}} + e \cdot h2p
\]  

(4.66)

where \(\Delta \theta_{c,LFG}\) is the estimated combustion duration of the landfill gas operation and \(\Delta \theta_{c,m}\) is the estimated combustion duration of the landfill gas with hydrogen mixture operation. \(h2p\) is the hydrogen percentage in the fuel mixtures of landfill gas and hydrogen. \(e\) is the empirical constant. The value for this constant was found to be \(e=0.064\) for the CFR engine used. The ignition lag was estimated by calculating the crank angle required for the mass fraction burned to attain 10% using the Wiebe function [5].

The Figures 4.13 and 4.14 show the comparison of the estimated combustion duration with experimentally derived values of landfill gas and its hydrogen mixtures respectively. It can be seen that the combustion duration estimates from the empirical relation are in good agreement with the experimentally derived values. The percentage deviations of the estimated values from the experimental values are within the experimental uncertainty as shown given in Table 4.4.
Using the above combustion duration estimation methods for methane, landfill gas and landfill gas with hydrogen mixtures the corresponding combustion duration can be estimated and this can be used as

Table 4.4

Percentage deviations of estimated combustion durations.

<table>
<thead>
<tr>
<th>EQR</th>
<th>Landfill gas</th>
<th>Landfill gas with Hydrogen mixture</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6</td>
<td>-</td>
<td>9.5</td>
</tr>
<tr>
<td>0.7</td>
<td>2.3</td>
<td>-</td>
</tr>
<tr>
<td>0.8</td>
<td>2.6</td>
<td>3.3</td>
</tr>
<tr>
<td>1</td>
<td>3.7</td>
<td>9.4</td>
</tr>
<tr>
<td>1.1</td>
<td>9.8</td>
<td>-</td>
</tr>
<tr>
<td>1.2</td>
<td>-</td>
<td>15.8</td>
</tr>
</tbody>
</table>

Table 4.5

Percentage deviations of predicted indicated power output.

<table>
<thead>
<tr>
<th>EQR</th>
<th>Methane</th>
<th>Landfill gas</th>
<th>Landfill gas with Hydrogen mixture</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.6</td>
<td>7.9</td>
<td>7.3</td>
<td>6</td>
</tr>
<tr>
<td>0.8</td>
<td>1.1</td>
<td>5.2</td>
<td>2.5</td>
</tr>
<tr>
<td>1</td>
<td>2.8</td>
<td>4.7</td>
<td>1.4</td>
</tr>
<tr>
<td>1.2</td>
<td>4.5</td>
<td>5.9</td>
<td>20</td>
</tr>
</tbody>
</table>
Figure 4.13 A comparison between estimated combustion duration vs. equivalence ratio and experimental data for landfill gas operation in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 4.14 A comparison between estimated combustion duration vs. equivalence ratio and experimental data for landfill gas with 20% hydrogen operation in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 4.15 A comparison between predicted indicated power output vs. equivalence ratio and experimental data for methane operation in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 4.16 A comparison between predicted indicated power output vs. equivalence ratio and experimental data for landfill gas operation and its 3% hydrogen mixture in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
an input for the computer model developed which can predict the pressure curve from which engine performance can be predicted. It can be seen from the Figures 4.15 and 4.16 that the predicted values of engine performance for methane, landfill gas and landfill gas with hydrogen mixture operation are in good agreement with the experimental values. The percentage deviations of the estimated values from the experimental values are given in Table 4.5.

4.7 Summary

A computer model was developed using the thermodynamic and heat transfer principles. Matlab software was used for this purpose. The combustion duration is estimated from the methods discussed. The process parameters are given along with the estimated combustion duration as input to the software to model the in-cylinder pressure curve. Using this in-cylinder pressure curve the engine performances including the power output can be predicted. This computer model can be used as a predictive model to predict the engine performances avoiding more number of costly time consuming experiments.
CHAPTER 5

RESULTS AND DISCUSSIONS

5.1 Methane experiments

The landfill gas and other biogases mainly consist of methane and other diluents like carbon dioxide and nitrogen. In order to understand the landfill gas and other biogases combustion properties better, it becomes necessary to understand the methane’s combustion properties. If the methane experimental results are available then it is possible to compare the landfill gas with the baseline fuel which is methane in this case. So methane was tested as a baseline fuel for the CFR spark ignition engine.

With methane as 100% fuel and the compression ratio at 8.5, the equivalence ratio was varied over a wide range from 0.6 to 1.2. From Figure 5.1 it is can be seen that the methane burned well from leaner side to richer side for a wide range of spark timings from 5° to 30° BTDC (Before top dead center). The best combustion was happening at the spark timings 10°, 12° and 20° BTDC. Among these three spark timings 12° BTDC produced the maximum power in most of the equivalence ratios. The minimum spark advance required to get the maximum brake torque is called the MBT spark timing. In this case since the spark timing 12° BTDC was producing the
maximum power it is the MBT spark timing for this compression ratio 8.5 with methane as fuel.

![Graph showing indicated power vs. equivalence ratio for various spark timings in the CFR engine with methane operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.](image)

Figure 5.1 The indicated power vs. equivalence ratio for various spark timings in the CFR engine with methane operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

In Figure 5.2 having spark timing at 12° BTDC and varying the equivalence ratio from 0.6 to 1.2 the performance was studied for different compression ratios from 7 to 10. Here again the stoichiometric combustion produced the maximum power and the fuel combusted well for the entire range from 0.6 to 1.2 equivalence ratio for all the compression ratios selected.
Figure 5.2 The indicated power vs. equivalence ratio for various compression ratios in the CFR engine with methane operation at spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.3 The indicated power vs. spark timing for various equivalence ratios in the CFR engine with methane operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
As the compression ratio is increased the performance improves in all the equivalence ratios. Figure 5.5 also shows the same performance improvement as the compression ratio is increased. This may not be true for very high compression ratios like 16, as the knocking phenomenon occurs in higher compression ratios which causes power loss and engine damage which is undesirable. So increasing the compression ratio is restricted by the knocking phenomenon.

In Figure 5.3 the variation of power with varying spark timing BTDC is shown. It can be seen that the equivalence ratio 1.0 and 0.8 gives good performance as compared to the equivalence ratios 0.6 and 1.2. It can be further seen that the spark timing 12° BTDC gives the maximum power which confirms that it is the MBT spark timing for compression ratio 8.5. This is further depicted in Figure 5.4 showing the maximum power is produced by the spark timing 12° BTDC.
Figure 5.4  The indicated power vs. spark timing for equivalence ratio 1.0 in the CFR engine with methane operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.5  The indicated power vs. compression ratio for various equivalence ratios in the CFR engine with methane operation at spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
5.2 Landfill gas experiments

Once the methane experiments were done as baseline experiments, the landfill gas composition was tested. Since the landfill gas composition is not a constant and it varies over time and with landfill sites, an average landfill gas composition was tested. Information about a wide range of landfill gas compositions was collected and the average of all the compositions was calculated. The various landfill gas composition collected from various resources [17, 18, 19, 20, 21] are shown in Table 5.1. Based on these data the average landfill gas composition was calculated to be 53% methane, 42% carbon dioxide and 5% nitrogen. All the experiments done in this study was done with this average composition. However to get a better idea of other landfill gas compositions the maximum and minimum methane percentages were also tested for minimum no. of experiments. The maximum landfill gas composition was taken as 60% methane, 31% carbon dioxide and 9% nitrogen. The minimum landfill gas composition was taken as 45% methane, 52% carbon dioxide and 3% nitrogen. These compositions were taken based upon the fact [14] that the carbon dioxide has a higher flame temperature reducing property compared to that of nitrogen.
Table 5.1

Landfill gas compositions.

<table>
<thead>
<tr>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
<th></th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Methane</td>
<td>45-60%</td>
<td>47.4%</td>
<td>54%</td>
<td>56%</td>
<td>45-60%</td>
</tr>
<tr>
<td>2</td>
<td>Carbon dioxide</td>
<td>40-60%</td>
<td>47%</td>
<td>40%</td>
<td>31%</td>
<td>40-60%</td>
</tr>
<tr>
<td>3</td>
<td>Nitrogen</td>
<td>2-5%</td>
<td>3.7%</td>
<td>4%</td>
<td>10%</td>
<td>2-5%</td>
</tr>
</tbody>
</table>

The similar procedure as done in methane experiments was followed for landfill gas experiments too. From Figure 5.6 it can be seen that while the equivalence ratio was varied from 0.6 to 1.2 for compression ratio 8.5, some of them did not burn at all. All of them did not burn in the richer side after 1.1 equivalence ratios. This was not the case in methane were it was burning for all spark timings till 1.2 equivalence ratio. In the rich side the amount of air becomes too low as the equivalence ratio is increased and the incombustible diluents like carbon dioxide and nitrogen in the landfill gas hinders the combustion process making it hard for the landfill gas to burn beyond equivalence ratio 1.1.
Figure 5.6 The indicated power vs. equivalence ratio for various spark timings in the CFR engine with landfill gas operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Similarly in the lean side the air is in excess than required for the combustion of methane in the landfill gas and the incombustible diluents hinders the combustion process reducing the lean operational limits. This shows that the lean and rich side is significantly constricted as compared to the methane by the presence of the incombustible diluents. The stoichiometric combustion gives the maximum power in all the spark timings. One more important observation to be made here is that the rich side performance was very poor in comparison with the stoichiometric and its lean equivalence ratio counter parts. This indicated that landfill gas cannot be effectively utilized in the rich equivalence ratios.
Figure 5.7 The indicated power vs. equivalence ratio for various spark timings in the CFR engine with landfill gas operation at compression ratio 12, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

The same experiments were repeated for a higher compression ratio 12 and the results are shown in Figure 5.7. There is a significant improvement in the performance of landfill gas as the compression ratio is increased from 8.5 to 12. The lean side combustion is better at equivalence ratio 0.6 as compared to that of the compression ratio 8.5. But the rich side equivalence ratio followed a similar trend and did not burn after equivalence ratio 1.1. Here too the stoichiometric combustion gives the maximum power. It can also be seen that the spark timing 25 ° BTDC is performing better than other spark timings.
Figure 5.8 The indicated power vs. equivalence ratio for various compression ratios in the CFR engine with landfill gas operation at spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Landfill gas was experimented with differing compression ratios as shown in Figure 5.8. For the spark timing 25 ° BTDC it can be seen that the landfill gas performs in all the compression ratios from equivalence ratio 0.6 to 1.1. The inadequate burning in the rich side is further confirmed here as it didn’t burn at equivalence ratio at 1.2 and beyond. It can be seen that as the compression ratio is increased from 7 to 12 the power increases significantly. Beyond compression ratio 12, when the compression ratio is further increased to 14, there is no significant increase in power output. This can be seen from Figure 5.9.
Figure 5.9 The indicated power vs. compression ratio for various equivalence ratios in the CFR engine with landfill gas operation at spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.10 The indicated power vs. spark timing BTDC for various equivalence ratios in the CFR engine with landfill gas operation at compression ratio 12, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
The Figure 5.10 shows the variation of power with spark timing at compression ratio 12. It can be seen that the maximum power occurs at spark timing $25^\circ$ BTDC. This can be attributed to the slower combustion process of the landfill gas as compared to the methane. This shows that an increased spark advance is necessary for the optimum landfill gas operation. The MBT spark timing for landfill gas was $25^\circ$ BTDC at compression ratio 12 in this engine.
5.3 Comparison of landfill gas to methane performances

The in cylinder pressure data with respect to the crank angle is the most important data collected in the experiments. The in cylinder pressure data can be used to calculate all other characteristics like indicated thermal efficiency, combustion duration and so on. Analysis of the experimental data gives useful information about the combustion processes of the landfill gas. The data was analyzed to generate the following characteristics.

1. Power
2. Exhaust gas temperature
3. Indicated thermal efficiency
4. Ignition lag
5. Combustion duration
6. Peak pressure
7. COV of power
8. COV of ignition lag
9. COV of combustion duration
10. COV of peak pressure

Indicated power is given by,

\[
\text{Power} := \frac{W}{t_s}
\]  \hspace{1cm} (5.1)
where $W$ is the work done per cycle given by,

$$W := \int P \, dV \quad (5.2)$$

and $t_s$ is the time taken per engine cycle. In this study the CFR engine speed was 600 rpm so the $t_s$ was 0.2 sec per engine cycle. $P$ is the instantaneous in cylinder pressure and $dV$ is the corresponding change in volume.

Indicated thermal efficiency is given by,

$$\eta_i := \frac{\text{Power}}{a_c \cdot m_f} \quad (5.3)$$

where $a_c$ is the energy content of the fuel and $m_f$ is the mass flowrate of the fuel.

The combustion process within the combustion chamber varies widely from cycle to cycle. These cyclic variations affect the performance and other combustion characteristics significantly. In order to study these cyclic variations 100 consecutive engine cycles were recorded in each engine run and the average pressure of these 100 engine cycles were calculated. Coefficient of variation is a statistical term to measure the variation of a characteristic in a given set of values. It is defined as,

$$\text{COV} := \frac{S_d}{\text{mean}} \quad (5.4)$$
where $S_d$ is the standard deviation and mean is the average of all the values of the characteristic. Sometimes COV is expressed as a percentage. In this study COV has been studied to find out the cyclic variations in a given engine run. The coefficients of variation of different characteristics are studied from the 100 cycles data recorded for each engine run.

The methods to calculate combustion duration and ignition lag have been explained in the section 4.6.1. The exhaust temperature was recorded for each engine run.

Since the landfill gas is composed of approximately 50% methane it is necessary to compare the landfill gas performance with the baseline experiments done with methane. The heating value of the landfill gas is almost only 50% of that of methane.

The Figure 5.11 shows the performance comparison of methane and landfill gas. The spark timings are the corresponding MBT spark timings of the gases. For methane it is 12° BTDC and for landfill gas it is 25° BTDC at compression ratio 8.5. There is a significant reduction in power output of the landfill gas as compared to that of methane and this reduction in power is about 5.8% at equivalence ratio of 1.0. It can be further seen that as the compression ratio is increased to 12 in the case of landfill gas, the power output significantly increases which is comparable to the power output of the methane at compression ratio 8.5. This indicates that the presence of diluents
Figure 5.11 The comparison of indicated power vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratios 8.5 and 12, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.12 The comparison of indicated power vs. spark timing BTDC for equivalence ratio 1.0 in the CFR engine with methane and landfill gas operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
in the landfill gas acts as a knock suppressant and allows for higher compression ratio operation which may not be possible with methane.

Similarly the MBT spark timings of compression ratio 8.5 for methane and landfill gas are compared in the Figure 5.12. For compression ratio of 8.5 and equivalence ratio of 1.0, it can be seen that the maximum power for methane is occurring at spark timing 12° BTDC and for landfill gas the maximum power is occurring at spark timing 25 BTDC. This shows that the landfill gas requires higher spark advance set to produce maximum power as compared to the spark advance required for methane. This indicates that the rate of combustion process is slower in the landfill gas as compared to that of methane and an increase in spark advance is required to increase the power output of the landfill gas operation.

These two observations are important in the case of modifications to be done for a natural gas run engine to run it in landfill gas. In such situations it becomes necessary to increase the compression ratio of the engine and increased spark advance to improve the performance of the engine with landfill gas. The heating value of landfill gas is only 50% of methane, but if the proper modifications are done in the engine then the performance of landfill gas will be very well comparable to that of methane.

The Figures 5.13 and 5.14 shows the power output for the maximum, minimum and average landfill gas composition operation of the CFR engine
Figure 5.13 The comparison of indicated power vs. equivalence ratio in the CFR engine with various landfill gas compositions’ operation at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.14 The comparison of indicated power vs. equivalence ratio in the CFR engine with various landfill gas compositions’ operation at compression ratio 12, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
at compression ratios 8.5 and 12 respectively for their corresponding MBT spark timings. It can be observed that the methane percentage increase in the landfill gas increases the power output and the rich limit operation is also increased from equivalence ratio 1.1 to 1.2. In Figure 5.14 it can be seen that the average landfill gas operation is producing a higher power than the maximum landfill gas composition operation at equivalence ratio 0.6. It can be attributed to the higher thermal efficiency at the average landfill gas operation than the maximum landfill gas operation. The exhaust gas temperature was 350 C for the average landfill gas operation as compared to 465 C for the maximum landfill gas operation which is an indication of lesser heat lost to the exhaust gases and a higher thermal efficiency. This indicates that because of the higher thermal efficiency the landfill gas operation will yield better performances comparable to methane operation (Figure 5.11) even though the heating value of the landfill gas is only around 50% of that of methane.

This is further confirmed from Figure 5.15 where the landfill gas operation produces more power than the methane operation at compression ratio 8.5 and spark timing 30 BTDC. This is because the MBT spark timing of methane operation for compression ratio 8.5 was 12 BTDC and retarding the spark timing to 30 BTDC is producing far lesser power. The increase in
Figure 5.15 The comparison of indicated power vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.16 The comparison of COV of power vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 5.17 The comparison of exhaust gas temperature vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.18 The comparison of indicated thermal efficiency vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
thermal efficiency with landfill gas operation is shown in Figure 5.18. This increase in thermal efficiency can be further confirmed from the lower exhaust gas temperature of landfill gas operation shown in Figure 5.17. It can be clearly seen that in the rich limit operation the thermal efficiency is lowered for landfill gas operation and there is a drop in power output as seen in Figure 5.15. Figure 5.16 shows the coefficient of variation of power which indicates that the variation in landfill gas operation is more pronounced compared to the methane operation. It can be further seen that the rich limit operation of landfill gas and methane have higher cyclic variations than the lean and stoichiometric operations.

The ignition lag and combustion duration are increased in the landfill gas operation as compared to methane operation. This can be mainly attributed to the flame temperature reduction caused by the diluents present in the landfill gas. Figure 5.19 shows the ignition lag and Figure 5.20 shows the COV of ignition lag for landfill gas and methane operation at compression ratio 8.5 and spark timing 30 BTDC. The COV of ignition lag was found to be increased by 250% for landfill gas operation at the equivalence ratio of 0.8. Similarly the combustion duration and its COV are shown in Figure 5.21 and 5.22 respectively and it was observed that the combustion duration increased by 13.7% and the COV by 142% at equivalence ratio of 0.8.
Figure 5.19 The comparison of ignition lag vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.20 The comparison of COV of ignition lag vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 5.21 The comparison of combustion duration vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.22 The comparison of COV of combustion duration vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 5.23 The comparison of peak pressure vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 5.24 The comparison of COV of peak pressure vs. equivalence ratio in the CFR engine with methane and landfill gas operation at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
The peak pressure and COV of peak pressure are shown in Figure 5.23 and 5.24 for compression ratio of 8.5 and spark timing 30 BTDC. The peak pressure dropped significantly with the landfill gas operation which was around 27% decrease from methane operation at equivalence ratio 1.0. The combustion characteristics variation significantly increases with landfill gas operation which can be attributed to the diluents present in them reducing the flame temperature in the combustion process.

5.4 Summary

Baseline experiments were conducted with methane as fuel. Landfill gas experiments were conducted and the results were compared with the methane operation. The optimum process parameters for landfill gas operation were discussed. The minimum and maximum landfill gas experiments were conducted in addition to the average landfill gas operation. The combustion characteristics of the landfill gas operation were compared with the baseline methane operation.
CHAPTER 6

THE EFFECTS OF ADDING SMALL AMOUNTS OF HYDROGEN TO LANDFILL GAS FUEL

6.1 Data analysis of landfill gas with hydrogen mixtures

In this set of experiments landfill gas was tested along with small amounts of hydrogen. Hydrogen was added with landfill gas at 3%, 5%, 10%, 20% and 30%. The percentages show the percentage of hydrogen in the landfill gas and hydrogen mixture. So 3% hydrogen means the remaining 97% is landfill gas with the composition of the landfill gas still maintained at the average landfill gas composition of 53% methane, 42% carbon dioxide and 5% nitrogen.

The pressure data collected from these experiments were further analyzed to get the engine performance data and combustion characteristics data. The characteristics studied to understand the effect of adding small amounts of hydrogen to landfill gas fuel were indicated power, indicated thermal efficiency, peak cylinder pressure, exhaust gas temperature, ignition lag and combustion duration. Since 100 pressure cycles were recorded in each engine run the cyclic variations of these characteristics were quantified by calculating the coefficient of variation for the corresponding characteristics.

In order to understand the effect of hydrogen, the power output is plotted against the percentage of hydrogen in the fuel for varying
equivalence ratios and for a given compression ratio and a given spark timing. Figure 6.1 shows one such plot for compression ratio 8.5 and spark timing $12^\circ$ BTDC. It can be seen that for equivalence ratio 1.0 and 0.8, the power output is almost constant and there is no significant increase or decrease. This can be attributed to the fact that the molar heating value of hydrogen is far less compared to the molar heating value of methane, which means by replacing a certain volume of landfill gas with hydrogen, the heating value of the fuel is decreased. But there are very significant changes in the equivalence ratios 0.6 and 1.1 as the hydrogen percentage of the fuel increases attributed to the better combustion enhanced by hydrogen.

At equivalence ratio 0.6 for compression ratio 8.5 and spark timing $12^\circ$ BTDC, the landfill gas did not burn and so the power is zero. Whereas when 3% hydrogen was added to the landfill gas, the mixture was very well ignitable and produced a power of almost 1.5 kW. As the hydrogen percentage increases in the fuel there is a slight increase in the power. This shows that hydrogen even in very little quantity like 3% increases the ignitability of the landfill gas and increases the lean limit. At equivalence ratio 1.1 the landfill gas did not perform well as it produced far less power than stoichiometric combustion. But the addition of hydrogen steadily
Figure 6.1 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.2 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.3  The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.4  The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.5  The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.6  The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.7  The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.8  The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.9 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.10 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.11 The indicated power vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.12 The COV of power vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
increases the power output till 30% hydrogen in the fuel. This can again be attributed to the better combustion characteristics induced by hydrogen. The Figure 6.2 shows the coefficient of variation of power for the corresponding engine runs at compression ratio 8.5 and spark timing 12 BTDC. It can be seen that the COV was very high at the rich equivalence ratio of 1.1 but this variation was decreased as the hydrogen percentage was increased.

The similar set of experiments was conducted at the spark timing 25° BTDC which is the MBT spark timing of the landfill gas at compression ratio 8.5. The Figure 6.3 shows that the equivalence ratio 1.0 and 0.8 did not have any significant power improvement for the entire range of hydrogen addition. But the equivalence ratio 0.6 shows a significant improvement in the power output for 3% of hydrogen addition in the fuel. But as the hydrogen percentage is increased in the fuel there is no significant improvement in power. Similarly for the equivalence ratio 1.1 the performance increases significantly after 20% hydrogen in fuel. This further confirms that hydrogen helps better combustion at lean and rich limits and increases the operational limits of landfill gas. The Figure 6.4 shows the COV of power for the corresponding parameters which confirms that the cyclic variations are significantly reduced by hydrogen addition in the lean and rich limits. The reduction in COV in the lean limit of equivalence ratio 0.6 was found to be 96%.
The Figure 6.5 shows the experiments conducted at compression ratio 8.5 and an increased spark advance of 30° BTDC. A similar trend of performance as observed in the spark timings 12 and 25 BTDC was repeated. The equivalence ratios 1.0 and 0.8 did not increase in power as the hydrogen percentage was increased. But the equivalence ratio 0.6 showed a drastic improvement in power by adding 3% hydrogen in the fuel from the point where it did not burn at all for landfill gas. This shows that the lean mixture’s ignitability is improved by the presence of hydrogen and improves the power output significantly. The COV of power shown in Figure 6.6 shows the cyclic variation of the rich mixture at equivalence ratio 1.1 was gradually reduced as hydrogen was added.

The spark timing 40° BTDC showed a slight difference in performance from other spark timings. From Figure 6.7 it can be seen that the equivalence ratios 1.0 and 0.6 increased a slight amount of power for 3% and 5% of hydrogen in fuel, but beyond 10% the power can be seen deteriorating slightly. This is can be again attributed to the lower molar heating value of hydrogen compared to that of methane or landfill gas. The equivalence ratio 1.1 shows a steady increase in power till 20% hydrogen and deteriorates there after. This can be attributed for the improved ignitability of the rich mixture because of the presence of hydrogen which otherwise produces very little power. This shows that hydrogen increases the rich limit of the fuel. The COV
of power is reduced significantly in the lean and rich limits as shown in Figure 6.8.

Similar experiments were conducted at higher compression ratios like 12 and 14. These higher compression ratio experiments with landfill gas and hydrogen were conducted with spark timing 25° BTDC. From Figure 6.9 it can be seen that the compression ratio 12 showed almost a similar trend like the results in compression ratio 8.5. The equivalence ratios 1.0 and 0.8 show a stable curve with no increase in power because of the hydrogen addition. The equivalence ratio 0.6 shows a significant increase in power as shown always in the compression ratio 8.5. But the equivalence ratio 1.1 did not show any significant change in the performance as the hydrogen addition is increased.

The Figure 6.11 shows that the compression ratio 14 shows a slightly different trend in the power curves as opposed to the curves in compression ratio 12. The significant difference is that both the equivalence ratio 1.0 and 0.8 shows a deteriorating trend as the hydrogen percentage is increased. Equivalence ratio 0.6 shows a very similar trend as seen in the previous plots. The equivalence ratio 1.2 did burn well and produced power. This was not the case with other compression ratios 8.5 and 12 where the rich limit equivalence ratio 1.2 never burned and produced power. This equivalence ratio 1.2 shows a steady increase in power as the hydrogen percentage is increased.
It can be seen from these performance curves for different compression ratios of 8.5, 12 and 14 that the hydrogen addition is in fact improving the performance in the lean and rich limits of landfill gas. In stoichiometric combustion and near stoichiometric, hydrogen addition did not do any significant improvement in power. Though there were significant improvements in the lean and rich limits, the improvement was more significant in the lean side. This is because in the lean limits the performance increased significantly even with the addition of a very little amount of hydrogen which is around 3% in the fuel. In rich limits there was a significant improvement in performance but this significant improvement was possible only with higher amounts of hydrogen addition to the fuel which is more than 10% to 20% in most cases and this improved power did not exceed the stoichiometric performance. Because of the cost involved in using hydrogen as an additive it can be said that hydrogen enhanced combustion of landfill gas is more attractive in the lean combustion of landfill gas as very small percentage of hydrogen is required to take advantage of the benefits of hydrogen. The corresponding COV of power are shown in Figures 6.10 and 6.12 which further indicates that the cyclic variations are reduced significantly in the lean and rich limit operations.
Figure 6.13 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.14 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.15 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.16 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.17 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.18 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.19 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.20 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.21 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.22 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.23 The average indicated thermal efficiency vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.24 The exhaust gas temperature vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Thermal efficiency is basically the measure of the energy contained in the fuel getting converted to useful power. Generally in internal combustion engines $1/3^{rd}$ of the energy is lost in heat transfer and $1/3^{rd}$ of the energy is lost to the exhaust gases. Only the remaining energy is converted as useful power. Moreover thermal efficiency is an indication of specific fuel consumption (sfc). A higher thermal efficiency means less fuel consumption for the same power. If there is better combustion improving the thermal efficiency, then the heat lost to the exhaust gas is reduced which reduces the exhaust gas temperatures. If the combustion is poor then also the exhaust gas temperature will drop down. This makes it necessary to study the thermal efficiency and the exhaust gas temperature in parallel.

The addition of hydrogen to landfill gas fuel improves the indicated thermal efficiency significantly in the very lean limit of equivalence ratio 0.6. At compression ratio 8.5 and spark timing 25 BTDC the thermal efficiency improved significantly in all the equivalence ratios as shown in Figure 6.13. The corresponding exhaust gas temperature is shown in Figure 6.14. In the equivalence ratios 0.6 to 1.0 there is a reduction in the exhaust gas temperature which corresponds to the increase in thermal efficiency. At equivalence ratio 1.1 the exhaust gas temperature increases as the hydrogen percentage is increased which can be attributed to the better combustion of
the rich mixture by the addition of hydrogen showing an increase in thermal efficiency.

At compression ratio 8.5 and spark timing 25 BTDC a drastic improvement in the indicated thermal efficiency was noted at the very lean limit equivalence ratio 0.6 as shown in Figure 6.15. The corresponding exhaust gas temperature is shown in Figure 6.16. At the equivalence ratio 0.6 it can be seen that the exhaust gas temperature increases significantly and a corresponding increase in thermal efficiency is noticed which suggests that the combustion was improved by the addition of hydrogen. In other equivalence ratio there is a reduction in thermal efficiency which can be attributed to the reduction in the heating value of the landfill gas and hydrogen mixture and the corresponding slight reduction in power.

At compression ratio 8.5 and spark timings 30 and 40 BTDC, the 3% and 5% addition of hydrogen increased the indicated thermal efficiency in the equivalence ratios 0.6, 0.8 and 1.0 shown in Figures 6.17 and 6.19. The corresponding exhaust gas temperatures are shown in Figures 6.18 and 6.20.

At higher compression ratios 12 and 14 and spark timing 25 BTDC, a significant increase in indicated thermal efficiency was noticed at the very lean limit equivalence ratio 0.6 which was around 15% increase shown in Figures 6.21 and 6.23. The corresponding exhaust gas temperatures are shown in Figures 6.22 and 6.24.
Figure 6.25 The indicated power vs. spark timing BTDC for various equivalence ratios at a hydrogen percentage of 5% in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

The indicated power variation with spark timing BTDC is shown in the Figure 6.25 shows that the spark timing around 25 BTDC gives the maximum indicated power for equivalence ratios 1.0 and 0.6 with the hydrogen percentage in the fuel being 5%.

The peak cylinder pressure is significantly improved in all conditions with the addition of hydrogen. This can be attributed to the better combustibility and faster burning rates induced by the hydrogen additive. At compression ratio 8.5 and spark timing 25 BTDC the addition of 30% hydrogen increases the peak cylinder pressure by 25% at the equivalence
Figure 6.26 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.27 The COV of average peak cylinder pressure vs. hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine for various equivalence ratios at compression ratio 8.5, spark timing 12 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.28 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.29 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.30 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.31 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 30 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.32 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.33 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 40 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.34 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.35 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.36 The average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.37 The COV of average peak cylinder pressure vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
ratio 0.8 shown in Figure 6.28. The corresponding COV of peak cylinder pressure is given in Figure 6.29 which shows that the COV is significantly reduced by the addition of hydrogen. It can be noticed that the variation was very high at the rich limit operation. The variation in the lean limit operation was significantly reduced by the addition of hydrogen at equivalence ratio 0.6. The reduction in COV was 70% even with the addition of hydrogen as little as 3% with landfill gas. The Figure 6.27 shows the significant reduction in COV of peak cylinder pressure as the hydrogen percentage is increased in the landfill gas for compression ratio 8.5 and spark timing 12 BTDC. Almost a very similar trend of peak cylinder pressure was observed in other spark timings of the compression ratio 8.5 and higher compression ratios of 12 and 14. The corresponding peak cylinder pressure and COV are shown in Figures 6.26 to 6.37.

The addition of hydrogen to landfill gas has a very significant effect in reducing the ignition lag and the combustion duration. At compression ratio 8.5 and spark timing 25 BTDC shown in Figure 6.38 there is a significant reduction in the ignition lag period as the hydrogen addition is increased. The corresponding COV of ignition lag is shown in Figure 6.39. In the very lean limit equivalence ratio 0.6 this reduction was very significant that the addition of even very little hydrogen of 3% to landfill gas reduced the ignition lag by 52% and the COV was reduced by 82%. A similar trend of
ignition lag reduction was observed for compression ratios 12 and 14 at spark timing 25 BTDC as shown in Figure 6.40 and 6.42. The corresponding COV of ignition lags are shown in Figures 6.41 and 6.43.

For landfill gas the combustion duration is too high in the lean and rich mixture operation. Longer combustion duration causes the combustion process to extend till the exhaust valve is opened which causes more heat lost to the exhaust gases rather than converting it to useful work. Hence longer combustion duration is undesirable. It can be seen that the addition of hydrogen even as little as 3% causes the combustion duration to be reduced significantly in the lean mixture operation of equivalence ratio of 0.6 at the compression ratio 8.5 and spark timing 25 BTDC (shown in Figure 6.44). This reduction in combustion duration is around 41%. The COV of combustion duration is given in Figure 6.45 and it can be seen that the COV is significantly reduced in the lean and rich limit operations. At compression ratio 12 and spark timing 25 BTDC the COV of combustion duration was reduced by 67% with the addition of 3% hydrogen (Figure 6.47). The combustion duration vs. equivalence ratios for various hydrogen percentages are shown in the Figure 6.46. A similar trend was observed in the compression ratio 14 and spark timing 25 BTDC shown in Figures 6.48 and 6.49.
Figure 6.38 The average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.39 The COV of average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.40 The average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.41 The COV of average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.42 The average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.43 The COV of average ignition lag vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.44 The average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.45 The COV of average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 8.5, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.46 The average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.47 The COV of average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.48 The average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.49 The COV of average combustion duration vs. equivalence ratio for various hydrogen percentage in landfill gas and hydrogen mixture as fuel in the CFR engine at compression ratio 14, spark timing 25 BTDC, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Figure 6.50 The pressure vs. crank angle showing the cyclic variations of using landfill gas as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.51 The pressure vs. crank angle showing the cyclic variations of using landfill gas with 3% hydrogen additive as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
The Figures 6.50 and 6.51 show the cyclic variations in the pressure curve for 100 cycles of landfill gas operation with and without hydrogen as additive. In the landfill gas operation the cyclic variations are extremely high. It can be seen that the cyclic variations are significantly reduced by the introduction of just 3% hydrogen additive. Though there are cyclic variations in the peak cylinder pressure, the variations are significantly reduced in the later phase of combustion after the peak pressure.

### 6.2 Economics of hydrogen usage as an additive for landfill gas

Hydrogen is the most abundant element in the universe. Despite this fact hydrogen is not freely available. The most important source of hydrogen is water. Other than water hydrogen is present in natural gas and other petroleum gases along with carbon. The natural gas steam reforming process is the most widely used method for hydrogen production in industrial scales. In this process a high temperature steam is used with the natural gas to produce hydrogen. This natural gas steam reforming process accounts for more than 90% of the world’s hydrogen production. Though it’s a cheaper process compared to other processes it is a fossil fuel dependent one which makes it a non-renewable source.

Hydrogen can also be produced from the electrolytic splitting of water by providing electricity. The main disadvantage of this process is the
requirement of electricity. Because of the high cost of electricity this method is not an attractive one always. Only 4% of the world’s hydrogen is produced using electrolysis of water. However because of the availability of water and the simplicity in the process, if cheap electricity is available then hydrogen production by electrolysis is an attractive option. Moreover its an attractive option of on board hydrogen production in an automobile.

Any new technology will be successful only if it is cost competitive. In this study of hydrogen used as an additive with landfill gas it becomes necessary to study the economics of hydrogen usage. One of the important observations from the study is that the hydrogen addition to the landfill gas is more advantageous with lower quantities of hydrogen than higher quantities of hydrogen. It can be seen from the Figures 6.9 and 6.11 that the power curve rises up initially as 3% hydrogen is added to the landfill gas and then remains almost constant for very lean limit of equivalence ratio 0.6. Addition of hydrogen beyond 3% or 5% does not increase the power to significant levels. Moreover this increase in power is prominent in the lean mixture operation as shown in Figure 6.9 for equivalence ratio 0.6.

At compression ratio 12, spark timing 25° BTDC and equivalence ratio 0.6, there were some important observations to be made. These are shown in Figures 6.52 and 6.53. The landfill gas operation and 3% hydrogen addition to landfill gas operation are compared. There was a 12.5% increase in power and
15.07% increase in thermal efficiency. Any increase in thermal efficiency can be viewed as increased fuel economy. The improved performance and improved fuel economy is a good justification for using hydrogen with landfill gas as an additive.

If the hydrogen used along with the landfill gas is assumed to be produced from electrolysis of water then the energy gain from the hydrogen usage and the energy required to produce that hydrogen can be compared.

For comparison purpose the efficiency factors were assumed based upon industry standards [27, 28].

Additional power got by adding hydrogen = 0.19 kW

Hydrogen flow rate = 0.53 SLPM

Additional mechanical energy got by adding hydrogen

\[
= 0.19 \times 1 \text{ hr} / (0.53\times60/1000)
= 5.96 \text{ kW-hr/m}^3 \text{ of hydrogen}
\]

At generator efficiency of 85% [27], the electrical energy generated

\[
= 5.966 \times 0.85
= 5.07 \text{ kW-hr/m}^3
\]

Electrical energy required for producing hydrogen on industrial scales with 70% efficiency [28]

\[
= 3.91 \text{ kW-hr/m}^3
\]
Figure 6.52 The increase in indicated power by adding 3% of hydrogen to the landfill gas as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

Figure 6.53 The increase in indicated thermal efficiency by 3% of hydrogen to the landfill gas as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.
Based on the above calculations it can be seen that the additional energy gained from using hydrogen as an additive with landfill gas is more than the energy required to produce that hydrogen from the electrolysis process. The Figure 6.54 shows that this energy gain which is 26.69%.

![Energy Gain Diagram]

Figure 6.54 The energy gain by adding 3% of hydrogen to the landfill gas as fuel in the CFR engine at compression ratio 12, spark timing 25 BTDC, equivalence ratio 0.6, intake temperature 303 K, intake pressure 98 kPa and 600 rpm.

6.3 Summary

The adversities caused by the diluents in the landfill gas affect the combustion process significantly. The addition of small amounts of hydrogen to landfill gas fuel improves the combustion of landfill gas in many different ways and reduces the adversities caused by the diluents to significant levels. The combustion of landfill gas in the very lean limit operation of equivalence
ratio 0.6 was drastically improved by the addition of very little amounts of hydrogen and it was found to be economical to use hydrogen as an additive for landfill gas fuel at these conditions. Since the combustion process is improved the emission levels are expected to reduce significantly.
CHAPTER 7

CONCLUSIONS AND RECOMMENDATIONS

The Alternative fuels research laboratory was established to investigate the performance of the SI engine fuelled with alternative gaseous fuels. The fuel control panel was fabricated and the data acquisition systems were developed. All data acquisition VIs were tested and proved for their functions. This engine setup provides the possibility to use different mixtures of gases in different proportions as a fuel. The established setup was further validated and proved for its correctness by conducting setup validation experiments. Engine experiments were conducted by varying different parameters such as the compression ratio, spark timing and equivalence ratio. With this kind of engine setup it was possible to test different kinds of gaseous fuels like landfill gas, biogas, gaseous mixtures containing hydrogen etc.

Baseline experiments were conducted with methane. The average composition of landfill gas was tested in different compression ratios, spark timings and equivalence ratios. Increasing the compression ratio improves the power output of the engine with landfill gas operation. Increasing the spark advance also improves the power output of the landfill gas fuelled engine. It was observed that the compression ratio 12, spark advance of 25°
BTDC and stoichiometric mixture operation of landfill gas provided an improved performance which was comparable with the methane operation at compression ratio 8.5.

The landfill gas was tested with small amounts of hydrogen. Adding hydrogen even in very little quantities improved the combustion process of landfill gas. Hydrogen addition to landfill gas increases the lean and rich operational limit of the landfill gas. The very high cyclic variations in the landfill gas operation were drastically reduced by very small quantities of hydrogen. Hydrogen addition causes significant improvement in the power output in the very lean mixture operation of equivalence ratio 0.6. Combustion duration was drastically reduced by the addition of hydrogen. The combustion characteristics improved significantly and the cyclic variations were reduced by the addition of small amounts of hydrogen to very lean limit operation of landfill gas. Because of the improved combustion the emissions can be expected to be relatively less than the landfill gas operation. Addition of hydrogen beyond 3% or 5% did not have significant effects in improving the performance of landfill gas.

The economics of using hydrogen with landfill gas was studied. At compression ratio 12, spark timing 25° BTDC and equivalence ratio 0.6 addition of 3% hydrogen showed significant improvements in performance. The power and thermal efficiency increased by 12.5% and 15.07%
respectively. Improved thermal efficiency is a measure of improved fuel economy. At this operational parameters and percentage of hydrogen used the energy gain obtained from the use of hydrogen was more than the energy required to produce that same amount of hydrogen.

A computer model was developed to simulate the combustion process of landfill gas in the engine. Empirical relations were developed for the estimation of the combustion duration of landfill gas and its hydrogen mixtures. The model was proved by comparing the modeling results with the corresponding experimental results. This model can be utilized to predict the performance of the engine by varying the parameters like compression ratio, spark timing and equivalence ratio.

The research work can be further extended to study the performance of other compositions of landfill gases and other biogases. Biogases mainly consist of methane and carbon dioxide. The same work can be extended to biogases and the effects of hydrogen addition to biogases can be studied. Moreover the study can be further extended for higher engine speeds.


17. http://msw.cecs.ucf.edu/


Appendix A

Partial differential equations used in the equilibrium combustion products module
Dependent variables

\[ y_7 := k_7 (y_6)^{0.5} \cdot p^{-0.5} \]

\[ c_7 := \frac{k_7}{p^{0.5}} \]

\[ y_8 := k_8 (y_4)^{0.5} \cdot p^{-0.5} \]

\[ c_8 := \frac{k_8}{p^{0.5}} \]

\[ y_9 := k_9 (y_6)^{0.5} (y_4)^{0.5} \]

\[ c_9 := k_9 \]

\[ y_{10} := k_{10} (y_4)^{0.5} (y_3)^{0.5} \]

\[ c_{10} := k_{10} \]

\[ y_2 := k_2 y_6 (y_4)^{0.5} \cdot p^{0.5} \]

\[ c_2 := k_2 p^{0.5} \]

\[ y_1 := k_1 y_5 (y_4)^{0.5} \cdot p^{0.5} \]

\[ c_1 := k_1 p^{0.5} \]

\[ y_{11} := k_{11} (y_3)^{0.5} \cdot p^{-0.5} \]

\[ c_{11} := \frac{k_{11}}{p^{0.5}} \]

\[ y_{12} := k_{12} y_{10} (y_4)^{0.5} \cdot p^{0.5} \]

\[ c_{12} := k_{12} p^{0.5} \cdot k_{10} \]

\[ y_{13} := k_{13} \frac{(y_6)^4 y_1}{(y_2)^2} \cdot p^{2} \]

\[ c_{13} := k_{13} p^{1.5} \cdot \frac{k_1}{(k_2)^2} \]
In the matlab program codes

\[ x := y_3 \]

\[ y := y_4 \]

\[ p := y_5 \]

\[ q := y_6 \]

4 non linear equations

\[ y_1 + y_2 + y_3 + y_4 + y_5 + y_6 + y_7 + y_8 + y_9 + y_{10} + y_{11} + y_{12} + y_{13} - 1 := 0 \quad \leftarrow f_1 \]

\[ (2y_2 + 2y_6 + y_7 + y_9 + 4y_{13}) - cc1 \cdot (y_1 + y_5 + y_{13}) := 0 \quad \leftarrow f_2 \]

\[ (2y_1 + y_2 + 2y_4 + y_5 + y_8 + y_9 + y_{10} + 2y_{12}) - cc2 \cdot (y_1 + y_5 + y_{13}) := 0 \quad \leftarrow f_3 \]

\[ (2y_3 + y_{10} + y_{11} + y_{12}) - cc3 \cdot (y_1 + y_5 + y_{13}) := 0 \quad \leftarrow f_4. \]
Non zero derivative

\[ D_{76} = k_7 p^{0.5} (y_6)^{0.5} \]

\[ D_{84} = k_8 p^{0.5} (y_4)^{0.5} \]

\[ D_{94} = k_9 (y_6)^{0.5} (y_4)^{-0.5} \]

\[ D_{96} = k_9 (y_4)^{0.5} (y_6)^{-0.5} \]

\[ D_{104} = k_{10} (y_3)^{0.5} (y_4)^{0.5} \]

\[ D_{103} = k_{10} (y_4)^{0.5} (y_3)^{0.5} \]

\[ D_{24} = k_2 y_6 p^{0.5} (y_4)^{-0.5} \]

\[ D_{26} = k_2 (y_4)^{0.5} p^{0.5} \]

\[ D_{14} = k_1 y_5 p^{0.5} (y_4)^{-0.5} \]

\[ D_{15} = k_1 (y_4)^{0.5} p^{0.5} \]

\[ D_{113} = k_{11} p^{-0.5} (y_3)^{0.5} \]

\[ D_{123} = k_{12} k_{10} y_4 p^{0.5} (y_3)^{0.5} \]

\[ D_{124} = k_{12} k_{10} (y_3)^{0.5} p^{0.5} \]

\[ D_{134} = \frac{k_{13}}{(k_2)^2} (y_6)^2 k_1 y_5 p^{1.5} (y_4)^{-0.5} \]

\[ D_{135} = \frac{k_{13}}{(k_2)^2} (y_6)^2 k_1 (y_4)^{0.5} p^{1.5} \]

\[ D_{136} = \frac{k_{13}}{(k_2)^2} k_1 (y_4)^{-0.5} y_5 p^{1.5} y_6 \]
Matrix equations for mole fractions solving

$$A_{11} := \frac{\partial}{\partial y_3} f_1 := 1 + D_{103} + D_{113} + D_{123}$$

$$A_{12} := \frac{\partial}{\partial y_4} f_1 := D_{14} + D_{24} + 1 + D_{84} + D_{94} + D_{104} + D_{124} + D_{134}$$

$$A_{13} := \frac{\partial}{\partial y_5} f_1 := D_{15} + 1 + D_{135}$$

$$A_{14} := \frac{\partial}{\partial y_6} f_1 := D_{26} + 1 + D_{76} + D_{96} + D_{136}$$

$$A_{21} := \frac{\partial}{\partial y_3} f_2 := 0$$

$$A_{22} := \frac{\partial}{\partial y_4} f_2 := 2D_{24} + D_{94} + 4D_{134} - c c 1 \cdot (D_{134} + D_{14})$$

$$A_{23} := \frac{\partial}{\partial y_5} f_2 := 4D_{135} - c c 1 \cdot (1 + D_{15} + D_{135})$$

$$A_{24} := \frac{\partial}{\partial y_6} f_2 := 2D_{26} + 2 + D_{76} + D_{96} + 4D_{136} - c c 1 \cdot D_{136}.$$
solution of this matrix equations gives $\Delta y_i$ required.

$$A_{31} := \frac{\partial f_3}{\partial y_3} \quad := 2D_{103} + 2D_{123}$$

$$A_{32} := \frac{\partial f_3}{\partial y_4} \quad := 2D_{14} + D_{24} + 2 + D_{84} + D_{94} + D_{104} + 2D_{124} - cc2 \left( D_{14} + D_{134} \right)$$

$$A_{33} := \frac{\partial f_3}{\partial y_5} \quad := 2D_{15} + 1 - cc2 \left( D_{15} + 1 + D_{135} \right)$$

$$A_{34} := \frac{\partial f_3}{\partial y_6} \quad := D_{26} + D_{96} - cc2 \cdot D_{136}$$

$$A_{41} := \frac{\partial f_4}{\partial y_3} \quad := 2 + D_{103} + D_{113} + D_{123}$$

$$A_{42} := \frac{\partial f_4}{\partial y_4} \quad := D_{104} + D_{124} - cc3 \left( D_{14} + D_{134} \right)$$

$$A_{43} := \frac{\partial f_4}{\partial y_5} \quad := -cc3 \left( 1 + D_{135} + D_{15} \right)$$

$$A_{44} := \frac{\partial f_4}{\partial y_6} \quad := -cc3 \cdot D_{136}$$

$[A] [\Delta y] + [f] = 0$ solution of this matrix equations gives $\Delta y_i$ required.
Partial derivatives of mole fractions w.r.t $T$

\[
\frac{\partial}{\partial T} y_7 := \left(y_6\right)^2 \frac{\partial}{\partial T} c_7 + D_{76} \left(\frac{\partial}{\partial T} y_6\right)
\]

\[
\frac{\partial}{\partial T} y_8 := \left(y_4\right)^2 \frac{\partial}{\partial T} c_8 + D_{84} \left(\frac{\partial}{\partial T} y_4\right)
\]

\[
\frac{d}{dT} y_9 := \left(y_4\right)^2 \left(y_6\right)^2 \frac{1}{d} \frac{d}{dT} c_9 + D_{96} \left(\frac{\partial}{\partial T} y_6\right) + D_{94} \left(\frac{\partial}{\partial T} y_4\right)
\]

\[
\frac{\partial}{\partial T} y_{10} := \left(y_4\right)^2 \left(y_3\right)^2 + D_{104} \left(\frac{\partial}{\partial T} y_4\right) + D_{103} \left(\frac{\partial}{\partial T} y_3\right)
\]

\[
\frac{\partial}{\partial T} y_{11} := \left(y_5\right)^2 + D_{113} \left(\frac{\partial}{\partial T} y_3\right)
\]

\[
\frac{\partial}{\partial T} y_{12} := \left(y_5\right)^2 y_4 + D_{123} \left(\frac{\partial}{\partial T} y_3\right) + D_{124} \left(\frac{\partial}{\partial T} y_4\right)
\]

\[
\frac{\partial}{\partial T} y_{13} := \frac{1}{y_6 y_4} + D_{136} \left(\frac{\partial}{\partial T} y_6\right) + D_{135} \left(\frac{\partial}{\partial T} y_5\right) + D_{134} \left(\frac{\partial}{\partial T} y_4\right)
\]

\[
\frac{\partial}{\partial T} y_1 := \left(y_6\right)^2 \frac{1}{y_5 \left(y_4\right)^2} + D_{15} \left(\frac{\partial}{\partial T} y_5\right) + D_{14} \left(\frac{\partial}{\partial T} y_4\right)
\]

\[
\frac{\partial}{\partial T} y_2 := \frac{1}{y_6 \left(y_4\right)^2} + D_{26} \left(\frac{\partial}{\partial T} y_6\right) + D_{24} \left(\frac{\partial}{\partial T} y_4\right)
\]
Expressions for \( x \)

\[
x_1 := \frac{y_1}{c_1} \quad x_2 := \frac{y_2}{c_1} \quad x_3 := \frac{y_3}{c_1}
\]

\[
x_4 := \frac{y_4}{c_1} \quad x_5 := \frac{y_5}{c_1} \quad x_6 := \frac{y_6}{c_1}
\]

\[
x_7 := \frac{y_7}{c_1} \quad x_8 := \frac{y_8}{c_1} \quad x_9 := \frac{y_9}{c_1}
\]

\[
x_{10} := \frac{y_{10}}{c_{10}} \quad x_{11} := \frac{y_{11}}{c_{11}} \quad x_{12} := \frac{y_{12}}{c_{12}}
\]

\[
x_{13} := \frac{y_{13}}{c_{13}}
\]

Partial derivatives:

\[
\frac{\partial f_1}{\partial T_1} := \left( \frac{\partial}{\partial T} c_1 \right) x_1 + \left( \frac{\partial}{\partial T} c_2 \right) x_2 + \left( \frac{\partial}{\partial T} c_3 \right) x_3 + \left( \frac{\partial}{\partial T} c_4 \right) x_4 + \left( \frac{\partial}{\partial T} c_5 \right) x_5 + \left( \frac{\partial}{\partial T} c_6 \right) x_6 + \left( \frac{\partial}{\partial T} c_7 \right) x_7 + \left( \frac{\partial}{\partial T} c_8 \right) x_8 + \left( \frac{\partial}{\partial T} c_9 \right) x_9 + \left( \frac{\partial}{\partial T} c_{10} \right) x_{10} + \left( \frac{\partial}{\partial T} c_{11} \right) x_{11} + \left( \frac{\partial}{\partial T} c_{12} \right) x_{12} + \left( \frac{\partial}{\partial T} c_{13} \right) x_{13}
\]

\[
\frac{\partial f_2}{\partial T_2} := \left( \frac{\partial}{\partial T} c_1 \right) x_1^2 + \left( \frac{\partial}{\partial T} c_2 \right) x_2 + \left( \frac{\partial}{\partial T} c_3 \right) x_3 + \left( \frac{\partial}{\partial T} c_4 \right) x_4 + \left( \frac{\partial}{\partial T} c_5 \right) x_5 + \left( \frac{\partial}{\partial T} c_6 \right) x_6 + \left( \frac{\partial}{\partial T} c_7 \right) x_7 + \left( \frac{\partial}{\partial T} c_8 \right) x_8 + \left( \frac{\partial}{\partial T} c_9 \right) x_9 + \left( \frac{\partial}{\partial T} c_{10} \right) x_{10} + \left( \frac{\partial}{\partial T} c_{11} \right) x_{11} + \left( \frac{\partial}{\partial T} c_{12} \right) x_{12} + \left( \frac{\partial}{\partial T} c_{13} \right) x_{13} - cc \left[ \left( \frac{\partial}{\partial T} c_1 \right) x_1 + \left( \frac{\partial}{\partial T} c_{13} \right) x_{13} \right]
\]

\[
\frac{\partial f_3}{\partial T_3} := 2 \left( \frac{\partial}{\partial T} c_1 \right) x_1 + \left( \frac{\partial}{\partial T} c_2 \right) x_2 + \left( \frac{\partial}{\partial T} c_3 \right) x_3 + \left( \frac{\partial}{\partial T} c_4 \right) x_4 + \left( \frac{\partial}{\partial T} c_5 \right) x_5 + \left( \frac{\partial}{\partial T} c_6 \right) x_6 + \left( \frac{\partial}{\partial T} c_7 \right) x_7 + \left( \frac{\partial}{\partial T} c_8 \right) x_8 + \left( \frac{\partial}{\partial T} c_9 \right) x_9 + \left( \frac{\partial}{\partial T} c_{10} \right) x_{10} + \left( \frac{\partial}{\partial T} c_{11} \right) x_{11} + \left( \frac{\partial}{\partial T} c_{12} \right) x_{12} - cc \left[ \left( \frac{\partial}{\partial T} c_1 \right) x_1 + \left( \frac{\partial}{\partial T} c_{13} \right) x_{13} \right]
\]

\[
\frac{\partial f_4}{\partial T_4} := \left( \frac{\partial}{\partial T} c_1 \right) x_{10} + \left( \frac{\partial}{\partial T} c_{11} \right) x_{11} + \left( \frac{\partial}{\partial T} c_{12} \right) x_{12} - cc \left[ \left( \frac{\partial}{\partial T} c_1 \right) x_1 + \left( \frac{\partial}{\partial T} c_{13} \right) x_{13} \right]
\]
Partial derivative of $c$

\[
\frac{\partial}{\partial T} c_1 := \frac{1}{2} \frac{dk_1}{dt} \\
\frac{\partial}{\partial T} c_2 := P^2 \frac{dk_2}{dt} \\
\frac{\partial}{\partial T} c_7 := \frac{1}{2} \frac{dk_7}{dt} \\
\frac{\partial}{\partial T} c_8 := P \frac{dk_8}{dt} \\
\frac{\partial}{\partial T} c_9 := \frac{dk_9}{dt} \\
\frac{\partial}{\partial T} c_{10} := \frac{dk_{10}}{dt} \\
\frac{\partial}{\partial T} c_{11} := \frac{1}{2} \frac{dk_{11}}{dt} \\
\frac{\partial}{\partial T} c_{12} := P^2 \left( k_{12} \frac{dk_{10}}{dt} + k_{10} \frac{dk_{12}}{dt} \right) \\
\frac{\partial}{\partial T} c_{13} := P^{1.5} \frac{d}{dt} \left( \frac{(k_2)^2}{dt} \left( k_{13} k_1 - k_{13} k_1^2 k_2 \frac{dk_2}{dt} \right) \right) \left( k_2 \right)^4 
\]

Differentiation of $k$ w.r.t $t$ in the program temperature is taken as $t$ so $t=T$

\[
\frac{dk}{dt}_{(gas, t)} := \frac{k(gas, t + 0.25) - k(gas, t - 0.25)}{0.5}
\]

central difference method used to find the differentiation of tabled values
Partial derivatives of c w.r.t P

\[ \frac{\partial}{\partial P} c_1 := k_1 \frac{1}{2} P^{\frac{-1}{2}} \]

\[ \frac{\partial}{\partial P} c_2 := k_2 \frac{1}{2} P^{\frac{-1}{2}} \]

\[ \frac{\partial}{\partial P} c_7 := k_7 \left( -\frac{1}{2} \right) P^{-1.5} \]

\[ \frac{\partial}{\partial P} c_8 := k_8 \left( -\frac{1}{2} \right) P^{-1.5} \]

\[ \frac{\partial}{\partial P} c_9 := 0 \]

\[ \frac{\partial}{\partial P} c_{10} := 0 \]

\[ \frac{\partial}{\partial P} c_{11} := k_{11} \left( -\frac{1}{2} \right) P^{-1.5} \]

\[ \frac{\partial}{\partial P} c_{12} := k_{12} k_{10} \left( -\frac{1}{2} \right) P^{\frac{-1}{2}} \]

\[ \frac{\partial}{\partial P} c_{13} := \frac{k_{13}}{k_1 \left( k_2 \right)^2} \cdot 1.5 P^{0.5} \]
Appendix B

Matlab programs developed for combustion modeling
Hierarchy structure of the Matlab programs for combustion modeling.
Matlab programs and its functions for combustion modeling.

<table>
<thead>
<tr>
<th>No.</th>
<th>Matlab Program</th>
<th>Function of the Module</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Heat release.m</td>
<td>Main program to integrate the differential equations.</td>
</tr>
<tr>
<td>2</td>
<td>Equations.m</td>
<td>All the differential equations are given here for compression, combustion and expansion phases.</td>
</tr>
<tr>
<td>3</td>
<td>Farg.m</td>
<td>Fuel air residual gas properties.</td>
</tr>
<tr>
<td>4</td>
<td>Ferguson3.m</td>
<td>Equilibrium combustion properties.</td>
</tr>
<tr>
<td>5</td>
<td>Tinitialize.m</td>
<td>Initializing the burned gas temperature at the start of combustion phase.</td>
</tr>
<tr>
<td>6</td>
<td>Solvegaussjordan.m</td>
<td>Solves linear equations given by ferguson3.m</td>
</tr>
<tr>
<td>7</td>
<td>DKDT.m</td>
<td>Differentiation of K used at the start of farg.m</td>
</tr>
<tr>
<td>8</td>
<td>dk.m</td>
<td>Differentiation of k from JANAF table interpolated values.</td>
</tr>
<tr>
<td>9</td>
<td>exm.m</td>
<td>To calculate the mass blow by</td>
</tr>
<tr>
<td>10</td>
<td>Vol.m</td>
<td>Instantaneous volume</td>
</tr>
<tr>
<td>11</td>
<td>DV.m</td>
<td>Differentiation of volume</td>
</tr>
<tr>
<td>12</td>
<td>Pm.m</td>
<td>Motoring pressure</td>
</tr>
<tr>
<td>13</td>
<td>Cp</td>
<td>cp from JANAF table interpolation</td>
</tr>
<tr>
<td>14</td>
<td>So</td>
<td>so from JANAF table interpolation</td>
</tr>
<tr>
<td>15</td>
<td>K</td>
<td>k from JANAF table interpolation</td>
</tr>
<tr>
<td>16</td>
<td>Ho</td>
<td>ho from JANAF table interpolation</td>
</tr>
</tbody>
</table>
Matlab Programs

Heat release.m

clear;clc;
global m0 f CR phi P1 t1 thetas thetab thetaig Tw mp co2p np Const
thetacomps

global dv1
dv1=[];
step=1;
f=0.11476;CR=8.5;phi=1;P1=0.9313;t1=307;Tw=180+273;Const=0.0125;%blowby constant;
%mp=1;co2p=0;np=0;%100% methane as fuel
mp=0.53;co2p=0.45;np=0.05;%LFG composition as fuel
IVC=26;
theta=180+IVC;thetae=430;thetab=45;thetas=356;thetaig=356;
thetacomps=180+IVC;
volueff=0.9;
y=[P1 10000 t1 0 0 0];
[y mole,Ru,Cpu,Cvu,hu,su,vu,DLVLtu,DLVLPu]=farg(y(1),y(3),phi,f);
m0=Vol(theta,CR)/vu*volueff;%m3 to cm3 conversion of volume
m0

options = odeset('RelTol',1e-4,'AbsTol',[1e-5 1e-5 1e-5 1e-4 1e-4 1e-4]);
[theta,Y] = ode113(@equations,(theta:step:thetas),y,options);
y1=[Y(size(Y,1),1) Y(size(Y,1),2) Y(size(Y,1),3) Y(size(Y,1),4) Y(size(Y,1),5)
Y(size(Y,1),6)];
y1
Tb=Tinitialize(y1(1),y1(3),phi,f);%temp initialization for combustion phase.
%Tb
y1=[Y(size(Y,1),1) Tb Y(size(Y,1),3) Y(size(Y,1),4) Y(size(Y,1),5) Y(size(Y,1),6)];
[theta1,Y1] = ode113(@equations,(thetas:step:thetas+thetab),y1,options);
Tu=10000;%unburne
d temp in expansion phase
y2=[Y1(size(Y1,1),1) Y1(size(Y1,1),2) Tu Y1(size(Y1,1),4) Y1(size(Y1,1),5)
Y1(size(Y1,1),6)];
[theta2,Y2] = ode113(@equations,(thetas+thetab:step:thetae),y2,options);
%
% for i=theta:1:thetas
%     pm(i)=Pm(i,CR);
% end
%
Xaxis=[theta; theta1; theta2];
ans1=[Y; Y1; Y2];
Figure(1);
plot(Xaxis,ans1(:,1));

% % Figure(2);
% plot(Xaxis,ans1(:,2),'.-');
% Figure(3);
% plot(Xaxis,ans1(:,3),'o-');
%
plot(theta,Y(:,1));
plot(theta,Y(:,2),'.-',theta,Y(:,3),theta,Y(:,4),theta,Y(:,5),theta,Y(:,6));
%ans=[Y(:,1),Y(:,2),Y(:,3),Y(:,4),Y(:,5),Y(:,6)]

Equations.m

function dy = equations(theta,y)
dy=zeros(6,1);
global m0 f CR phi P1 t1 thetab thetas thetaig Tw Const thetacomps

%input parameters to be given for modeling

global dv1
dia=8.255; pi=3.1415927; stroke=11.43; % stroke length in cm
rpm=600;
w=2*pi*rpm/60;
Sp=2*stroke/100*rpm/60; % mean piston speed in m/s.

x=1-exp(-5*((theta-thetaig)/thetab)^5.5);
%x=0.5*(1-cos(pi*(theta-thetas)/thetab));
if (theta<=thetas)
    x=0;
end
if (theta>thetas+thetab)
    x=1;
end

m=m0*exm(theta,Const); % m0 is a constant value to be given, Const is blowby
const to be given
A=1/m*(DV(theta)+Vol(theta,CR)*Const/w);
theta
if (x<=0.0001) % compression stroke

[y1, R, Cpu, Cv, H, su, uu, vu, DLVLtu, DLVLPu] = farg(y(1), y(3), phi, f); % calling
farg fuel air residual gas composition properties.

C1=2.28; C2=0; Vd=611.729; % displacement volume in cm3.
% T=y(1)*Vol(theta,CR)/m/R/10; % average gas temp. div by 10 for units bar
and cm3
gasvel=C1*Sp+C2*Vd*t1/P1/Vol(thetacomps,CR)*(y(1)-Pm(theta,CR)); % gas
velocity in m/s
h=3.26*(dia/100)^(-0.2)*(y(1)*100)^0.8*(y(3)^(-0.55))*gasvel;

B=h*(pi*dia^2/2+4*Vol(theta,CR)/dia)/w/m/10000 * (vu/Cpu*DLVLtu*(1-
Tw/y(3))); % why 10000 from ferguson may be units
C=0;
D=0;
E=(vu^2/Cpu/y(3)*DLVLtu^2 + 10*vu/y(1)*DLVLPu);

dy(1)=((A+B+C)/(D+E))*10; % pressure P differential
% dy(1)=(Pm(theta+0.1,CR)-Pm(theta-0.1,CR))/0.2;
dy(2)=0;%Tb differential
% dy(3)=h*(pi*dia^2/2+4*Vol(theta,CR)/dia)/w/m/10000/Cpu*(y(3)-Tw) + 
vu/Cpu*DLVLtu*((A+B+C)/(D+E))*10/10;%Tu differential
dy(3)=h*(pi*dia^2/2+4*Vol(theta,CR)/dia)/w/m/10000/Cpu*(y(3)-Tw) + 
vu/Cpu*DLVLtu*((A+B+C)/(D+E))*10/10;%Tu differential

end

if (x>0.0001 && x<=0.999)%Combustion

[y1,Ru,Cpu,Cvu,hu,su,uu,vu,DLVLtu,DLVLPu]=farg(y(1),y(3),phi,f);%calling farg fuel air residual gas composition properties.unburned gas region

[moles,Rb,Cpb,Cvb,hb,ub,vb,DLVLtb,DLVLPb]=ferguson3(y(1),y(2),phi);% calling ECP equilibrium composition products. burned gas region

C1=2.28;C2=0.00324;Vd=611.729;%displacement volume in cm3.
% R=Ru*(1-x)+Rb*x;
% T=y(1)*Vol(theta,CR)/m/R/10;%average gas temp. div by 10 for units bar and cm3
T=y(3)*(1-x)+y(2)*x;
gasvel=C1*Sp+C2*Vd*t1/P1/Vol(thetacomps,CR)*y(1)-Pm(theta,CR));%gas velocity in m/s
h=3.26*(dia/100)^(-0.2)*(y(1)*100)^0.8*T^(-0.55)*gasvel;
%h=h^0.4;
%checktemp=[checktemp; y(3) y(2) T Ru Rb x];

B=h*(pi*dia^2/2+4*Vol(theta,CR)/dia)/w/m/10000 *
(vb/Cpb*DLVLtb*x^0.5*(y(2)-Tw)/y(2) + vu/Cpu*DLVLtu*(1-x^0.5)*(y(3)-
Tw)/y(3));%why 10000 from ferguson may be units
DxDtheta=(1-x)*27.5/thetab*((theta-thetaig)/thetab)^4.5;
%DxDtheta=0.5*sin(pi*(theta-0thetas)/thetab)*pi/thetab;
C = -(vb - vu) \* DxDtheta - vb \* DLVLtb \* (hu - hb) / Cpb \* y(2) \* (DxDtheta - (x - x^2) * Const / w);
D = x * (vb^2 / Cpb \* y(2) * DLVLtb^2 + 10 * vb / y(1) \* DLVLpb);
E = (1 - x) * (vu^2 / Cpu \* y(3) * DLVLTu^2 + 10 * vu / y(1) \* DLVLpu);

\[ dy(1) = (A + B + C) / (D + E) \* 10; \] % pressure P differential
\[ dy(2) = -h * (pi * dia^2 / 2 + 4 \* Vol(theta, CR) / dia) * x^0.5 * (y(2) - Tw) / w / m / Cpb / x / 10000 + vb / Cpb * DLVLtb * dy(1) / 10 + (hu - hb) / x / Cpb * (DxDtheta - (x - x^2) * Const / w); \] % Tb differential
\[ dy(3) = -h * (pi * dia^2 / 2 + 4 \* Vol(theta, CR) / dia) * (1 - x^0.5) * (y(3) - Tw) / w / m / Cpu / (1 - x) / 10000 + vu / Cpu * DLVLTu * dy(1) / 10; \] % Tu differential
Hl = ((1 - x^2) * hu + x^2 * hb);

\textbf{if} (x > 0.999) \% Expansion
\textbf{x} = 1;

[moles, Rb, Cpb, Cvb, Hl, sb, ub, vb, DLVLtb, DLVLpb] = ferguson3(y(1), y(2), phi); % calling ECP equilibrium composition products. burned gas region

C1 = 2.28; C2 = 0.00324; Vd = 611.729; \% displacement volume in cm3.
% T = y(1) * Vol(theta, CR) / m / Rb / 10; \% average gas temp. div by 10 for units bar and cm3
gasvel = C1 * Sp + C2 * Vd * t1 / P1 / Vol(thetacomps, CR) * (y(1) - Pm(theta, CR)); \% gas velocity in m/s
h = 3.26 * (dia / 100)^(-0.2) * (y(1) * 100)^0.8 * y(2)^(-0.55) * gasvel;
% h = 0; \% exhaust side curve is good match with h = 0

B = h * (pi * dia^2 / 2 + 4 * Vol(theta, CR) / dia) / w / m / 10000 * (vb / Cpb * DLVLtb * (y(2) - Tw) / y(2)); \% why 10000 from ferguson may be units
C = 0;
D = (vb^2 / Cpb / y(2) * DLVLtb^2 + 10 * vb / y(1) * DLVLpb);
E = 0;
\[ dy(1) = (A + B + C) / (D + E) \* 10; \] % pressure P differential
\[ dy(2) = -h * (pi * dia^2 / 2 + 4 * Vol(theta, CR) / dia) * (y(2) - Tw) / w / m / Cpb / 10000 + vb / Cpb * DLVLtb * dy(1) / 10; \] % Tb differential
\[ dy(3) = 0; \] % Tu differential
end

\[ dy(4) = y(1) * DV(\text{theta}) / 10; \] % work W differential
dy(5)=h/w*(pi*dia^2/2+4*Vol(theta,CR)/dia)/10000*(x^0.5*(y(2)-Tw)+(1-x^0.5)*(y(3)-Tw));%heat loss Ql differential
dy(6)= Const*m/w*Hl;%enthalpy loss Hl differential
dv1=[dv1;theta y(1)];

Farg.m

% FUEL AIR RESIDUAL GAS PROPERTIES farg
%
function [y,R,Cp,Cv,H,S,U,v,DLVLt,DLVLP]=farg(P,t,phi,f)

global mp co2p np

if phi==1
    phi=1.001;%To avoid instability near phi=1.000
end

P=0.986923267 *P;% converting the pressure in bars to atmospheres
%All the equations have been developed having reference pressure Po=1 atm.
%But the SI unit is bars; so this conversion to atm within the program.
a=1;b=4;gamma=0;del=0;%For Methane as Fuel
%These values should be changed if we have a different fuel.
e=0.21/(a+0.25*b-0.5*gamma)/mp;%Stoichiometric fuel-air ratio.
%
%Tolerance for delx=0 checking and f1=0 checking respectively

tol1=10^(-9); %for t=1000 this value is 10^-25
%tol2=10^(-1);
%maxiter=10000;

%Initial guess calculation from Fuel Air Residual Gas calculations.
K=k(5,t)*k(2,t)/k(1,t)/k(6,t);
\%  
\% aa=1-K;  
\% bb=0.42*phi*e*(2*a-gamma)+K*(0.42*(phi-1)+a*phi*e);  
\% cc=-0.42*a*phi*e*(phi-1)*K;  

\% DccDt=-0.42*a*phi*e*(phi-1)*DKDT(t); \% slightly different from ferguson  
\% may be reqd to change later.  

\% v5=(-bb+sqrt(bb^2-4*aa*cc))/(2*aa);  

\% Modified for LFG analysis  

if phi>1  

a1=phi*e*(a*mp+co2p);  
a2=phi*e*b*mp;  
a3=phi*e*gamma*mp+0.42+phi*e*co2p*2;  
a4=phi*e*del*mp+1.58+phi*e*np*2;  

sf=solve('(a3-n5-2*(a1-n5))*n5/(a1-n5)/(a2/2-(a3-n5-2*(a1-n5))-K','n5');  
A=eval(sf);  
if (A(1)>0)  
  n5=A(1);  
else if (A(2)>0)  
  n5=A(2);  
else  
  n5=10000;  
end  
end  

x1=a1-n5;  
x2=a3-n5-2*x1;  
x=a4/2;  
y=tol1;  
p=n5;
\[ q = \frac{a_2}{2} - x_2; \]

else
\[ x_1 = a \cdot \phi \cdot e \cdot mp + \phi \cdot e \cdot co2; \]
\[ x_2 = b \cdot \phi \cdot e \cdot mp / 2; \]
\[ x = 0.79 + \delta \cdot \phi \cdot e \cdot mp / 2 + \phi \cdot e \cdot np; \]
\[ y = 0.21 \times (1 - \phi); \]
\[ p = tol1; \]
\[ q = tol1; \]
end

moles = [x1; x2; x; y; p; q];

molemass = [44.0098 18.01528 28.0134 31.9988 28.0104 2.01588];

% COMPUTE MOLE FRACTIONS AND MOLECULAR WEIGHT OF RESIDUAL
\[ t_{moles} = x_1 + x_2 + x + y + p + q; \]
\[ m_{res} = 0; \]
for \( i = 1:6 \)
\[ y(i,1) = moles(i,1) / t_{moles}; \]
\[ m_{res} = m_{res} + y(i,1) \times molemass(1,i); \]
end

% COMPUTE MOLE FRACTIONS AND MOLECULAR WEIGHT OF FUEL-AIR
fuelmain = \( e \cdot \phi \cdot mp / (1 + e \cdot \phi); \]
\[ co2 = \phi \cdot e \cdot co2 / (1 + e \cdot \phi); \]
\[ o2 = 0.21 / (1 + e \cdot \phi); \]
\[ n2 = (0.79 + \phi \cdot e \cdot np) / (1 + e \cdot \phi); \]
mfa = fuelmain * (12.01*a + 1.008*b + 16*gamma + 14.01*del) + 32*o2 + 28.02*n2 + co2*44.01;

% COMPUTE MOLE FRACTIONS OF FUEL-AIR-RESIDUAL GAS
yres = \( f / (f + m_{res} / mfa * (1 - f)); \)
for \( i = 1:6 \)
\[ y(i,1) = y(i,1) \times yres; \]
end
    yfuelmain=fuelmain*(1-yres);
y(1,1)=y(1,1)+co2*(1-yres);
y(3,1)=y(3,1)+n2*(1-yres);
y(4,1)=y(4,1)+o2*(1-yres);

%COMPUTE COMPONENT PROPERTIES
% this part is not necessary because we already have the properties from the % tables.

%COMPUTE PROPERTIES OF MIXTURE
DLVLt=1.0;
DLVLP=-1.0;
MW=molemass*y;% in g/mol molecular weight of the mixture.
MW=MW+16.04276*yfuelmain;% above continuation
Ru=8.31434;%/mol/K..........................(Universal gas constant)
R=Ru/MW;%/g/K.............................(Gas constant)
H=ho(13,t)*yfuelmain;
Cp=cp(13,t)*yfuelmain;

S=-R*log(P);
S=S+yfuelmain*(so(13,t)-R*log(yfuelmain));

table=[-1,1,0,0,1,-1];%used for Cp calculation from ferguson

for i=1:6
    H=H+ho(i,t)*y(i,1);
    if (y(i,1)>1^(-25))
        S=S+y(i,1)*(so(i,t)-R*log(y(i,1)));%S in J/mol/K............(Entropy)eq.3.21
    else
        S=S+y(i,1)*(so(i,t)-R*log(1^(-25)));
    end
    %Cp=Cp+cp(i,t)*y(i,1);%maybe if want the secondary part having DCDT for accuracy then we have to include.
    Cp=Cp+cp(i,t)*y(i,1)+ho(i,t)*table(1,i)*DccDt*yres/tmoles;
% the last part is removed for LFG modelling as DccDT was not able to get
end
H=H*1000/MW;% in J/g unit ---- kJ to J conversion
Cp=Cp/MW;% check it out
U = H - R*t;
v = 10*R*t/(P*1.01325); % in cm3/g, pressure in atm to bars. so specific volume
% specific volume in cm3/g
Cv = Cp + P*v/t*DLVLt^2/DLVLP/10; % why becos vol in cm3 and P in bar

Ferguson3.m

% Equilibrium Combustion Products ECP
% To find out the mole fractions when a fuel is burned in equilibrium
% Fuel is C a H b O gamma N del with e as the stoichiometric fuel/air ratio
% Assumed species are : CO2, H2O, N2, O2, CO, H2, H,O, HO, NO, N, NO2, CH4--13 species
% Input: P (in bars), t (in K), phi (in dimensionless ratio).
% Output: Mole fractions of 13 species (in dimensionless) and other properties.
% Method Followed: Using chemical reaction equations and chemical
% equilibrium equations we get 4 atom balance equation and 9 equilibrium
% constant equations respectively. Which are Non-Linear in nature. These 13
% equations are solved using Newton Raphson Method (nonlinear) and Gauss
% Jordan Method (linear).
% In this 4 mole fractions x3 x4 x5 x6 are independent variables so named as
% x, y, p, q and the other 9 mole fractions
% x1, x2, x7, x8, x9, x10, x11, x12, x13 are dependent variables. f1 f2 f3 f4 are the
% four long equations to be solved. Newton Raphson Method bring down the
% system to [A][del.x y p q]+[F]=0; This matrix equation is solved by Gauss
% Jordan method of Linear equation solving to bring down the del x y p q
% values less than a tolerance value which is 10^-9 (0.000000001) in this
% case in order to get the convergence in all cases.

function [moles, R, Cp, Cv, h, s, u, v, DLVLt, DLVLP] = ferguson3(P, t, phi)
global mp co2p np

if phi==1
    phi=1.001; % To avoid instability near phi=1.000
end

P=0.986923267 *P; % converting the pressure in bars to atmospheres
% All the equations have been developed having reference pressure Po=1 atm.
% But the SI unit is bars; so this conversion to atm within the program.

a=1;b=4;gamma=0;del=0; % For Methane as Fuel
% These values should be changed if we have a different fuel.
e=0.21/(a+0.25*b-0.5*gamma)/mp; % Stoichiometric fuel-air ratio.

% Tolerance for delx=0 checking and f1=0 checking respectively

tol1=10^(-9); % for t=1000 this value is 10^-25
tol2=10^(-1);
maxiter=10000;

% Equilibrium constant values for the considered equilibrium reactions.

k7=k(7,t)/k(6,t)^0.5;
k8=k(8,t)/k(4,t)^0.5;
k9=k(9,t)/k(6,t)^0.5/k(4,t)^0.5;
k10=k(10,t)/k(4,t)^0.5/k(3,t)^0.5;
k2=k(2,t)/k(6,t)/k(4,t)^0.5;
k1=k(1,t)/k(5,t)/k(4,t)^0.5;
k11=k(11,t)/k(3,t)^0.5;
k12=k(12,t)/k(10,t)/k(4,t)^0.5;
k13=k(13,t)*k(2,t)^2/k(6,t)^4/k(1,t);

% constants appearing in the f1 f2 f3 f4 equations

cc1=(e*phi*b*mp/(e*phi*a*mp+e*phi*co2p));
cc2=(e*phi*gamma*mp+0.42+e*phi*co2p*2)/(e*phi*a*mp+e*phi*co2p);
cc3=(e*phi*del*mp+1.58+e*phi*np*2)/(e*phi*a*mp+e*phi*co2p);
% Initial guess calculation from Fuel Air Residual Gas calculations.

\[ K = k(5,t) \times k(2,t) / k(1,t) / k(6,t); \]
% \( a = 1 - K; \)
% \( b = 0.42 \times \phi \times e \times (2 \times a - \gamma) + K \times (0.42 \times (\phi - 1) + a \times \phi \times e); \)
% \( c = -0.42 \times a \times \phi \times e \times (\phi - 1) \times K; \)
% \( v = (-b + \sqrt{b^2 - 4 \times a \times c}) / (2 \times a); \)

if \( \phi > 1 \)

\( a = \phi \times e \times (a \times mp + co2p); \)
\( a = \phi \times e \times b \times mp; \)
\( a = \phi \times e \times gamma \times mp + 0.42 + \phi \times e \times co2p \times 2; \)
\( a = \phi \times e \times del \times mp + 1.58 + \phi \times e \times np \times 2; \)

\[ sf = \text{solve}((a3 - n5 - 2 \times (a1 - n5)) \times n5 / (a1 - n5) / (a2 / 2 - (a3 - n5 - 2 \times (a1 - n5))) - K', 'n5'); \]
A = eval(sf);
if (A(1) > 0)
    \( n5 = A(1); \)
else if (A(2) > 0)
    \( n5 = A(2); \)
else
    \( n5 = 10000; \)
end
end

\( x1 = a1 - n5; \)
\( x2 = a3 - n5 - 2 \times x1; \)
\( x = a4 / 2; \)
\( y = tol1; \)
\( p = n5; \)
\( q = a2 / 2 - x2; \)
else
    x1 = a * phi * e * mp + phi * e * co2p;
    x2 = b * phi * e * mp / 2;
    x = 0.79 + del * phi * e * mp / 2 + phi * e * np;
    y = 0.21 * (1 - phi);
    p = tol1;
    q = tol1;
end

x7 = 0; x8 = 0; x9 = 0; x10 = 0; x11 = 0; x12 = 0; x13 = 0; moleschanget(1:13, 1) = 0; moleschangeP(1:13, 1) = 0;

if t > 1200

% Iterations started here for Newton Raphson Method
cont = 1; % for running the while loop
count = 1; % for counting the no. of iterations

while (cont == 1) && (count < maxiter)

% Dependent variables

    x7 = P^(-0.5) * (q^0.5) * k7;
    x8 = P^(-0.5) * (y^0.5) * k8;
    x9 = (q^0.5) * (y^0.5) * k9;
    x10 = (y^0.5) * (x^0.5) * k10;
    x2 = (P^(0.5) * q * (y^0.5) * k2);
    x1 = P^0.5 * p * (y^0.5) * k1;
    x11 = (P^(-0.5) * (x^0.5) * k11);
    x12 = (P^(0.5) * x10 * (y^0.5) * k12);
    x13 = k13 * q^4 * x1 * P^2 / x2^2;

% f functions
f1 = x1 + x2 + x + y + p + q + x7 + x8 + x9 + x10 + x11 + x12 + x13 - 1; % this minus 1 is what makes
% the moles into molefractions itself. whereas in the farg.m program these
%are different and molefractions are calculated separately.
f2=2*x2+2*q+x7+x9+4*x13-cc1*(x1+p+x13);
f3=2*x1+x2+2*y+p+x8+x9+x10+2*x12-cc2*(x1+p+x13);
f4=2*x+x10+x11+x12-cc3*(x1+p+x13);

%Non-Zero derivatives
D76=k7*P^(-0.5)*0.5*q^(-0.5);
D84=k8*P^(-0.5)*0.5*y^(-0.5);
D94=k9*q^0.5*0.5*y^(-0.5);
D96=k9*y^0.5*0.5*q^(-0.5);
D104=k10*x^0.5*0.5*y^(-0.5);
D103=k10*y^0.5*0.5*x^(-0.5);
D24=k2*q*P^0.5*0.5*y^(-0.5);
D26=k2*y^0.5*P^0.5;
D14=k1*p*P^0.5*0.5*y^(-0.5);
D15=k1*y^0.5*P^0.5;

D113=k11*P^(-0.5)*0.5*x^(-0.5);
D123=k12*k10*y^0.5*0.5*x^(-0.5);
D124=k12*k10*x^0.5*P^0.5;
D134=k13*q^2*k1*p^0.5*P^1.5*(-0.5)*y^(-1.5)/k2^2;
D135=k13*q^2*k1*y^(-0.5)*P^1.5/k2^2;
D136=k13*k1*(y^(-0.5))*p*(P^1.5)*2*q/k2^2;

%Matrix elements
A11=1+D103+D113+D123;
A12=D14+D24+1+D84+D94+D104+D124+D134;
A13=D15+1+D135;
A14=D26+1+D76+D96+D136;

A21=0;
A22=2*D24+D94+4*D134-cc1*(D134+D14);
A23=4*D135-cc1*(1+D15+D135);
A24=2*D26+2+D76+D96+4*D136-cc1*D136;

A31=D103+2*D123;
A32=2*D14+D24+2+D84+D94+D104+2*D124-cc2*(D14+D134);
A33=2*D15+1-cc2*(D15+1+D135);
A34=D26+D96-cc2*D136;
A41=2+D103+D113+D123;
A42=D104+D124-cc3*(D14+D134);
A43=-cc3*(1+D135+D15);
A44=-cc3*D136;

A=[A11 A12 A13 A14;A21 A22 A23 A24;A31 A32 A33 A34;A41 A42 A43 A44];
F=[f1;f2;f3;f4];
B=-F;

A=[A B];
A=solvegaussjordan(A);%Gauss Jordan Method for Linear equations solving

delx=A(1,5);
dely=A(2,5);
delp=A(3,5);
delq=A(4,5);

%Tolerance value check for delx dely delp delq.
if
((abs(delx)<tol1)&(abs(dely)<tol1)&(abs(delp)<tol1)&(abs(delq)<tol1))==1
    if((abs(f1)<tol2)&(abs(f2)<tol2)&(abs(f3)<tol2)&(abs(f4)<tol2))==1
        cont=0;
        break
    end
end

%x=(x+delx);y=(y+dely);p=(p+delp);q=(q+delq);
x=abs(x+delx);y=abs(y+dely);p=abs(p+delp);q=abs(q+delq);

count = count+1;
end %end for while loop iteration
%count

% % % %To find the change of mole fractions wrt temp.

c1=k1*P^0.5;
c2=k2*P^0.5;
\[c7 = k7 \cdot P^{-0.5};
\]
\[c8 = k8 \cdot P^{-0.5};
\]
\[c9 = k9;
\]
\[c10 = k10;
\]
\[c11 = k11 \cdot P^{-0.5};
\]
\[c12 = k12 \cdot k10 \cdot P^{0.5};
\]
\[c13 = k13 \cdot k1\cdot P^{1.5}/k2^2;
\]

\[X1 = x1/c1;
\]
\[X2 = x2/c2;
\]
\[X7 = x7/c7;
\]
\[X8 = x8/c8;
\]
\[X9 = x9/c9;
\]
\[X10 = x10/c10;
\]
\[X11 = x11/c11;
\]
\[X12 = x12/c12;
\]
\[X13 = x13/c13;
\]

\[dk7dt = (k(6,t)^{0.5}\cdot dk(7,t) - k(7,t)\cdot 0.5\cdot k(6,t)^{0.5}\cdot k(6,t)^{-0.5}\cdot dk(6,t))/k(6,t);
\]
\[dk8dt = (k(4,t)^{0.5}\cdot dk(8,t) - k(8,t)\cdot 0.5\cdot k(4,t)^{0.5}\cdot k(4,t)^{-0.5}\cdot dk(4,t))/k(4,t);
\]
\[dk9dt = (k(6,t)^{0.5}\cdot k(4,t)^{0.5}\cdot dk(9,t) - k(9,t)\cdot (k(6,t)\cdot 0.5\cdot k(4,t)^{-0.5}\cdot dk(4,t) + k(4,t)^{0.5}\cdot k(6,t)^{-0.5}\cdot dk(6,t)))/(k(6,t)\cdot k(4,t));
\]
\[dk10dt = (k(4,t)^{0.5}\cdot k(3,t)^{0.5}\cdot dk(10,t) - k(10,t)\cdot (k(4,t)\cdot 0.5\cdot k(3,t)^{-0.5}\cdot dk(3,t) + k(3,t)^{0.5}\cdot k(4,t)^{-0.5}\cdot dk(4,t)))/(k(3,t)\cdot k(4,t));
\]
\[dk2dt = (k(6,t)^{0.5}\cdot k(4,t)^{0.5}\cdot dk(2,t) - k(2,t)\cdot (k(6,t)\cdot 0.5\cdot k(4,t)^{-0.5}\cdot dk(4,t) + k(4,t)^{0.5}\cdot dk(6,t)))/(k(6,t)^2\cdot k(4,t));
\]
\[dk1dt = (k(5,t)^{0.5}\cdot k(4,t)^{0.5}\cdot dk(1,t) - k(1,t)\cdot (k(5,t)\cdot 0.5\cdot k(4,t)^{-0.5}\cdot dk(4,t) + k(4,t)^{0.5}\cdot dk(5,t)))/(k(5,t)^2\cdot k(4,t));
\]
\[dk11dt = (k(3,t)^{0.5}\cdot dk(11,t) - k(11,t)\cdot 0.5\cdot k(3,t)^{-0.5}\cdot dk(3,t))/k(3,t);
\]
\[dk12dt = (k(10,t)^{0.5}\cdot k(4,t)^{0.5}\cdot dk(12,t) - k(12,t)\cdot (k(10,t)\cdot 0.5\cdot k(4,t)^{-0.5}\cdot dk(4,t) + k(4,t)^{0.5}\cdot dk(10,t)))/(k(10,t)^2\cdot k(4,t));
\]
\[dk13dt = ((k(6,t)^{4}\cdot k(1,t)*k(13,t)^{2}\cdot k(2,t)^{2}\cdot dk(2,t) + k(2,t)^{2}\cdot dk(13,t)) - k(13,t)^{2}\cdot k(2,t)^{2}\cdot k(6,t)^{4}\cdot dk(1,t) - k(1,t)^{4}\cdot k(6,t)^{3}\cdot dk(6,t)))/(k(6,t)^8\cdot k(1,t)^2);
\]

\[Dc1DT = P^{0.5}\cdot dk1dt;
\]
\[Dc2DT = P^{0.5}\cdot dk2dt;
\]
\[Dc7DT = P^{(-0.5)}\cdot dk7dt;
\]
\[Dc8DT = P^{(-0.5)}\cdot dk8dt;
\]
\[Dc9DT = dk9dt;
\]
\[Dc10DT = dk10dt;
\]
\[Dc11DT = P^{(-0.5)}\cdot dk11dt;\]
Dc12DT=P^0.5*(k12*dk10dt+k10*dk12dt);
Dc13DT=P^1.5*(k2^2*(k13*dk1dt+k1*dk13dt)-k13*k1*2*k2*dk2dt)/k2^4;

Df1DT=Dc1DT*X1+Dc2DT*X2+Dc7DT*X7+Dc8DT*X8+Dc9DT*X9+Dc10DT*X10+Dc11DT*X11+Dc12DT*X12+Dc13DT*X13;
Df2DT=2*Dc2DT*X2+Dc7DT*X7+Dc9DT*X9+4*Dc13DT*X13-cc1*(Dc1DT*X1+Dc13DT*X13);
Df3DT=2*Dc1DT*X1+Dc2DT*X2+Dc8DT*X8+Dc9DT*X9+Dc10DT*X10+2*Dc12DT*X12-cc2*(Dc1DT*X1+Dc13DT*X13);
Df4DT=Dc10DT*X10+Dc11DT*X11+Dc12DT*X12-cc3*(Dc1DT*X1+Dc13DT*X13);

A=[A11 A12 A13 A14;A21 A22 A23 A24;A31 A32 A33 A34;A41 A42 A43 A44];
DFDT=[Df1DT;Df2DT;Df3DT;Df4DT];
B=-DFDT;

A=[A B];
A=solvegaussjordan(A);%Gauss Jordan Method for Linear equations solving

DxDT=A(1,5);
DyDT=A(2,5);
DpDT=A(3,5);
DqDT=A(4,5);

Dx7DT=q^0.5*Dc7DT+D76*DqDT;
Dx8DT=y^0.5*Dc8DT+D84*DyDT;
Dx9DT=y^0.5*q^0.5*Dc9DT+D96*DqDT+D94*DyDT;
Dx10DT=Dc10DT*y^0.5*x^0.5+D104*DyDT+D103*DxDT;
Dx11DT=Dc11DT*x^0.5+D113*DxDT;
Dx12DT=Dc12DT*x^0.5*y+D123*DxDT+D124*DyDT;
Dx13DT=Dc13DT*q^2*p^y^(-0.5)+D136*DqDT+D135*DpDT+D134*DyDT;
Dx14DT=Dc14DT*p*y^0.5+D145*DpDT+D14*DyDT;
Dx2DT=Dc2DT*q^y^0.5+D26*DqDT+D24*DyDT;

moleschanget=[Dx1DT;Dx2DT;DxDT;DyDT;DpDT;DqDT;Dx7DT;Dx8DT;Dx9DT;Dx10DT;Dx11DT;Dx12DT;Dx13DT];
% To find the change of mole fractions wrt temp.
% % % % % % % % % % % % % % % % % % % % % % % % % % % % % % % % %

% % % % % To find the change of mole fractions wrt pressure.
% This is required for Cv calculation.
% Only Dc1DP is found newly; all other equations are same as the above set
% but just replacing DfiDT with DfiDP and also DxiDT with DxiDP

Dc1DP = k1 * 0.5 * P^(-0.5);
Dc2DP = k2 * 0.5 * P^(-0.5);
Dc7DP = k7 * (-0.5) * P^(-1.5);
Dc8DP = k8 * (-0.5) * P^(-1.5);
Dc9DP = 0;
Dc10DP = 0;
Dc11DP = k11 * (-0.5) * P^(-1.5);
Dc12DP = k12 * k10 * 0.5 * P^(-0.5);
Dc13DP = k13 * 1.5 * P^(0.5) / (k1 * k2^2);

Df1DP = Dc1DP * X1 + Dc2DP * X2 + Dc7DP * X7 + Dc8DP * X8 + Dc9DP * X9 + Dc10DP * X10 + Dc11DP * X11 + Dc12DP * X12 + Dc13DP * X13 - cc1 * (Dc1DP * X1 + Dc13DP * X13);
Df2DP = 2 * Dc2DP * X2 + Dc7DP * X7 + Dc9DP * X9 + 4 * Dc13DP * X13 - cc1 * (Dc1DP * X1 + Dc13DP * X13);
Df3DP = 2 * Dc1DP * X1 + Dc2DP * X2 + Dc8DP * X8 + Dc9DP * X9 + Dc10DP * X10 + Dc12DP * X12 - cc2 * (Dc1DP * X1 + Dc13DP * X13);
Df4DP = Dc10DP * X10 + Dc11DP * X11 + Dc12DP * X12 - cc3 * (Dc1DP * X1 + Dc13DP * X13);

A = [A11 A12 A13 A14; A21 A22 A23 A24; A31 A32 A33 A34; A41 A42 A43 A44];
DFDP = [Df1DP; Df2DP; Df3DP; Df4DP];
B = -DFDP;

A = [A B];
A = solvegaussjordan(A); % Gauss Jordan Method for Linear equations solving

DxDP = A(1, 5);
DyDP = A(2, 5);
DpDP = A(3, 5);
DqDP = A(4,5);

Dx7DP = q^0.5*^0.5*DC7DP + D76*DqDP;
Dx8DP = y^0.5*DC8DP + D84*DyDP;
Dx9DP = y^0.5*q^0.5*DC9DP + D96*DqDP + D94*DyDP;
Dx10DP = Dc10DP*y^0.5*x^0.5 + D104*DyDP + D103*DxDP;
Dx11DP = Dc11DP*x^0.5 + D113*DxDP;
Dx12DP = Dc12DP*x^0.5 + D123*DxDP + D124*DyDP;
Dx13DP = Dc13DP*q^2*p*y^(-0.5) + D136*DqDP + D135*DpDP + D134*DyDP;
Dx1DP = Dc1DP*p*y^0.5 + D15*DpDP + D14*DyDP;
Dx2DP = Dc2DP*q*y^0.5 + D26*DqDP + D24*DyDP;

moleschangeP = [Dx1DP; Dx2DP; DxDP; DyDP; DpDP; DqDP; Dx7DP; Dx8DP; Dx9DP; Dx10DP; Dx11DP; Dx12DP; Dx13DP];

%To find the change of mole fractions wrt pressure.%%%%%%%%%%%%%%%%%%%
end % end for if t>1200

moles = [x1; x2; x; y; p; q; x7; x8; x9; x10; x11; x12; x13];


Mt = molemass * moleschange; %g/K..............(change of mass wrt temp.)
MP = molemass * moleschangeP; %g/K..............(change of mass wrt pressure.)

MW = molemass * moles; % in g/mol..............(Mass of Equilibrium combustion products mixture)

Ru = 8.31434; %J/mol/K..................(Universal gas constant)
R = Ru / MW; %J/g/K......................(Gas constant)
h=0;
s=-R*log(P);
Cp=0;

for i=1:13
    h=h+ho(i,t)*moles(i,1);% h in kJ/mol...........(Enthalpy)
    if moles(i,1)~=0
        s=s+moles(i,1)*(so(i,t)-R*log(moles(i,1)));% s in J/mol/K...........(Entropy)eq.3.21
    end
    Cp=Cp+moles(i,1)*cp(i,t)+ho(i,t)*1000*moleschanget(i,1);% in J/mol/K
end

Cp=(Cp-h*1000*Mt/MW)/MW; %'Cp in J/g/K'........................(Specific heat at const. pressure)becos here mol=1;
h=h*1000/MW; %'h in J/g'
s=s/MW;%'s in J/g/K'
u=h-R*t;%'u in J/g'
v=10*R*t/(P*1.01325);% 'in cm3/g' % pressure in atm to bars. so specific volume

DLVLt=1-t*Mt/MW; %reqd for Cv calculation in future.
DLVLP=-1-P*MP/MW; %reqd for Cv calculation in future.

Cv=Cp+P*v/t*DLVLt^2/DLVLP/10; %'why i dont know just followed the ferguson book it gives this formula and in the program 10 is divided as done here.'

Solvegaussjordan.m

function [A]=solvegaussjordan(A)

% Gauss Jordan Method for Linear equations solving

% if singular rows will be swapped
row=0;
singular=0;
for l=1:3
    for i=1:3
        if A(i,i)==0
            swaprow=A(i,:);
            row=i;
            break
        end
    end

    if row==0
        break
    end

    for j=1:3
        if A(j,row)~=0
            A(row,:)=A(j,:);
            A(j,:)=swaprow;
            break
        end
    end

    row=0;
end

% row manipulations to make it diagonally 1 everywhere
for i=1:4
    divisor=A(i,i);

    for j=1:5
        A(i,j)=A(i,j)/divisor;
    end

for kw=1:4
    mult=A(kw,i);
    for j=1:5
        if kw~=i
A(kw,j) = A(kw,j) - mult*A(i,j);
end
end
end
end
Appendix C

LabVIEW VIs developed for data acquisition and analysis
LabVIEW VIs and its functions for data acquisition.

<table>
<thead>
<tr>
<th>No.</th>
<th>LabVIEW VI</th>
<th>Function of the Module</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Continuous pressure100cycle.vi</td>
<td>Records 100 cycle pressure</td>
</tr>
<tr>
<td>2</td>
<td>Flow data. vi</td>
<td>Acquires flow data</td>
</tr>
<tr>
<td>3</td>
<td>Mass fraction burned. vi</td>
<td>Combustion duration analysis</td>
</tr>
<tr>
<td>4</td>
<td>Composition equation solver. vi</td>
<td>Solves the linear equations of flow to get the required composition and equivalence ratio</td>
</tr>
<tr>
<td>5</td>
<td>Continuous pressure spark timing. vi</td>
<td>Setup validation of spark timing correctness.</td>
</tr>
<tr>
<td>6</td>
<td>Formotorvolt. vi</td>
<td>For recording motor pressure</td>
</tr>
<tr>
<td>7</td>
<td>CoV power, iglag, combustion duration, Pmax. vi</td>
<td>Calculates the coefficient of variation</td>
</tr>
<tr>
<td>8</td>
<td>Data analysis retrieval. vi</td>
<td>Retrieves the data from a data file which was already run with CoV. vi</td>
</tr>
<tr>
<td>9</td>
<td>Run data retrieval. vi</td>
<td>Collects the power data against the CR, ST, EQR and H%.</td>
</tr>
</tbody>
</table>
Block diagram of pressure recording VI.
Block diagram of flowdata acquiring VI.
Block diagram of mass fraction burned VI
Block diagram of data analysis retrieval VI.
Appendix D

Alternative fuels research lab
CFR engine.

In-cylinder pressure transducer.
Fuel control panel and air intake system.
Rotary encoder fitted on the crank shaft.

Inlet and exhaust thermocouples.
Data flow stream