

QATAR UNIVERSITY
COLLEGE OF ENGINEERING
COMBUSTION CHARACTERISTICS AND EMISSION OF A DI DIESEL ENGINE
UTILIZING NEW INDUCTION MANIFOLD DESIGNS AND RUNNING ON
ALTERNATIVE FUEL BLENDS

BY
ABDELLATIF MOHAMMAD SADEQ

A Thesis Submitted to
the Faculty of the College of
Engineering
in Partial Fulfillment
of the Requirements
Dfor the Degree of
Masters of Science in Mechanical Engineering

January 2018

© 2018 Abdellatif Sadeq. All Rights Reserved

COMMITTEE PAGE

The members of the Committee approve the Thesis of Abdellatif Sadeq defended on
19/12/2017.

Dr.Samer Ahmed
Thesis/Dissertation Supervisor

Dr.Ahmad Sleiti
Committee Member

Dr.Madjid Birouk
Committee Member

Approved:

Khalifa Al-Khalifa, Dean, College of Engineering

ABSTRACT

Sadeq, Abdellatif, M., Masters : January : [2018], Masters of Science in Mechanical Engineering

Title: Combustion Characteristics and Emission of a DI Diesel Engine Utilizing New Induction Manifold Designs and Running on Alternative Fuel Blends

Supervisor of Thesis: Samer, Fikry, Ahmed.

The demands for increasing the swirl in the combustion chamber and for decreasing the exhaust emissions on diesel engines have increased rapidly over the past few years. Consequently, the researchers' attention has been attracted significantly for innovating and testing a new design for the induction manifold that can match these demands. In this project, some possible alternative designs for the normal induction manifold are presented. The design of these new manifolds is inspired from the previous researches and studies about automobiles inlet manifolds.

The test for the new manifolds involves swirl number calculations as well as a detailed performance and emission experimental test on the engine. The test also considers taking the readings for the exhaust gases (HC, CO, CO₂ and NO) and the smoke intensity using advanced measurement sensitive devices. Furthermore, this study aims to be more advanced by tracking out the pressure corresponding to each crank shaft angle by using a GW-Instek digital storage oscilloscope.

As to make this work more beneficial, the performance of the engine is also diagnosed using alternative fuels such as GTL (Gas to Liquid) fuel and using biofuels (Waste cooking oil &

Corn oil) in a blended form with diesel fuel . The results for any used alternative fuel or fuel blend in this experiment is compared with the result of diesel fuel in order to track any enhancement in engine performance or emission.

It was found that the use of the $1D$ (where D is the manifold inner diameter) new manifold can minimize the pressure variation with the crank angle position, the in-cylinder peak pressure and the particulate emission by a considerable amount due to the enhanced air-fuel mixing caused by the swirl motion generated when using this newly shaped manifold designs.

The use of GTL fuel has significantly improved the engine performance and lower its emission due to its high cetane number and low Sulfur and Aromatics content. However, the use of the new fuel blends was found to be effective in some criteria such as lowering the PM and NO emission rate due to its high oxygen content.

ACKNOWLEDGMENTS

First of all, I want to thank Allah (God) Almighty for providing me the courage and the blessing to complete this project.

I am sincerely grateful to all who supported me in completing this project effectively and on time. I would like to express my profound and sincere gratitude and appreciation to my supervisor, Associate Professor Dr. Samer Fikry, for his real encouragement, wise guidance, critical comments, and correction of the thesis. Dr. Samer entrusted me with this interesting project and provided me with all the resources necessary for accomplishing my research objectives.

Last but not least, I would like to thank my dear parents and my dear wife, who have extended their support for enabling me to study and helping me out morally and financially in this endeavor.

TABLE OF CONTENTS

ACKNOWLEDGMENTS	V
LIST OF TABLES	IX
LIST OF FIGURES	XI
CHAPTER 1. INTRODUCTION	1
1.1 Background	1
1.1.1 Induction Manifold's Design	2
1.1.2 Alternative Fuels	5
1.1.3 Fuel Blends	7
1.2 Research Objectives	7
1.3 Organization of Thesis	8
CHAPTER 2. LITERATURE REVIEW.....	11
2.1 Induction Manifold	11
2.1.1 Induction Manifold's Design	13
2.1.2 Simulation of the Fluid Motion on the Combustion Chamber	15
2.2 Effect of Induction Manifold Designs on Engine Performance and Emission	18
2.2.1 Engine Performance	18
2.2.2 Engine Emissions	23
2.3 Alternative Fuels	27
2.3.1 Alternative Fuels and Their Importance	28
2.3.2 Considerations for Alternative Fuels	29
2.3.3 Alternative Fuels for Diesel Engine	30
2.3.4 The Effect of Alternative Fuels on GHG Emission.....	45
2.4 Fuel Blends.....	48

2.4.1 Diesel-Biodiesel Blend.....	49
2.4.2 Diesel-Ethanol Blend	52
2.4.3 Diesel-Butanol Blend	56
CHAPTER 3. EXPERIMENTAL SETUP.....	60
3.1 Experimental Method	60
3.2 New Induction Manifold Designs	72
3.3 Fuels Used	78
3.4 Research Methodology	81
CHAPTER 4. RESULTS and DISCUSSION	83
4.1 Effect of Induction Manifold Design	83
4.1.1 Combustion Characteristics	84
4.1.2 Engine Performance.....	97
4.1.3 Engine Emissions	113
4.2 Effect of Using Fuel Blends	144
4.2.1 Combustion Characteristics	144
4.2.2 Engine Performance.....	154
4.2.3 Engine Emissions	165
4.3 Summary of the Results	188
CHAPTER 5. CONCLUSION and RECOMMENDATION.....	193
5.1 Conclusions	193
5.2 Future Work and Recommendations.....	195
REFERENCES	196
APPENDIX A: AIR MASS FLOW RATE CHART	204

APPENDIX B: ABSORPTION "K" TO OPACITY "%"	205
APPENDIX C: CALIBRATION CERTIFICATES	236
APPENDIX D: GTL MSDS	239
APPENDIX E: ABSORPTION "K" TO OPACITY "	250
APPENDIX F: SAFETY RULES	251

LIST OF TABLES

Table 1: Comparison between the major properties of diesel and biodiesel	34
Table 2 : Comparison between the major properties of diesel and DME.....	36
Table 3: Comparison between the major properties of diesel and JP-8.....	37
Table 4: Comparison between diesel and waste cooking oil properties [44]	39
Table 5: Comparison between diesel and corn oil basic properties [46]	43
Table 6: Specifications of the used engine	62
Table 7: Uncertainty in the readings of measurement and the results	71
Table 8: Comparison between the normal and the three new manifolds	77
Table 9: Main properties of fuels used	79
Table 10: Main properties of used fuel blends.....	80
Table 11: Comparison between new intake manifolds at a constant engine load (Load=1N.m)	189
Table 12: Comparison between new intake manifolds at a constant engine speed (speed=1700 rpm).....	190
Table 13: Comparison between tested fuels at a constant engine load (Load=1N.m).....	191
Table 14: Comparison between tested fuels at a constant engine speed (speed=1700 rpm)	192
Table 15: Principle gases and their allowed ranges	215
Table 16: Physical and technical specifications of the gas analyzer	216
Table 17: The technical specifications of the gas analyzer's probe.....	217
Table 18: AEROCERT 531 main features with the accompanying accessories [69].....	222
Table 19: Smoke meter's technical specifications.....	229
Table 20: Smoke meter's electrical specifications.....	230

Table 21: Smoke meter's general specifications	230
Table 22: Technical specifications of the speed tachometer	234
Table 23: General specifications of the used speed tachometer	234

LIST OF FIGURES

Figure 1: An induction manifold used with 4-cylinder engine [1]	2
Figure 2: Single point fuel injection used in spark ignition engine [4]	3
Figure 3: Direct injection diesel engine [5]	4
Figure 4: Standard intake manifold design for diesel engine. [6].....	5
Figure 5: Helical, spiral and helical spiral manifold [10].	15
Figure 6: Load vs. b.s.f.c [17].....	19
Figure 7: Load vs. volumetric efficiency [17]	20
Figure 8: Load vs. brake mean effective pressure [17].....	21
Figure 9: Load vs. exhaust gas temperature [17].....	22
Figure 10: Load vs. hydrocarbons [17].....	24
Figure 11: Load vs. carbon monoxide [17].....	25
Figure 12: Load vs. NOx emissions [17].....	26
Figure 13: Conceptual schematic of conventional diesel combustion [21].	31
Figure 14: Average emission impacts of biodiesel fuels in compression ignition engines [38]...	35
Figure 15: Specific fuel consumption at various engine loads for diesel and diesel-waste cooking oil blend [44].....	40
Figure 16: Brake thermal efficiency at various engine loads for diesel and diesel-waste cooking oil blend [44].....	41
Figure 17: Mechanical efficiency at various engine loads for diesel and diesel-waste cooking oil blend [44]	42

Figure 18: Comparison between diesel and diesel-corn oil fuel blend in bsfc at a constant engine load [46]	44
Figure 19: Comparison between diesel and diesel-corn oil fuel blend in CO emission at a constant engine load [46]	45
Figure 20: Global transportation demand by fuel [51]	47
Figure 21: Well-to-wheel emissions reductions from transport in 4DS and 2DS [51].....	47
Figure 22: Comparison of theoretical results and experimental data for effective power [55]	50
Figure 23: Comparison of theoretical results and experimental data for effective efficiency [55]	51
Figure 24: Comparison of theoretical results and experimental data for effective power [57]	53
Figure 25: Effect of ethanol addition to break specific fuel consumption [57]	54
Figure 26: Comparison between diesel, diesel –ethanol blend and the blend of diesel-ethanol and DME in particulate matter emission [58].....	55
Figure 27: Comparison between diesel, diesel –ethanol blend and the blend of diesel-ethanol and DME in smoke density [58].....	55
Figure 28: Comparison between diesel, diesel –ethanol blend and the blend of diesel-ethanol and DME in nitric oxides emission [58].....	56
Figure 29: Variation of brake thermal efficiency with load for different butanol blends percentage [59].....	57
Figure 30: Variation of carbon monoxide emission with load for different butanol blend percentage [59].....	58
Figure 31: Schematic of the test rig [60]	61
Figure 32: Dynamometer coupled to the engine with the crankshaft	63

Figure 33: Control board that has the main switches.....	65
Figure 34: Fuel tanks	66
Figure 35: Fuel calibrated burette	67
Figure 36: Fuel supply valves.....	67
Figure 37: Air tanks	68
Figure 38: Induction manifold with one normal manifold diameter and 30 degree outlet angle.	74
Figure 39: Induction manifold with doubled inlet diameter and 30 degree outlet angle.....	75
Figure 40: Induction manifold with tripled inlet diameter and 30 degree outlet angle	76
Figure 41: Pressure vs. crank angle position for normal and new manifolds using GTL fuel at a speed=1700rpm and a load =1N.m	84
Figure 42: $Dp/d\theta$ vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	86
Figure 43: Peak pressure vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	89
Figure 44: $Dp/d\theta$ vs. engine Load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	92
Figure 45: Peak pressure vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	94

Figure 46: Brake specific fuel consumption vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	97
Figure 47: Volumetric efficiency vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	100
Figure 48: Exhaust temperature vs. engine speed for:(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	102
Figure 49: Brake specific fuel consumption vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	105
Figure 50: Volumetric efficiency vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	108
Figure 51: Exhaust temperature vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	110
Figure 52: Carbon monoxide vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	113

Figure 53: Carbon dioxide vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend..... 116

Figure 54: Hydrocarbons emission vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend..... 119

Figure 55: Nitric oxide emission vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend..... 122

Figure 56: Particulate emission vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend..... 125

Figure 57: Smoke emission vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend..... 127

Figure 58: Carbon monoxide vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend..... 129

Figure 59: Carbon dioxide vs. engine load for: (a) diesel fuel, (b) GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend..... 131

Figure 60: Hydrocarbons emission Vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	134
Figure 61: Nitric oxide emission vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	137
Figure 62: Particulate emission vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	140
Figure 63: Smoke emission Vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend.....	142
Figure 64: Pressure vs. crank angle position for six different types of fuels using the 1D manifold at a speed=1700rpm and a load =1N.m	144
Figure 65: $Dp/d\theta$ vs. engine speed for different types of fuel using the 1D manifold	146
Figure 66: Peak pressure vs. engine speed for different types of fuel using the 1D manifold ...	148
Figure 67: $Dp/d\theta$ Vs. engine load for different types of fuel using the normal manifold	150
Figure 68: Peak pressure Vs. engine load for different types of fuel using the normal manifold	152
Figure 69: Brake specific fuel consumption vs. engine speed for different types of fuel using the 1D manifold	154
Figure 70: Volumetric efficiency Vs. engine speed for different types of fuel using the 1D manifold	156

Figure 71: Exhaust temperature vs. engine speed for different types of fuel using the 1D manifold	157
Figure 72: Brake specific fuel consumption vs. engine load for different types of fuel using the normal manifold.....	159
Figure 73: Volumetric efficiency vs. engine load for different types of fuel using the normal manifold	161
Figure 74: Exhaust temperature vs. engine load for different types of fuel using the normal manifold	163
Figure 75: Carbon monoxide emission vs. engine speed for different types of fuel using the 1D manifold	165
Figure 76: Carbon dioxide emission vs. engine speed for different types of fuel using the 1D manifold	167
Figure 77: Hydrocarbons emission vs. engine speed for different types of fuel using the 1D manifold	169
Figure 78: Nitric oxide emission vs. engine speed for different types of fuel using the 1D manifold	171
Figure 79: Total particulate emission vs. engine speed for different types of fuel using the 1D manifold	173
Figure 80: Smoke emission vs. engine speed for different types of fuel using the 1D manifold	174
Figure 81: Carbon monoxide emission vs. engine load for different types of fuel using the normal manifold	176
Figure 82: Carbon dioxide emission vs. engine load for different types of fuel using the normal manifold	178

Figure 83: Hydrocarbons emission vs. engine load for different types of fuel using the normal manifold	180
Figure 84: Nitric oxide emission vs. engine load for different types of fuel using the normal manifold	182
Figure 85: Total particulate emission vs. engine load for different types of fuel using the normal manifold	184
Figure 86: Smoke emission vs. engine load for different types of fuel using the normal manifold	186
Figure 87: GW-Instek oscilloscope	205
Figure 88: How the waveform is displayed on the oscilloscope's screen	206
Figure 89: PCB pressure transducer and PCB charge converter	209
Figure 90: The pressure transducer used in this experiment.....	210
Figure 91: Pressure sensor calibration	211
Figure 92: HM5000 handled gas analyzer	213
Figure 93: Met one instruments Aerocet-531 mass particle counter dust monitor.....	221
Figure 94: Front side of the smoke meter	227
Figure 95: Rear side of the smoke meter	227
Figure 96: Testo 465 speed tachometer used to detect engine speed	233
Figure 97: Calibration certificate for the pressure transducer	236
Figure 98: Calibration certificate for the gas analyzer.....	237
Figure 99: Calibration certificate for the particulate meter	238

CHAPTER 1. INTRODUCTION

The main purpose of this chapter of the report is to give an overview about the suggested topics, which are covered on this thesis project. It is written to justify the need for this work and to determine the motivation behind this research. Hence, it starts by giving a general background about these topics and then it presents the main objectives of this thesis.

1.1 Background

It is claimed that there are several parameters that controls the engine thermal efficiency such as brake mean effective pressure, break thermal efficiency, volumetric efficiency, break specific fuel consumption and air-fuel mixing rate. One unique way to enhance those performance parameters is by changing the design or geometry of the induction manifold. In addition, the design of the induction manifold can play a vital role in decreasing emission gases quantities. Section 1.1.1 is dedicated to give an overview about the design of the induction manifold and its role in the engine and what is new to be added on this project.

As the oil crisis has begun in 1970, this has resulted in a high increment of crude oil prices in addition to a reduction of its supply. Consequently, this has forced the researchers and nations to start looking for other types of fuels that can replace conventional fuels called 'Alternative fuels'. These alternative fuels can replace conventional Diesel and Gasoline and they can be used in a pure form as will be discussed in section 1.1.2 or they can be blended with conventional fuels as section 1.1.3 discusses. The use of alternative fuels will

reduce the need for conventional fuels. Moreover, it is to be used to enhance engine performance and reduce emission.

1.1.1 Induction Manifold's Design

One unique approach to enhance air-fuel mixing rate is by changing the design or geometry of the induction manifold. An induction manifold can be characterized as a collection or arrangement of pipes or tubing with several outlet sections through which the incoming air to the combustion chamber is assembled or distributed as it is shown on Figure 1.



Figure 1. An induction manifold used with 4-cylinder engine [1]

The manifold might be a fabrication or a casting that is composed from a relatively light weight material. Manifolds are generally recognized by the function provided, as the

exhaust manifold and the intake manifold on the internal combustion engines. A few types of manifolds are used for handling oil, water, and different liquids such as engine exhaust gases which are usually called headers. Both the intake and the exhaust manifold are an integral component of multi-cylinder engine compartment and necessary for its operation at any internal combustion engine. [2]

The intake manifold is mainly a casting or a collection of pipes in which air-fuel mixture flows from the engines throttle valves to its intake valve ports in the cylinder block or cylinder head. In a gasoline engine, both air and fuel are mixed together in a specified ratio to initiate the combustion process. The fuel that is injected into the cylinder is to be mixed with air using throttle-body fuel injection and a carburetor in the past as it appears in the diagram of Figure 2. [3]

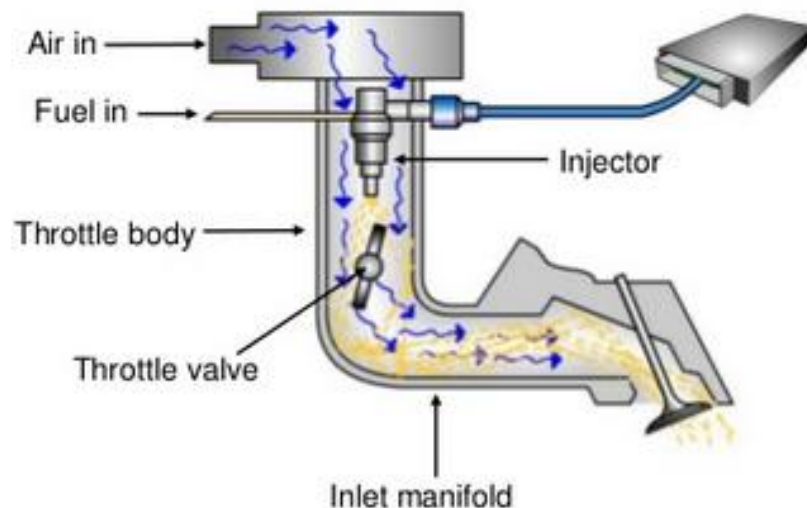


Figure 2. Single point fuel injection used in spark ignition engine [4]

In compression ignition (Diesel) engine, fuel is injected alone through ports. Air is inducted into the cylinder because of the pressure drop between the atmosphere and inside the cylinder. For that reason, air should be inducted into the cylinder by the minimum pressure drop possible. In a direct injection diesel engine, fuel is injected directly into the cylinder and it is sprayed into a combustion cup located in the top of the piston. Figure 3 shows an anatomy for a direct injection diesel engine.

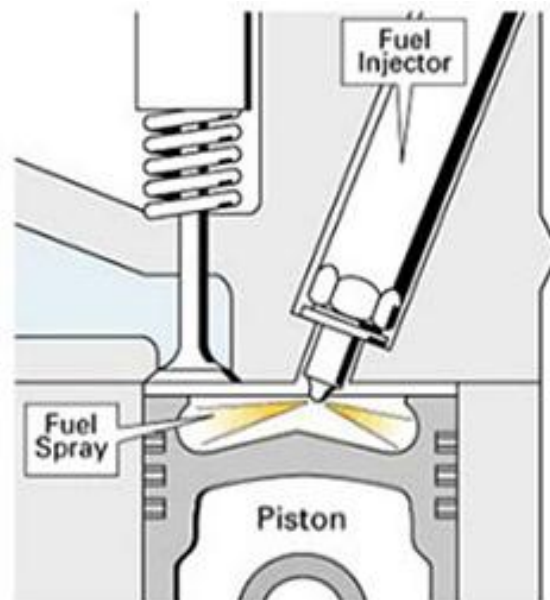


Figure 3. Direct injection diesel engine [5]

In this context, the intake manifold's role is to distribute the air in diesel engine or the air fuel mixture in the gasoline engine uniformly to each cylinder in addition to its assistance

in the vaporization of fuel droplets. The most common intake manifold for diesel engine would look like as shown in Figure 4.



Figure 4. Standard intake manifold design for diesel engine. [6]

The purpose of this research is to develop a new design for the induction manifold and to test this new design experimentally on a diesel engine. The results obtained with the new designs are to be compared with the standard normal manifold to study any improvement in the performance and the emission rates of the engine.

1.1.2 Alternative Fuels

As the fossil fuels are being consumed in everyday life in an increasing rate for different purposes, this has led the researchers to start thinking about new alternative energy resources. The current transportations are based on the use of petrol and diesel which can

be supplied in a limited quantity. Also, the big increase in population around the world and the new life style and living standards in both developing and developed countries all have caused a quick depletion of fossil fuels. Greenhouse gases (GHGs) that are produced from fossil fuels are one of the main causes of global warming which can cause several environmental problems and agricultural disorder.

Some other reasons such as the increasing demand for oil and the fluctuations in fuel prices have grabbed the attentions of scientists and researchers to put their efforts on finding alternative fuels that can be renewable and less harmful in terms of emission levels for the environment. Furthermore, the discovery of alternative fuels can help the humankind to be more energy dependent. The use of alternative automotive fuels can be considered as a safe and clean energy resource which can play a vital role in enhancing the air quality and dependency on non-renewable conventional fuels.

Vehicles that do not run on conventional fuels are named alternative fuels vehicles (AFVs). A significant progress has been made on the last few years to develop such vehicles that can run on alternative fuels such as Ethanol, bio-diesel, Hydrogen and Natural gas, which can thus produce less emission. Consequently, the researches on the production of fuels derived from biological renewable feed stocks have been greatly intensified over the past decade. The role of this project is to experimentally test some alternative fuels such as GTL (gas to liquid) fuel and to compare the effect of using them in comparison with conventional diesel fuel.

1.1.3 Fuel Blends

Fuel blends which are mixtures between two or more fuels can be considered one of smart alternatives for diesel fuel in compression ignition engines. By using them instead of pure diesel, they can decrease the energy consumption and reduce the environmental pollution in order to meet the demand of the current tough emission worldwide legislations. In the recent years, biofuels have attracted a lot of attention as they are clean and renewable fuels especially biodiesel, ethanol and dimethyl ethers

When compared with diesel fuel, biofuels have a lot of advantages over pure diesel as they have more oxygen content and less aromatic hydrocarbon composition as well as sulfur in the emission [2]. For that reason, it is essential to study the effect of using diesel engine fueled with biofuels. These types of fuels can be used as a direct substitution for diesel fuel or can be mixed with it to enhance the engine performance and reduce the emissions.

This project emphasis on the concept of fuel blends as it experimentally test some fuel blends samples namely; diesel-GTL fuel blend, diesel-waste cooking oil fuel blend and diesel-GTL-Waste cooking oil fuel blend. The results are compared with those of pure diesel fuel in terms of combustion characteristics and emissions.

1.2 Research Objectives

This work aims to study the effect of using different designs of induction manifolds that have some modifications apart from the standard normal manifold geometry such as the outlet angle and the inner diameter. One more objective of this research is to experimentally test the effect of using some selected alternative fuels and fuel blends as a substitute for diesel fuel in

compression ignition engine. The main objectives of this work can be summarized in the following points:

- 1) To develop new designs of induction manifolds for diesel engine that can enhance fuel-air mixing quality
- 2) To prepare and characterize a selected number of alternative fuels.
- 3) To diagnose the effect of using the new induction manifold designs on the combustion characteristics and emissions of the engine.
- 4) To study the effect of using the selected blends on the engine performance and emissions while utilizing the new induction manifolds.
- 5) To compare the performance of all used induction manifolds as well as all fuel blends.

1.3 Organization of Thesis

Chapter 1: Introduction

This chapter introduces the research background and presents a literature survey about engine's induction manifold, alternative fuels and fuel blends .It presents the main objectives for thesis experiments and contains the outlining of the thesis organization.

Chapter 2: Literature Review

This chapter further adds to the literature survey as it goes through the historical background for the induction manifold and for the alternative fuels. It presents the past designs for the intake manifold and how far they performed better in comparison with the

standard normal manifold. Moreover, it gives some examples for previously used alternative fuels and fuel blends and it compares them to conventional diesel fuel in terms of performance and emission criteria. Above that, it mentions the most important performance and emission criteria that are used in this research to compare between various induction manifolds and fuels.

Chapter 3: Experimental Setup and Procedure

This chapter aims to introduce the engine test bed's different components and how they are linked to each other. It describes how various measuring devices are connected to the engine and how data is acquired. Furthermore, it gives a description for each measuring device and explains how it was calibrated. In addition, the general features and specifications are listed. Moreover, it discusses the principle of operation for those devices. Finally it presents the safety procedures which should be followed when operating experiments in the laboratory.

Chapter 4: Results and Discussion

This chapter expresses the data that was acquired and measured during the experiments in the form of analytical plots and graphs. These graphs are studied and analyzed as it present and compare the difference in performance between tested manifolds and fuels. In addition to that, this chapter discusses the behavior of these graphs and finds a relation between interrelated components.

Chapter 5: Conclusion & Recommendations

The conclusions are summarized, and the significant findings are highlighted in this chapter, and further research is proposed for any future work.

CHAPTER 2. LITERATURE REVIEW

This chapter presents the previous designs of the intake manifolds (section 2.1) and it compares their performance and emission characteristics with that of the standard intake manifold (section 2.2). Moreover, it gives some examples for previously used alternative fuels (section 2.3) and fuel blends (section 2.4) and it compares them with conventional Diesel fuel in terms of engine performance and emissions.

2.1 Induction Manifold

The intake manifold is a connection between the intake system of the engine to intake valves and through which the mixture of air or air-fuel is drawn into the engine's cylinders. Intake manifolds comprise regularly of a plenum, to the inlet of which bolts the throttle body, with the individual runners feeding every cylinder. The general design criteria are: a low resistance for air flow ; fair distribution of fuel and air between cylinders; runner and branch lengths that take the advantage of tuning effects and ram; adequate (however not over the top) warming to guarantee sufficient fuel vaporization with carbureted or throttle-body injection engine [7]. In addition, the induction manifold plays a major role in ensuring a sufficient amount of air-fuel mixing for combustion in the cylinder. For the diesel engine, the induction manifolds play a major role in creating strong induction swirl inside the cylinder to enhance the fuel-air mixing quality.

The air pressure that is generated into each cylinder varies during the intake process as a result of the variability of speed of every cylinder, valve open zone variation, and the unsteady gas stream impacts that outcome from these geometric variation. The mass of air

that is induced into the cylinder, and henceforth the volumetric efficiency, is totally determined by the pressure level in the intake port during the short time frame that precedes the intake valve closure. [8]

During induction process, and as the piston reaches its highest speed, the pressure instantly upstream of the valve achieves its lowest value. This starts a refraction wave that travels upstream in the inlet duct to be reflected as a compression wave at its open end. Tuning happens when this compression wave comes back at the valve when it is closing (IVC). The conduct of the inlet duct can be displayed as that of a Helmholtz resonator or even, in straightforward cases, as a quarter-wave pipe [9].

Margary et al. [9] researched the impact of intake duct length on the volumetric efficiency and in the flow field of the cylinder of a single cylinder four stroke DI Diesel engine, motored at a speed of 1000–3000 rpm. They considered three lengths of straight duct upstream of the helical inlet port of the engine, and reported estimations of immediate mass flow rate and pressure drop over the port as a function of duct length and engine speed. The outcomes demonstrate a large increment of swirl velocity values as resonance of the induction system is achieved.

The variation of the pressure created by the pulsating stream can be utilized to enhance the intake pressure level by configuring the intake manifold improving the pressure waves in the intake system. In this way, a static intake manifold must be enhanced for one particular rpm, so it is useful to build up a strategy to fluctuate the intake length and on the other hand volume. Recently, the manufacturing of a variable length induction manifold has become more popular, which enhances torque conveyance at low speed without harming high speed

power. Most outlines utilize two intake manifolds with variable length on account of the trouble of delivering a continuously variable size induction manifold. Of these manifolds, the shorter one is utilized for high-rpm, and the longer one for low-rpm.

2.1.1 Induction Manifold's Alternative Designs

2.1.1.1 Swirl Induction

Swirl is one of the methods that are used to ensure rapid mixing between the mixture of fuel and air in Diesel engine. The swirl level at the end of the compression stroke does depend on the swirl produced during intake stroke and on the amount amplified among the compression stroke. In Direct Injection (DI) diesel engine, as fuel is injected, the swirl converts over it far from the fuel injector making the ambient air is accessible for the fuel going to be injected. The induction swirl is produced either by utilizing directed ports or by pre swirling the approaching flow by utilization of a spiral or helical or helical-spiral induction manifold. Helical manifolds are more compact than typical normal manifold. They are able to create more swirl than directed ports can do at low lifts; however at higher lifts they are inferior [10].

Parameters like manifold and combustion chamber configuration, engine speed (Chen et al., 1998) [11] specifically impact the swirl in DI Diesel engines. Optimization of swirl concept becomes an important issue in the design of the intake systems in Diesel engines.

Bugrake (1981) [12] displayed a flow model to foresee the turbulence and swirl vortices in an open chamber cup in piston engine. Those models have been compared with the

experimental work that has used different intake manifold designs and combustion chamber configurations and it was found that they are in agreement with each other.

Akira et al. (1990) [13] exhibited an exploratory examination for turbulence inside the combustion chamber for Diesel engine of direct injection. It has been found that engine speed, piston bowl shape and the intake manifold design plays a major role in varying the flow fields in diesel engine.

Subsequently, the previous research about the new intake manifold design shows how it plays a major role on affecting the swirl speed at TDC and during intake and compression stroke. In addition to that, it varies the turbulent kinetic energy as well as the volumetric efficiency up to an engine speed of 3000rpm.

2.1.1.2 Previous Designs

From previous researches which were done on manifold design, it can be said that the design of the inlet manifold is very important on the IC engine. Hence, this information is very important on the study of the effect of using a spiral, helical, or a helical-spiral manifold design on the induced average swirl speed in the piston bowl at TDC, swirl ratio during suction and compression stroke, variation in turbulent kinetic energy and volumetric efficiency at engine speed up to 3000 rpm. Figure 5 shows those new intake manifold design shapes.



Figure 5. Helical, spiral and helical spiral manifold [10].

In this view, an experimental study has been conducted on a four stroke, single-cylinder diesel engine to study the effect of using the new manifold designs on the engine performance and emission. It was concluded that using the helical-spiral manifold results in a higher engine performance and lower exhaust emission in comparison to the normal, helical and the spiral manifold. Consequently, in this project the new intake manifolds are designed using a helical- spiral manifold's shape.

2.1.2 Simulation of the Fluid Motion on the Combustion Chamber

The fluid motion in the internal combustion engine is induced during the induction period and later modified during the compression stroke. The incoming charge enters the chamber by going through the intake manifold. The incoming fluid has a stored kinetic energy that will result in turbulence causes rapid mixing between air and fuel, if the fuel is injected straightforwardly into the cylinder. In-cylinder fluid motion controls the flame propagation in the SI engines, and governs the air-fuel mixing and premixed burning in the Diesel

engine. Subsequently, it is particularly fundamental to comprehend the in-cylinder flow motion comprehensively in request to upgrade the combustion chambers for the present day internal combustion engines like gasoline direct injection (GDI), homogeneous charge compression ignition (HCCI) engines and so forth.

Heywood (1998) [7] has expressed that generating a remarkable swirl and/or tumble motion in the intake stroke was one of the promising approaches to get high in-chamber turbulent intensity.

Valentino et al (1993), Reeves et al (1999), Li et al (2001), Yasar et al (2006) and Stansfield et al (2007) [14] have used PIV technique with different engines, and reported that the incoming flow structure changes considerably across the cylinder length due to the geometry of the intake valve port and the tumble motion that was created during induction process.

Lee et al (1993) and Justham et al (2006) [15] have investigated that using intake ports with smaller entry angles produce more effective tumble motion than at higher entry angles.

They have noticed that the strong tumble motion brought about 15% abatement in combustion duration.

Nadarajah et al (1998) and Auriemma et al (2001) [16] have implemented empirical examinations utilizing the Laser Doppler Anemometry (LDA) procedure to diagnose the engine in-cylinder flow behavior. Their outcomes indicated that the flow structure during the intake was especially influenced by the intake valve lifts with the development of down and up flows and a solid reversed flow underneath the intake valve created during the

intake ending time. Likewise, addition of swirl into the engine was altering the structure of the flow, more specifically underneath the intake valve.

In view of the PIV studies which were done on single-cylinder engines with various manifold inclinations at different intake valve lift conditions at similar rated speed, the accompanying conclusions are drawn:

- 1) It is observed that there is a formation of reversal flow below the intake valve when using all the manifolds
- 2) When using intake valve lifts with a 0 degree intake manifold's inclination, the flow of air takes a form of jet near the exit of the intake valve, however for other inclinations, formation of jet is not common for all lifts.
- 3) It is observed that when using intake manifold with 30 degrees, there are large scaled vortex below the intake valve at all of the inclinations.
- 4) It is also noticed that the use of 30 degree manifold angle can give the highest TKE when compared to other manifold inclination at lower valve lift. The reason behind that may be due to higher flow diversion angle obtained at this manifold inclination. Consequently, at low intake manifold inclinations, the minimum friction loss occurs when the air flows in the manifold leading to the highest TKE.
- 5) The maximum TKE (Total Kinetic Energy) could be achieved when using the intake valve with a 60 degrees intake manifold for all the lifts.
- 6) Finally, it can be deduced that using these previous observations can be used widely to optimize new intake manifold's geometries and orientations, which can successfully help in manufacturing modern internal combustion engines.

It is clear from the above that the spiral-helical manifold, the manifold length and the inclination angle affect the generation of the induction swirl inside the cylinder. All these parameters have been studied individually. However, compiling all these parameters in one design is expected to have different effect on the generated swirl and turbulence. Moreover, the effect of helical diameter, as another important design parameter, has not studied before. The present investigation has considered all these parameters in developing new designs for the induction manifold utilizing the 30 degrees as the manifold outlet angle for all new designs as it gives the highest TKE at low valve lift.

2.2 Effect of Induction Manifold Designs on Engine Performance and Emission

2.2.1 Engine Performance

Brake specific Fuel Consumption

The bsfc is a tool for measuring engine efficiency. The bsfc and engine brake thermal efficiency are conversely related, so that the engine becomes better as the bsfc is lower. Figure 6 shows the relation between brake specific fuel consumption with respect to engine load for the spiral, helical and helical spiral manifold in addition to the normal manifold. Brake specific fuel consumption of various inlet manifolds looks very similar to normal manifold. Bsfc increases with load up to 0.5kW, however as load further increments from 0.5 to 3 kW. It can be seen from Figure 6 that brake specific fuel consumption for all new manifolds is less contrasted with normal manifold. It is important to note that 4.28% increment in brake thermal efficiency has been observed at 2.5kW load for helical spiral inlet manifold compared to normal inlet manifold. [17]

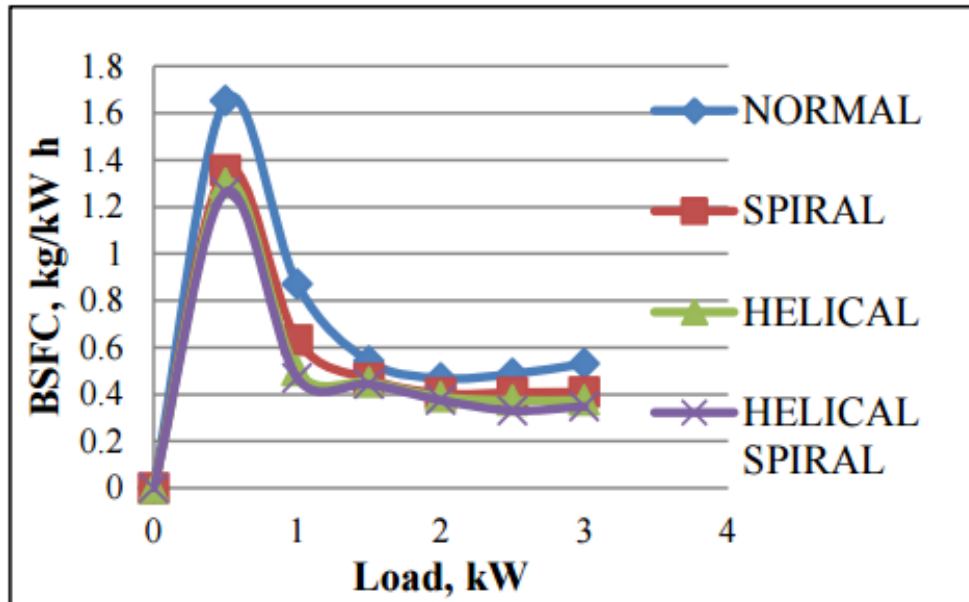


Figure 6. Load vs. b.s.f.c [17]

Volumetric Efficiency

It is essential to increase the volumetric efficiency of an engine to its maximum because the amount of the fuel that can be combusted and power generated for a given engine displacement is expanded to its maximum. The volumetric efficiency relies on the geometrical configuration of the intake manifold, valve size, lift, and timing. Despite the fact that it doesn't impact in any way the thermal efficiency of the engine, it will impact the efficiency of the system in which it is introduced in. Plainly, heavier engines used in a vehicle result on a reduction of fuel economy. The variation of volumetric efficiency with load for normal and new manifolds is shown on Figure 7. It might be noticed that, the volumetric efficiency is the highest for helical manifold and least for normal manifold and in the middle of these two spiral and helical-spiral at a given load. [17]

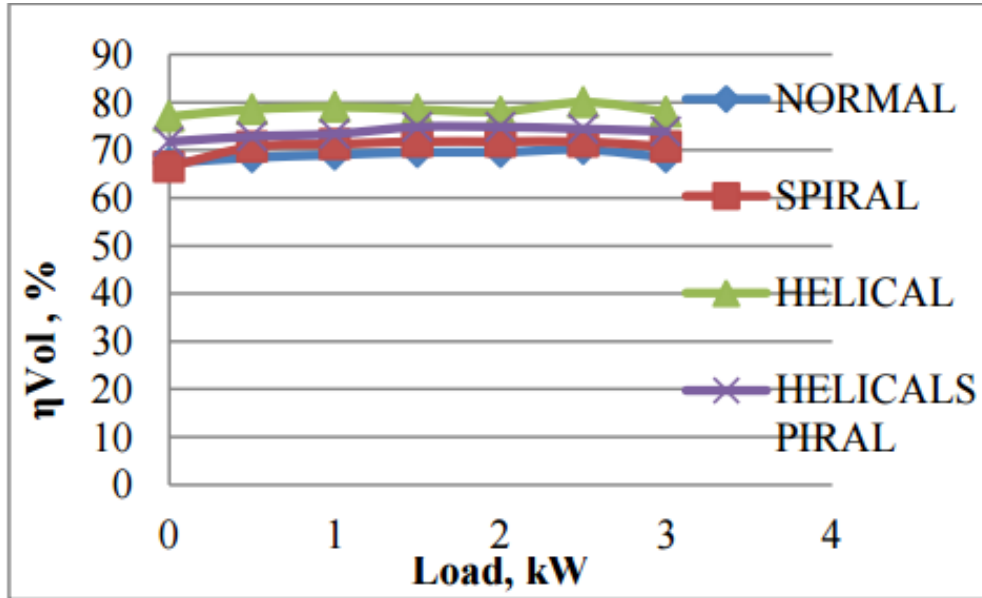


Figure 7 .Load vs. volumetric efficiency [17]

Volumetric efficiency for normal manifold at 2.5kW load is 70.147% and for spiral, helical and helical-spiral is 71.79%, 80.02% and 74.46% respectively. Volumetric efficiency is somewhat increased for all new new manifolds contrasted with normal manifold. The helical manifold having the highest volumetric efficiency contrasted with all other inlet manifolds at an efficiency of 80.02%. It is essential to note that 9.873% of increased in volumetric efficiency observed at 2.5kW load for helical spiral inlet manifold contrasted with normal manifold.

Brake Mean Effective Pressure

Figure 8 demonstrates the variation of Brake mean effective pressure concerned at various loads. The brake mean effective pressure is considered as an indication of external shaft

work per unit displacement volume exerted by the engine. Brake mean effective pressures were higher for new intake manifolds than normal manifold. [17]

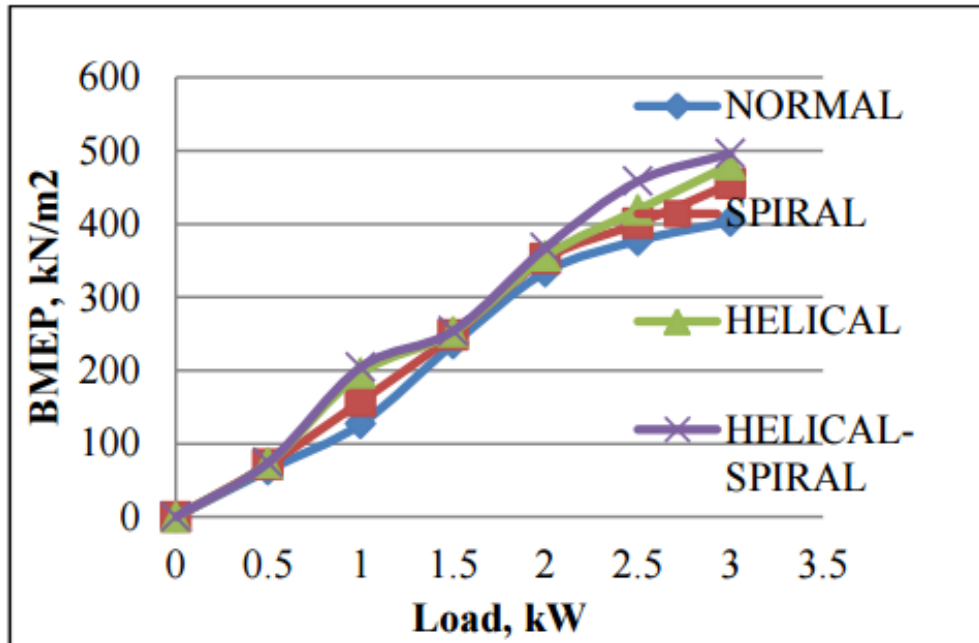


Figure 8. Load vs. brake mean effective pressure [17]

The values for brake mean effective pressure at 2.5kW of helical, spiral and helical-spiral manifolds are 400.63, 419.30, 458.22 kN/m² whereas it is 377.05 kN/m² for normal inlet manifold. The increment in brake mean effective pressure may enhance the output power and decrease exhaust emanation. It is important to note that the use of the helical-spiral manifold has increased the brake mean effective pressure by 81.17kN/m² at 2.5KW when compared to the normal manifold.

Exhaust Gas Temperature

Figure 9 demonstrates the variation of exhaust gas temperature for spiral, helical, helical-spiral and normal manifold at various loads. Exhaust gas temperature is an indication for transformation of heat into work that happens inside the cylinder. The exhaust gas temperature is higher for helical, spiral and helical-spiral than the normal manifold. At different load conditions it is observed that the exhaust gas temperature increments with load since more fuel is combusted to meet the power required. It can be noted that in the case of normal manifold operation the exhaust gas temperature is 217 C at 2.5kW load. [17]

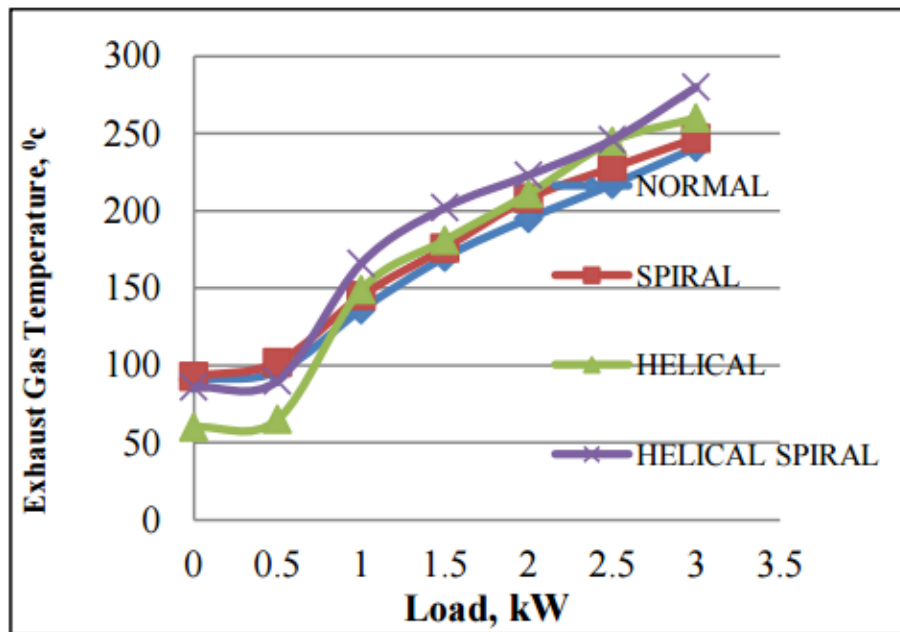


Figure 9. Load vs. exhaust gas temperature [17]

For helical, spiral and helical-spiral inlet manifolds exhaust gas temperature quietly increments to 228, 245 and 246 0C respectively. The exhaust gas temperature is higher for helical spiral manifold which is 281 0C at 3kW load.

2.2.2 Engine Emissions

Hydrocarbons

Figure 10 demonstrates the variation of hydrocarbons with regard to load for inlet manifolds that were tested. Unburned hydrocarbon emissions are brought about by incomplete combustion of fuel air blend. HC emissions shift from no load to full load and unburned hydrocarbons are higher in the case of spiral manifold contrasted with normal manifold, however it is less in the case of helical and helical-spiral manifold. The estimations of unburned hydrocarbons of spiral, helical and helical-spiral manifolds for steady speed at 2.5kw load are 46, 24 and 22 ppm when contrasted with 27 ppm of normal manifold. The plausible explanation behind this emission might be some bit of the fuel-air mixture in the burning chamber comes into direct contact with combustion chamber surface wall and get extinguished. [17]

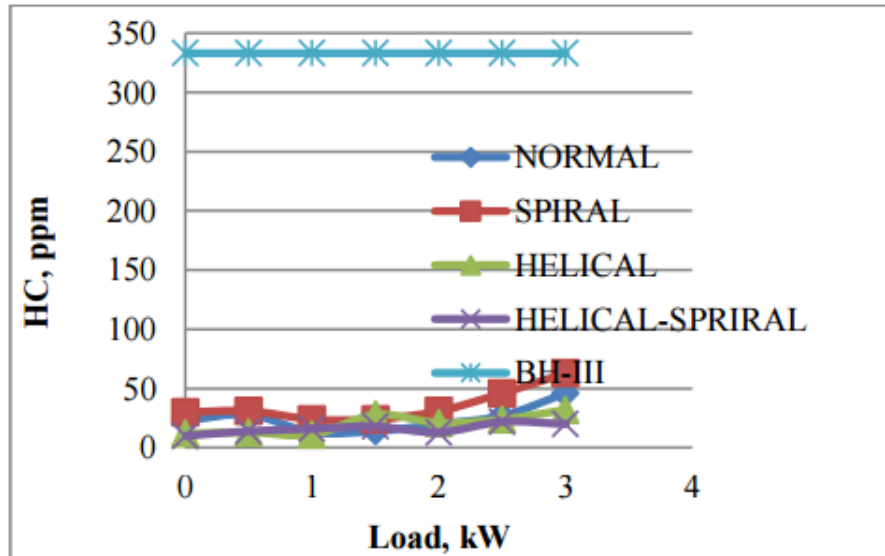


Figure 10. Load vs. hydrocarbons [17]

Some of this extinguished fuel-air mixture is constrained out during the exhaust which adds to the high HC outflow from the results, it can be seen that the concentration of hydrocarbon of helical-spiral manifold is somewhat lower than normal manifold. Assist it can be noted from Figure 10 that emissions of the engine HC are far underneath the passable levels of according to BS-III standards at all the loads.

Carbon Monoxide

Carbon monoxide is produced as the result of deficient combustion. From Figure 11, the variation of carbon monoxide concerning load can be observed as the load increases the CO emission is increased. CO emissions are of smaller amount at low load and high at full load for normal manifold contrasted with different manifolds. It can be noticed that CO emissions are diminished in the case of helical-spiral manifold until the load of 2kW. [17]

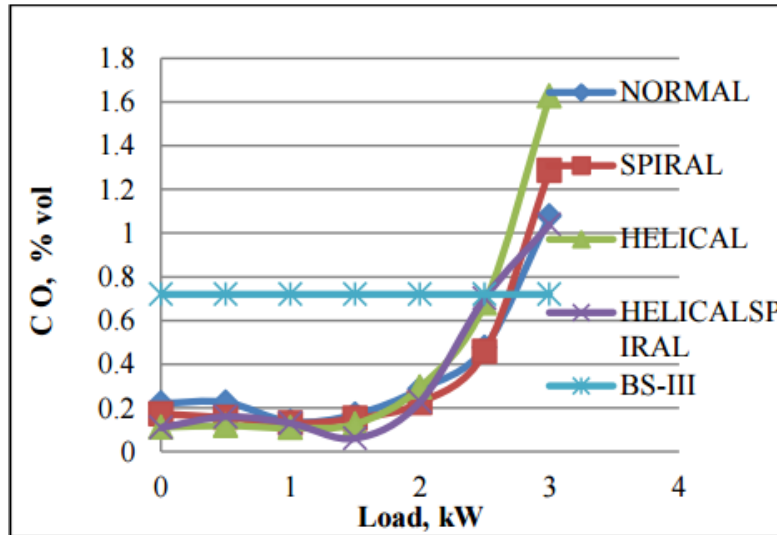


Figure 11. Load vs. carbon monoxide [17]

The reason for increased CO emission might be due to incomplete combustion. The greatest CO emission was noticed at the full load 3kW. The estimations of carbon monoxide of helical, spiral and helical-spiral at load 2.5kW are 0.457, 0.68, 0.742% by volume respectively, whereas the esteem is 0.447% by volume for normal manifold at 2.5kW load. From the chart, it can be gathered that at all loads which are below 3kw, the CO emission of the engine is in the allowable norms of confinement according to BS-III standards and beyond which it is more than the permissible standards. Henceforth it is recommended to run the engine at loads beneath 3kW load.

Nitrogen Oxides

Figure 12 depicts the oxide of nitrogen from the engine exhaust at various loads. NO_x results because of interaction of Nitrogen and oxides at moderately high temperature. NO is the major segment in the NO_x emission. As the load is increased, the oxides of Nitrogen

emission becomes larger. The oxides of nitrogen were higher for helical and spiral manifold at lower loads, However when the load increases the emissions were less for all new manifolds in comparison with normal manifold. The estimations of NO_x of helical, spiral and helical spiral inlet manifolds at a constant engine speed at 2.5kW load are 392,344 and 259 respectively with regard to 430 ppm for normal manifold. [17]

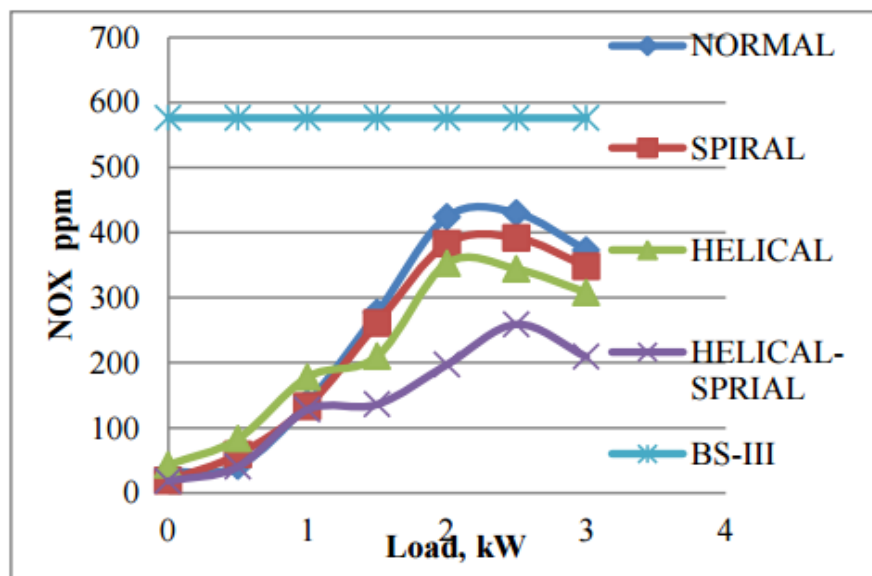


Figure 12. Load vs. NO_x emissions [17]

All the three new manifolds considered by the previous investigations yielded lower amount of emissions.

2.3 Alternative Fuels

The earth-wide temperature boost and the extreme need for energy resources are among the most imperative issues that undermine the serene presence of the humankind. More use of energy alternatives and the need for minimizing the exhaust gases amount from car engines can be a viable solution for this issue. Moreover, Population development throughout the most recent decades has prompted to enormous development in fossil energy request. Forecasts of fossil fuel fatigue continue growing, inferable from the change of penetrating innovations, and the development of substantial amounts of shale gas holds. In this manner, in spite of the development of present day and renewable energy sources, for example, atomic, sun powered, and wind energy; fuel combustion will keep on playing a critical part in the energy change field.

The two noteworthy powers that had been produced and generally utilized alongside the improvement of the ICEs and the car enterprises over the previous century are Gasoline and Diesel. The burning of gaseous fuel experiences flame spread after an underlying spark occasion lighting the homogeneous air–fuel mixture in SI motors, while the ignition of Diesel fuel is driven by the auto-ignition of the fuel presented to high temperature gas, warmed by compression in CI motors. Despite the long history of an enduring store network and the decided position of gas and Diesel as traditional car fills in the market, the scan for alternative fuels step by step began to rise back in the 1980s.

2.3.1 Alternative Fuels and their Importance

The meaning of alternative fuels may vary depending upon the specific situation. The present study characterizes alternative fuels as those other than conventional Gasoline and Diesel fuel, covering a wide assortment as far as manufacturing sources and final form. For instance, Ethanol fuel is viewed as an alternative for SI motors, paying little consideration to its unique source from either traditional raw petroleum or any renewable biomass. The alternative fuels characterized by the Energy Policy Act (EPAct) additionally cover an endless measure of non-traditional fuels, including alcohols. For example, ethanol (counting mixes with gas more than 85%); regular gas and condensed energizes locally got from normal gas; Liquefied petroleum gas (LPG); Coal to Liquid fuels (CTL); Hydrogen (H₂); Biodiesel (B100); and fuel that is generously non-petroleum that yields significant energy security and ecological advantages. The importance of alternative fuels can be ascribed to the accompanying points:

- 1) Seeking after energy supportability through the expanded utilization of those alternative fuels got from renewable energy sources and moderating the worries of restricted fossil fuel energy.
- 2) Enhancing engine productivity and its performance as the alternative fuels can enhance the physical and chemical properties of conventional fuels when they are blended.
- 3) Relieving the unbalanced utilization of conventional petroleum-based fossil energizes.

2.3.2 Considerations for Alternative Fuels

A portion of the common considerations for using alternative fuels for both SI and CI engines are listed below:

- 1) Chemical properties, for example, octane and cetane number.
- 2) Physical properties (splash or blend formation for ignition, and engine operability over an extensive variation of temperatures).
- 3) Lower Heating Value (LHV).
- 4) Compatibility (counting approval by engine and vehicle makers and expenses).
- 5) Manufacturing expense and foundations.
- 6) Volatility
- 7) Sulfur content

The combustion properties straightforwardly indicate whether or not the given alternative fuel is suit-capable for engine operation. The physical properties are additionally essential, as they decide the development of flammable blend. Moreover, Octane and Cetane number play a vital role in determining the timing of ignition and its delay. The LHV decides the viability of the fuel as a vitality transporter. The practically identical level of LHV to traditional fuels is favored; otherwise penalties in using fuel may get to be distinctly risky. Material similarity with current engine equipment or fuel supply framework is vital for the infiltration of alternative fuels into the market as these alternative fuels could be corrosive for some fuel system components. Something else, an extraordinary measure of extra cost would be exhausted for equipment adjustments in case the alternative fuel was not suitable for use in the engine.

2.3.3 Alternative Fuels for Diesel Engine

2.2.3.1 Fuel Requirements for Diesel Engines

Diesel fuel is the ordinary used fuel for CI engines. The particular distinction of burning in CI contrasted with SI engine is that the fuel is directly injected into the chamber, and auto-ignited because of the high encompassing temperature toward the end of the pressure stroke. Hence, the auto-ignition capacity of the fuel is critical for its utilization in CI engines. A few vital criteria that exist to quantify the nature of alternative fuels related to CI engine are [18]:

- 1) Cetane number
- 2) Boiling point
- 3) Narrow density and viscosity spread
- 4) Low aromatic compounds (particularly poly- aromatic compounds) content

The cetane number (CN) firmly decides the ignition quality of the fuel. It is likewise an essential definitive component for whether a given alternative fuel is suitable for use in a CI engine application. The fuel and air blend extensively, and shape a burnable blend during the ignition delay period. The physical deferral incorporates warming and dissipation of the fluid fuel [19], shaping a flammable blend. The chemical delay incorporates pre-start responses that separate the hydrocarbon fuel and create radicals, taken after by very exothermic response prompting to the premixed period of Diesel spray ignition. Heat of vaporization and auto-ignition temperature have a major role in these stages. Figure 13 demonstrates a theoretical model of the ordinary Diesel spray ignition

during the semi enduring period. The dispersion fire remains a separation far from the spout downstream, where the separation is alluded to as the lift-off length (LOL) [20]. Formaldehyde (HCHO), which is a marker for the cool-fire during the transient start process, is at first framed upstream of the LOL, and is devoured downstream of the LOL in the fuel-rich premixed response zone. Soot is shaped as an aftereffect of both rich the premixed response zone [21] and the hot fuel-rich center of the Diesel stream.

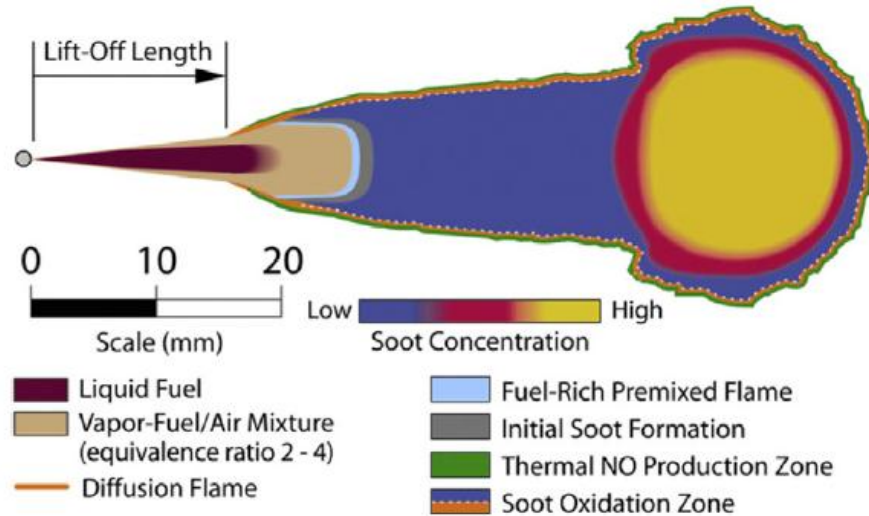


Figure 13. Conceptual schematic of conventional diesel combustion [21].

Generally, smoke discharge gained from the tail-pipe is a last aftereffect of rivalry between ash development and the oxidation procedure. The soot formation process can be divided into molecule arrangement and molecule development. The items after oxidation as well as pyrolysis from the fuel particle principally comprise of different unsaturated hydrocarbons,

especially acetylene and its higher analogs (C_2nH_2), and polycyclic aromatic hydrocarbons (PAH) [22]. These two sorts of particles are considered in all probability antecedents of residue on fire. The measure of air entrainment, which is vital for bringing down the locally fuel-rich district and smothering ash arrangement, increments with expanding LOL. The proportionality proportion at the LOL [23-25] is dictated by both engine parameters, for example, surrounding O_2 fixation and so forth. Furthermore the best possible ties of the fuel itself, for example, the auto-ignition temperature and stoichiometric air-fuel proportion.

In conclusion, alternative fuels can have different physic-chemical properties which may bring about various engine execution and engine out emissions, because of various blend arrangement forms.

2.3.3.2 Biodiesel

Biodiesel is an alternative fuel in view of bio-started feedstock. An assortment of oil feed stocks can be converted to fuels. These incorporate vegetable oils, creature fat, and waste cooking oil. Rapeseed and soybean oils are the most ordinarily utilized crude materials for biodiesel fuel. Soybean oil alone represents around 65% of the U.S. biodiesel creation in 2013 [26]. The reutilization of waste cooking oil has pulled in consideration, since it can dispense with disposal problems [27, 28]. Waste cooking oil determined Biodiesel demonstrated the most minimal GHG outflows, with around 82% GHG discharges sparing contrasted with ordinary Diesel [29]. Inclinations of feedstock vary in various nations, in light of local generation cost, ecological effect, and agrarian procedures.

The LHV of Biodiesel is not as much as that of ordinary Diesel fuel [30], while it largely has a higher cetane number [31]. Biodiesel likewise shows a higher flash point looked in comparison with conventional Diesel [32], which is beneficial for fuel stockpiling and transportation wellbeing. Biodiesel has higher cloud and pour points looked at toward ordinary Diesel, because of the higher bit of immersed unsaturated fats. Cloud and pour points demonstrate the most minimal temperatures at which a fuel can be pumped, before transforming into a wax of precious stones (crystals) [33]. Higher cloud and pour points mean impediments in cold start, and is considered as an obstacle to the utilization of mixes with huge Biodiesel portions. Biodiesel has higher density and more viscosity; also it has larger surface pressure, contrasted with conventional Diesel.

The lower heating value of Biodiesel may turn into an obstacle to accomplishing most extreme torque under full load condition. Tests performed with different soybean Biodiesel mixes from B10 to B100 demonstrated that the normal most extreme brake torques diminished by 1.57%–4.7% [34]. The indicated specific fuel consumption (ISFC) of the Biodiesel was up to 15% higher than that of the Diesel fuel, exclusively due to the lower LHV in mass premise, notwithstanding the reality of unaffected engine efficiency [35]. Table 1 compares between the major properties of diesel and biodiesel.

Table 1

Comparison between the major properties of diesel and biodiesel

Item	Diesel fuel	Biodiesel
Chemical structure		CH ₃ -O-CH ₃
Cetane number	52.8	46-64
Liquid density (kg /m ³)	0.82	0.70-0.89
Low heating value (MJ/kg)	42.5	41-42
Kinetic viscosity (at 313 K) (cSt)	2.6	4.5
Auto - ignition temperature (K)	508	
Boiling point at 1 atm (K)	450 - 643	588 - 623
Vapor pressure at 298 K (Kpa)	<< 10	0.27

The U.S. EPA delivered a report of distributed Biodiesel emissions information for substantial heavy-duty engines. Figure 14 outlines the general outcomes for CO, HC, NO_x and PM discharges [36]. A sufficient amount of CO and HC outflows was observed to be reduced which implies higher burning rates. These outcomes were expected due to the oxygenated content of Biodiesel, where more oxygen was accessible for combustion, and for diminishing the outflows in the exhaust [37]. It likewise advanced steady and finish combustion by conveying oxygen to the pyrolysis zone. The oxygen can diminish locally over-rich districts and point of confinement essential molecule arrangement.

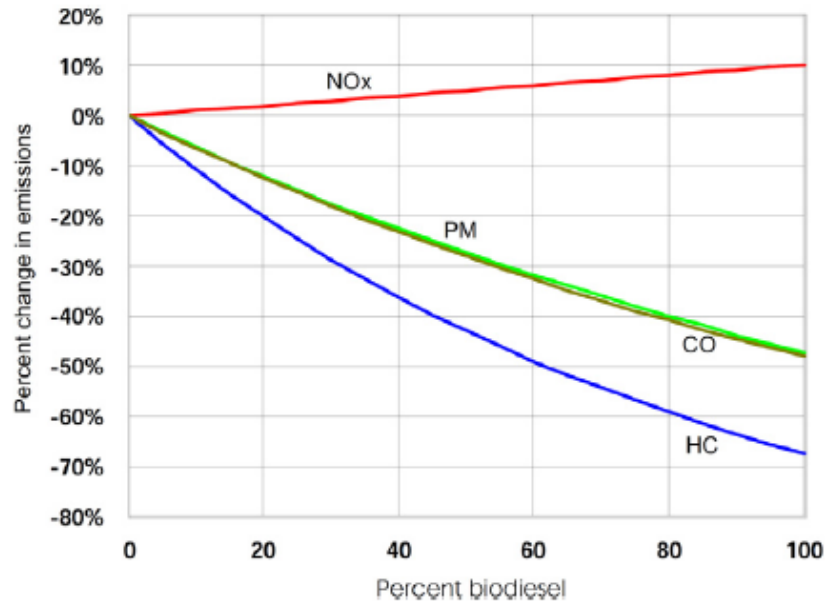


Figure 14. Average emission impacts of biodiesel fuels in compression ignition engines [38]

2.3.3.3 Di-methyl ether

DME has been utilized for quite a long time as an airborne force in the individual care industry. It has picked up consideration as a perfect alternative fuel to LPG, Diesel and gas, with physical properties fundamentally the same as those of LPG. It can be obtained from many sources, including fossil fuels (normal gas and coal) and renewable materials (biomass, squander and agricultural products) .The cetane number of DME is higher than that of Diesel, which makes it a promising alternative fuel for CI engines. The low vapor pressure at 298 K is advantageous in melting the fuel for capacity or transportation; notwithstanding, at unassumingly higher temperatures of around 80–100 °C (or 353–373

K) during steady state operation of the engine [39]. Table 2 compares between the major properties of diesel and DME.

Table 2

Comparison between the major properties of diesel and DME

Item	Diesel fuel	DME
Cetane number	52.8	> 55
Liquid density (kg /m ³)	0.82	0.667
Low heating value (MJ/kg)	42.5	27.6
Kinetic viscosity (at 313 K) (cSt)	2.6	< 0.1
Auto-ignition temperature (K)	508	523
Boiling point at 1 atm (K)	450 - 643	248.1
Vapor pressure at 298 K (Kpa)	<< 10	530

2.3.3.4 JP-8

JP-8 is a commercial fuel of kerosene-type (8–16 carbon particles for every atom) that incorporates four added substances: a static dissipater added substance, erosion in-hibitor, lubricity improver, and fuel framework icing in-hibitor. JP-8 is delivered in an unrefined petroleum refining process, which is traditional petroleum refining. Its boiling point lies between that of Gasoline fuel and Diesel fuel [40]. The cost of JP-8 is less expensive than that of Diesel fuel, on the grounds that JP-8 does not require any procedures to upgrade its cetane number. The lower fluid density of JP-8 could bring about lower most extreme

engine power, and influence specific fuel consumption [41,42]. JP-8 additionally has a lower distillation temperature, which implies unrivaled evaporation characteristics [43]. The cetane number of JP-8 ranges from 39 to 45, depending upon its production area, which is by and large lower than that of Diesel fuel. The ignition delay of JP-8 (cetane number of 38) measured in a steady volume vessel was 25%–50% higher contrasted with Diesel fuel (cetane number of 46). Table 3 compares between the major properties of diesel and JP-8.

Table 3

Comparison between the major properties of diesel and JP-8.

Item	Diesel fuel	JP-8
Auto - ignition temperature (K)	508	483
Cetane number	52.8	45
Boiling point at 1 atm (K)	450 - 643	486
Liquid density (kg /m ³)	0.82	0.79
Low heating value (MJ/kg)	42.5	43.4

2.3.3.5 Waste Cooking Oil

Fried food is exceptionally well known in India. The most general used cooking oil for frying are sunflower oil, coconut oil, palm oil as they can be easily accessed, and particularly the coconut oil which is significantly accessible in south India. It is verifiable truth that, when those are warmed for a broadened time, they experience oxidation and

offer rise to oxides. A large number of these, for example, hydro peroxides, peroxides and polymeric substances have demonstrated unfriendly wellbeing/natural impacts, for example, growth problems, dangerous medical impact in liver and kidney in addition to cell damage to various organs when they are fed to laboratory animals .[44]

Consequently, used cooking oil is a waste that results from activities mostly related to the food sector (especially restaurants), which have significantly increased in the upcoming years. The majority of this oil is disposed in an inappropriate way such as leaving it into the city drainage resulting in water pollution. However, waste cooking oil has been recently utilized as a fuel in the industrial, residential and combustion sector. [44]

It was found that waste cooking oil has very comparable properties to diesel oil such as a low flash point and viscosity in addition to a low calorific value. Table 4 compares between the physical and chemical properties of both fuels

Table 4

Comparison between diesel and waste cooking oil properties [44]

Properties	Diesel	WCO biodiesel
Chemical formula	C ₁₂ H ₂₃	C ₁₇ H ₃₁ O ₂
Viscosity (N/ms)	5.2 (at 25°C)	4.9 (at 25°C)
Calorific value (KJ/Kg)	42000	42650
Density (Kg/Kg)	834	862.6
Cetane number	46	48.7
Flash point(°C)	53	160
Sulfur contents (mg/kg)	57	8
Carbon (% w)	86.2	76.4
Ash Content (%)	0.008	0.0258

An experimental test that has been conducted on a single cylinder ,four-stroke water cooled diesel engine to compare between pure Diesel and the blend of Diesel with 10% WCO (B10),20% WCO (B20) and 30% WCO (B30) in performance and emission characteristics at a constant engine speed condition (speed = 1500 rpm). The brake specific fuel consumption for the three blends was higher than pure diesel at low and high loads as shown in Figure 15.

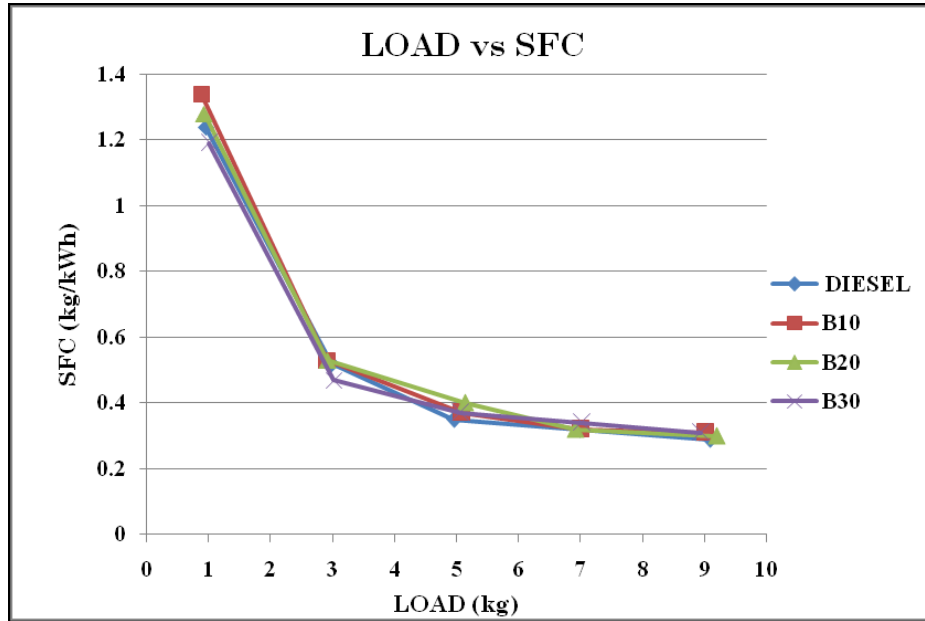


Figure 15. Specific fuel consumption at various engine loads for diesel and diesel-waste cooking oil blend [44]

Figure 16 shows the variation of brake thermal efficiency with the change in engine load. At no load condition, BTE of the three blends and diesel was the same. As the load is further increased, BTE increases because it is a function of brake power. At part load conditions, B20 has achieved a higher BTE than diesel because its calorific value is less than diesel. BTE of B10 and B30 was almost similar at part loads and lower than diesel. When the engine is operating at full load condition, BTE of the three blends was almost similar but lower than that of diesel.

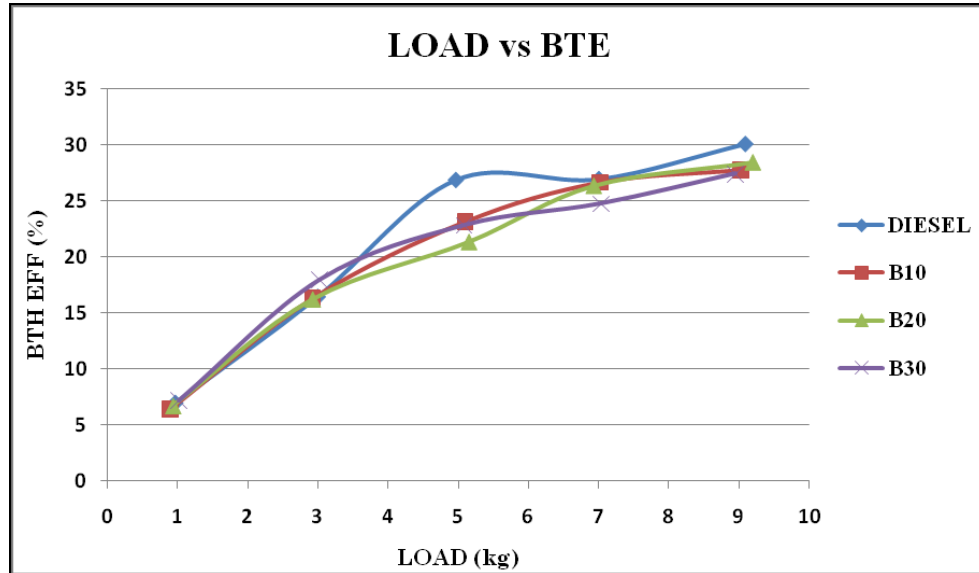


Figure 16. Brake thermal efficiency at various engine loads for diesel and diesel-waste cooking oil blend [44]

Figure17 demonstrates the variation of mechanical efficiency with the change in engine load. At low load conditions, Mechanical efficiency of the three blends was more than pure diesel. After that, the mechanical efficiency for all the fuels was increasing with load but was higher for the three blends. The use of the three blends caused the mechanical efficiency to increase more than Diesel (50.79 and 50.31% respectively). The reason behind that may be referred to the higher reaction interactivity and the better fuel spray in addition to the lower flame temperature.

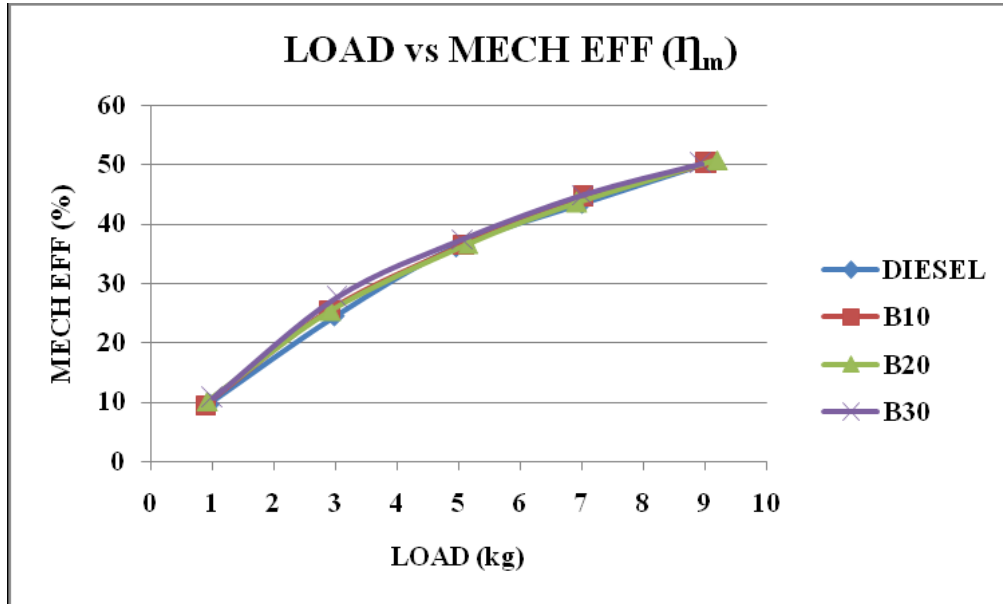


Figure 17. Mechanical efficiency at various engine loads for diesel and diesel-waste cooking oil blend [44]

2.3.3.6 Corn Oil

Corn oil is a vegetable oil used in cooking and is extracted from the germ of corn. It is used widely in frying because of its high smoke point. It is also found in some margarines as a gradient. Generally, corn oil is less expensive than other vegetable oils. Apart from cooking, corn oil is used as a feedstock for producing Diesel fuel. Also, it is used in some other industries such as paint, soap, rustproofing, inks, nitroglycerin and textiles. In addition to that, it is used for drug molecules as a carrier. [45]

Corn oil is used with diesel as a fuel blend due to its higher cetane number and oxygen content which can result in less exhaust emission rates. The use of raw corn oil as a fuel in diesel engine is considered to be harmful because it has a low volatility and high viscosity

which may cause problems for fuel injector and transforming the lubricant oil into gum in addition to piston ring sticking as it has a low volatility. In order to overcome this problem, corn oil is mixed with diesel in an advanced fuel process where the fuel properties are enhanced. Table 5 is used to compare between the properties of corn oil and diesel. [45]

Table 5

Comparison between diesel and corn oil basic properties [46]

Properties	Diesel Fuel	Raw Corn oil
Viscosity (mm ² /s)	4.3 (at 27 C)	46 (at 27 C)
Density (kg /cm ³)	815	915
Flash Point (C)	58	270 - 295
Heating Value (KJ/Kg)	43350	37825
Cetane Number	55	37.6

It was found that the use of diesel-corn oil blend can decrease the bsfc rate by an average percentage of 10.53%. This enhancements in bsfc is due to the lower heating value and the higher density of diesel-corn fuel blend in comparison with pure diesel. Figure 18 shows a comparison between diesel and diesel-Corn oil fuel blend in bsfc at a constant engine load. [46]

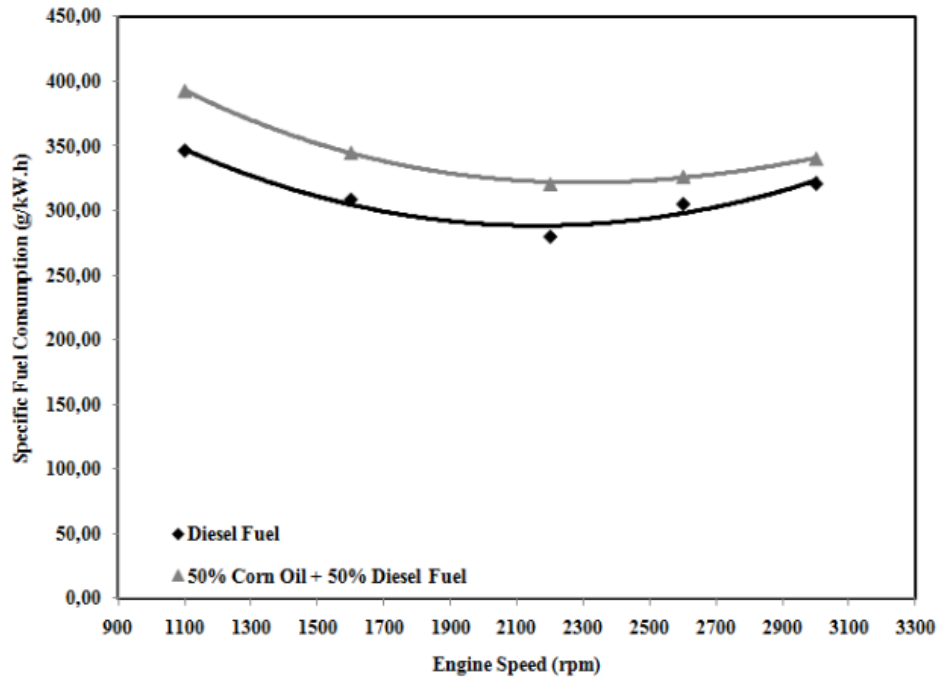


Figure 18. Comparison between diesel and diesel-corn oil fuel blend in bsfc at a constant engine load [46]

Engine design parameters such as combustion chamber design, fuel type ,engine speed, atomization rate, and air fuel ratio affect all emissions that result from IC especially CO in exhaust. From Figure 19 it was found that the use of diesel-corn fuel blend causes a reduction in CO emission by an average percentage of 35.2% due to higher oxygen content in the blend in comparison to pure diesel. [46]

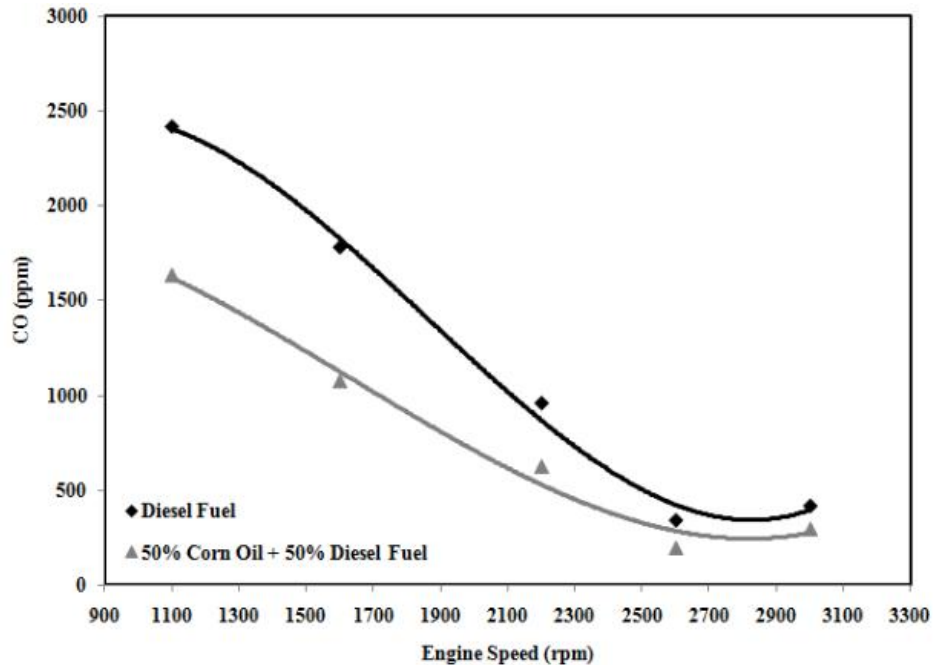


Figure 19. Comparison between diesel and diesel-corn oil fuel blend in CO emission at a constant engine load [46]

2.3.4 The Effect of Alternative Fuels on GHG Emission

The search for stable energy supplies from different energy assets together with the associated move toward renewable energies is important, in light of the fact that most energy resources as of now depend on fossil fuels, which are of limited accessibility. In spite of the projection that conventional petroleum-based fuel would in any case command transportation fills for the following 50 years [47], the craving for energy security and sympathy toward reducing greenhouse gas (GHG) emission have additionally prompted to the expanded utilization of renewable biofuels.

The utilization of fossil fuels is a noteworthy cause to CO₂ emanations, where human exercises produce around 25 billion tons of CO₂ every year [48]. The utilization of renewable biofuels can frame a CO₂ life cycle, which from a social point of view can add to aggregate CO₂ emission moderation. For instance, a full life cycle assessment (LCA) that was conducted to look at the GHG effects of conventional fuels and biofuels demonstrated that the biofuels got from vegetable oil, and also corn-and wheat-based ethanol when they are created utilizing biomass as an essential energy source, would cut GHG outflows on a well-to-wheel (WTW) premise to about half of that of conventional fuels [49]. This examination considered all the GHG outflows created from their generation, transport and capacity, and emission connected with their utilization in vehicles, subtracted by the measures of CO₂ taken in from the atmosphere by the biomass in its development stage.

Biofuels represented around 3% of the aggregate worldwide transportation fills by 2012 [50]. Figure 20 demonstrates that as the energy request in the transportation section continues expanding in the close term, the proportion of biofuel to the aggregate sum of fuel is likewise anticipated that would increment, which from a long-term point of view that may in the end prompt to impressive mitigation of the net GHG emission as shown in Figure 21.

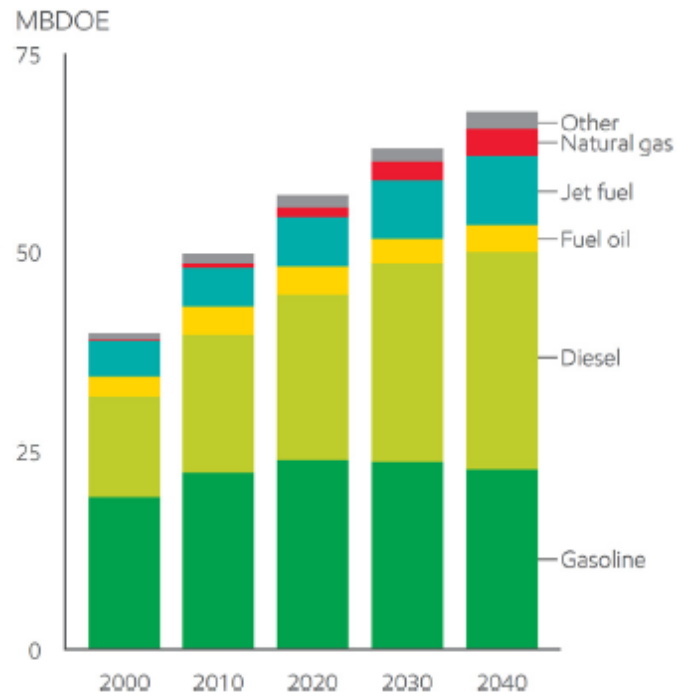


Figure 20. Global transportation demand by fuel [51]

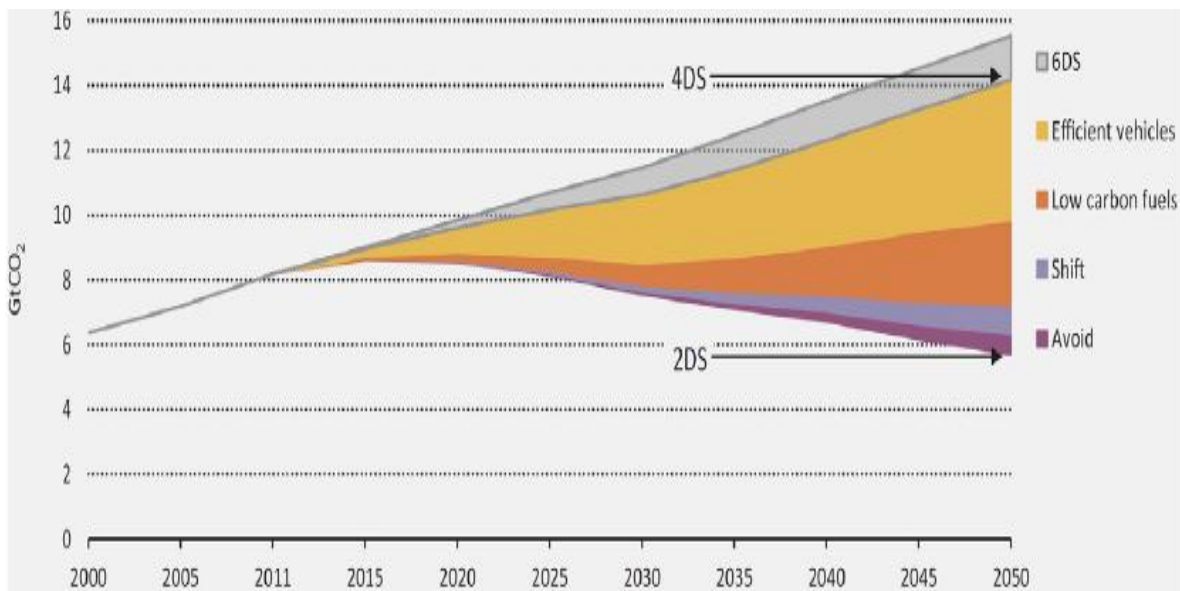


Figure 21. Well-to-wheel emissions reductions from transport in 4DS and 2DS [51]

Actualizing low-carbon transportation fuels, such as the second-generation biofuels adding to that the efficiency enhancement of vehicles, is relied upon bringing future GHG emission down to the present level in the best estimation.

From a specialized perspective, the wide utilization of renewable biofuels can likewise straightforwardly add to enhancements in engine overall performance and emission qualities. The harmony between fuel efficiency and dangerous emission, for example, unburned hydrocarbon (HC), carbon monoxide (CO), nitrogen oxides (NO_x) and particulate matter (PM), has dependably been an important issue in the engine research field. Since the burning of fuel is dictated by the air–fuel mixing, the advantages of the unique properties of alternative fuels can be profitably and specifically used during the engine combustion period. Research is being led worldwide on the utilization of alternative fuels to ICEs to uncover their possibilities.

2.4 Fuel Blends

With expanding energy utilization and natural contamination, the advancement of clean renewable fuels and modern combustion technologies to fulfill more stringent emission controls has turned into an attractive research point, especially in the field related to internal combustion engines. As of late, biofuels have gotten much consideration as spotless and renewable fuels, especially Ethanol, n-Butanol, polyoxymethylene dimethyl ethers (PODE), and Biodiesel are the most illustrative biofuels.

Biofuels have many favorable characteristics as alternative fuels to be used for Diesel engines. They are biodegradable and are gotten from organic crude materials. Besides,

they contain characteristic oxygen that declines the substance of sweet-smelling hydrocarbon and sulfur and deplete exhaust emissions. In this manner, it is fundamental to examine the combustion and discharge attributes of Diesel engines powered with biofuels. Since the vast majority of the biofuels can be utilized on the engine without alteration, they have been investigated universally as blended components or Diesel fuel substitutes for Diesel engines. [52, 53].

Many tests have been led to research the combustion and emission qualities of Diesel engines operated with biofuels. In the following subsections some of the most representative biofuels that have been used are considered on this study.

2.4.1 Diesel- Biodiesel Blend

Recently, renewable energy resources were proposed to be as another alternative to petroleum-based fuels. Biodiesel, got from vegetable oil or creature fat, is considered as an alternative renewable fuel for use in Diesel engines , Biodiesels have both points of interest .what is more, inconveniences, which can be recorded as follow, respectively. The upsides of Biodiesel as Diesel fuel, aside from their renewability, are their insignificant sulfur and low aromatic content, higher flash point, higher lubricity, higher cetane number and higher biodegradability and non-toxicity. Likewise, Biodiesel contains around 10 to 11% oxygen by weight. Then again, weaknesses incorporate their higher viscosity, higher pour point, lower calorific value and lower volatility. In addition, their oxidation stability is lower, they are hygroscopic and as solvents, they might cause consumption of parts (corrosion), assaulting some plastic materials utilized for seals, hoses, paints and covering. They

indicate expanded weakening and polymerization of engine sump oil, along these lines requiring more successive oil changes. [54, 55]

In this review, execution of a four stroke, single cylinder, direct injection (DI) Diesel engine powered with Diesel Biodiesel blend has been tested experimentally. The results appeared that, effective power increases constantly, effective efficiency increases to a specified value and then starts to decrease with increasing engine load at constant speed, as shown in Figure22 and Figure23.

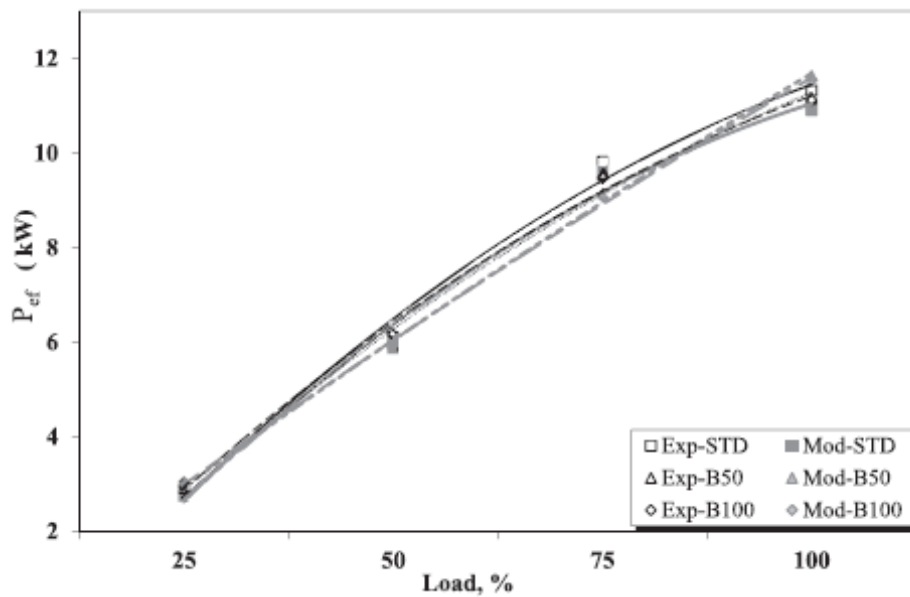


Figure 22. Comparison of theoretical results and experimental data for effective power [55]

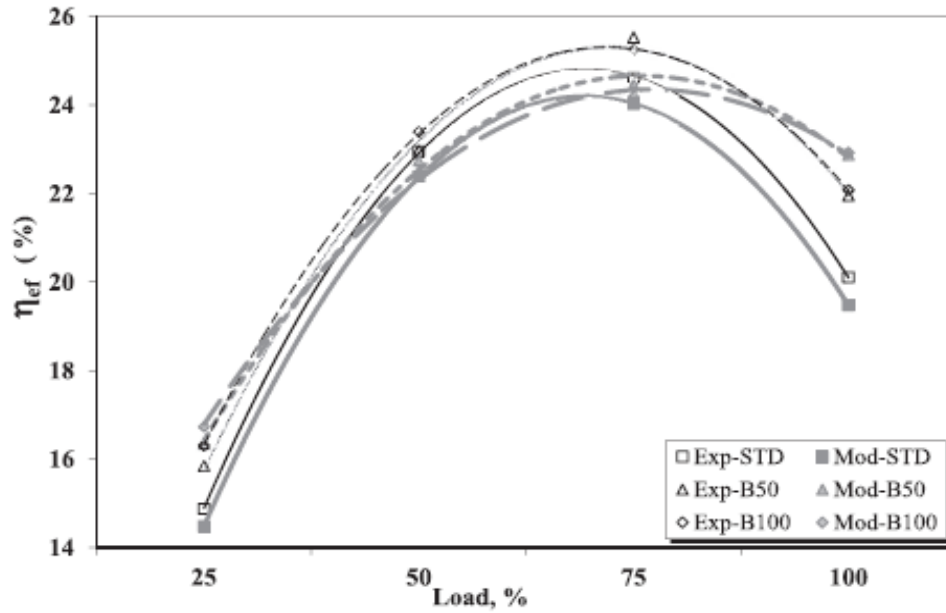


Figure 23. Comparison of theoretical results and experimental data for effective efficiency [55]

Figures 22 and Figure 23 indicate for experimental and hypothetical consequences of effective power and effective efficiency as for change of engine load for various engine modes. It can be noticed that the effective power increments with expanding engine load and it spans to most extreme esteem at 100% load. The most extreme viable power is acquired at STD condition as 11.32 kW. The effective efficiency increments up to 75% load and after that begin to diminish. It reaches its peak value at 25.53% which is gotten at B50 condition. As can be found in the figures above, the simulation results agree with the experimental data and the most extreme difference between them is below 4.5%.

2.4.2 Diesel- Ethanol Blend

Using the blend of Ethanol with Diesel as an alternative fuel is one of the best strategies to battle air contamination from vehicles. This comes from the fact that it has a biodegradable nature to decrease environmental emissions from vehicles; Ethanol gives a tool to address ecological worries without requiring a completely new path for products and individuals to get starting with one place then onto the next. Ethanol contains 35% oxygen and with adding oxygen to fuel brings about more entire fuel ignition, decreasing destructive tailpipe emissions [56]. In this review, the mixing between Ethanol and Diesel Fuel were E0, E2.5, E5, E7.5 and E 10. The operation of diesel engine is at 1,000-1,500 rpm with 0, 10, 20, 30, 40, 50 and 60 Nm engine loads. The direct mixing of Ethanol and Diesel fuel has favorable advantages in decreasing emissions of CO, Soot and NO_x rates. The engine brake power of pure Diesel is marginally lower than those of E2.5-E10, particularly for speed above than 1400 rpm [57].

The variation of engine brake power with the variation of speed for Diesel fuel and mixed ethanol from E0, E2.5, E5, E7.5 and E 10 was displayed on Figure 24. It appears that the engine break power is very insensitive above than 1400 rpm. Unexpectedly, beneath 1400 rpm the engine break power is sensitive. However, the engine power break of pure Diesel (E0) is marginally lower than those of E2.5-E10, particularly for speed above than 1400 rpm.

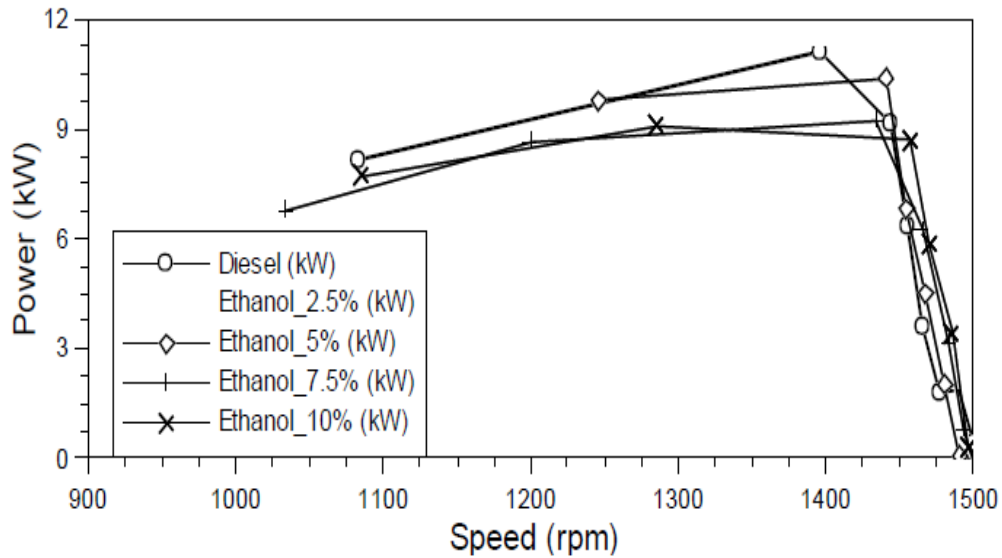


Figure 24. Comparison of theoretical results and experimental data for effective power [57]

The brake specific fuel consumption (BSFC (g/kWh)) is characterized as the proportion of the rate of fuel consumption (g/h) and brake power (kW). Figure 25 shows that the variation of the BSFC with speed (rpm) for Diesel fuel and mixed ethanol. For all of fuels experimented at range of 1475 rpm – 1500 rpm least BSFC was acquired at 490 rpm as 450 g/kWh for Diesel fuel, 1000 g/kWh for E7.5% and 2000 g/kWh for E10. Also from the results of BSFC, it is contended that the rate of brake specific fuel consumption is dependent upon the lower heating value where the Ethanol has a lower heating value than diesel fuel.

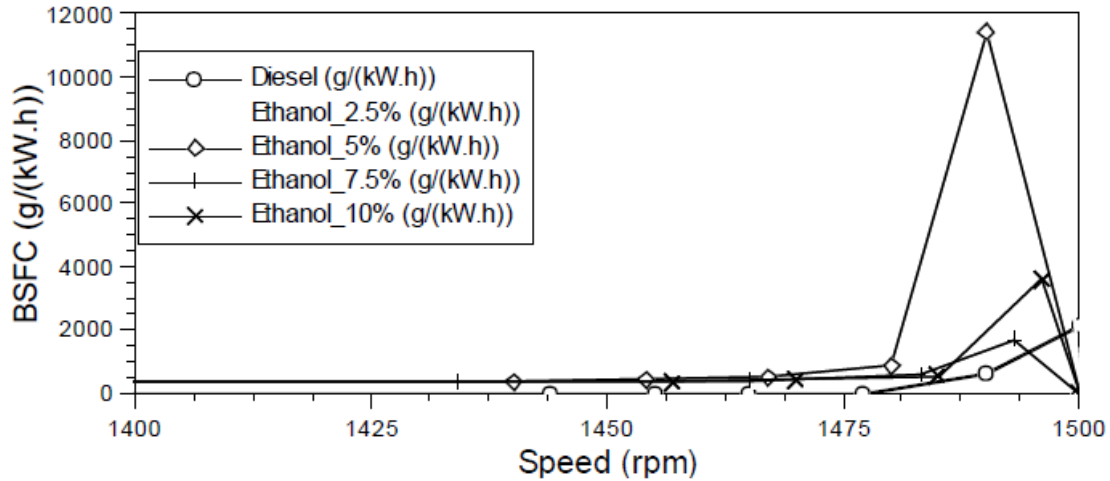


Figure 25. Effect of ethanol addition to break specific fuel consumption [57]

Another experiment that was conducted on a four stroke, single cylinder diesel engine at a constant speed (speed = 1500 rpm) stated that using a fuel blend that constitutes from 50% volume Ethanol and 50 % Diesel can decrease the particulate matter and the smoke opacity, however the emission rate of Nitric oxides is increased significantly because of the low cetane number of Ethanol resulting in a high flame temperature and higher NO_x level. The NO_x emission could be decreased to be less than the emission of diesel by adding a 7% DME to the blend .The addition of DME to the blend has reduced the NO_x emission by a 22.4%.The reason behind that is referred to the high cetane number of DME which has increased the blend total cetane number and resulted in a lower ignition delay and less NO_x level. Figure 26, Figure 27 and Figure 28 compare between diesel, diesel –ethanol blend and the blend of diesel-ethanol and DME in Particulate matter emission, Smoke density and Nitric Oxide emission respectively. [58]

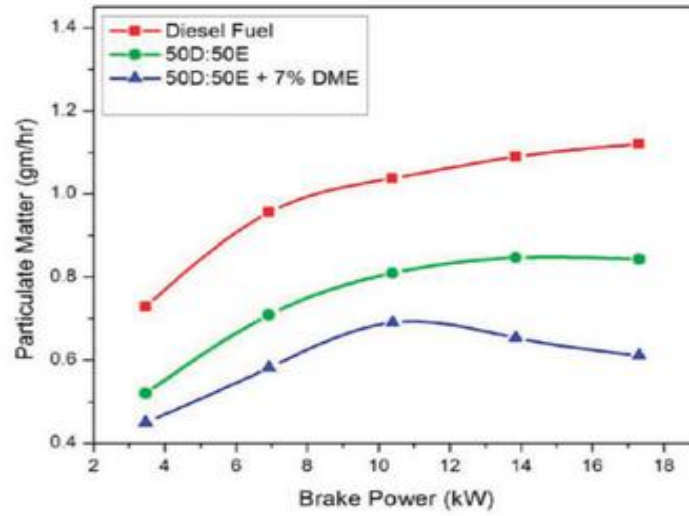


Figure 26. Comparison between diesel, diesel –ethanol blend and the blend of diesel-ethanol and DME in particulate matter emission [58]

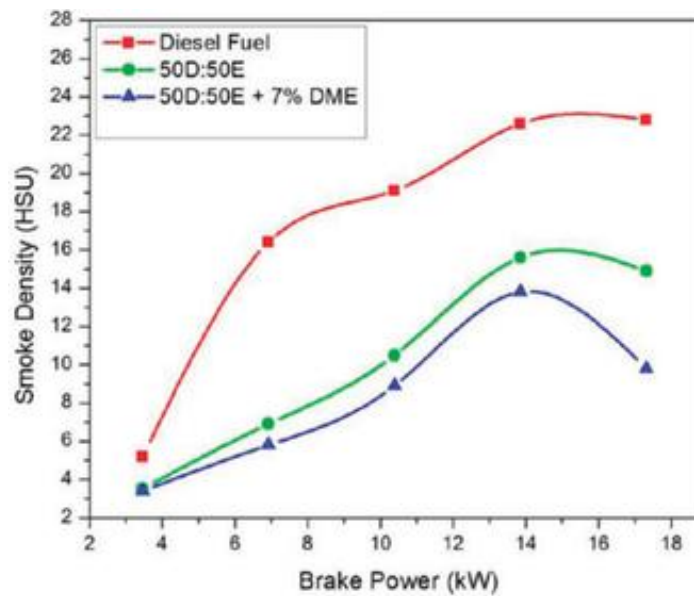


Figure 27. Comparison between diesel, diesel –ethanol blend and the blend of diesel-ethanol and DME in smoke density [58]

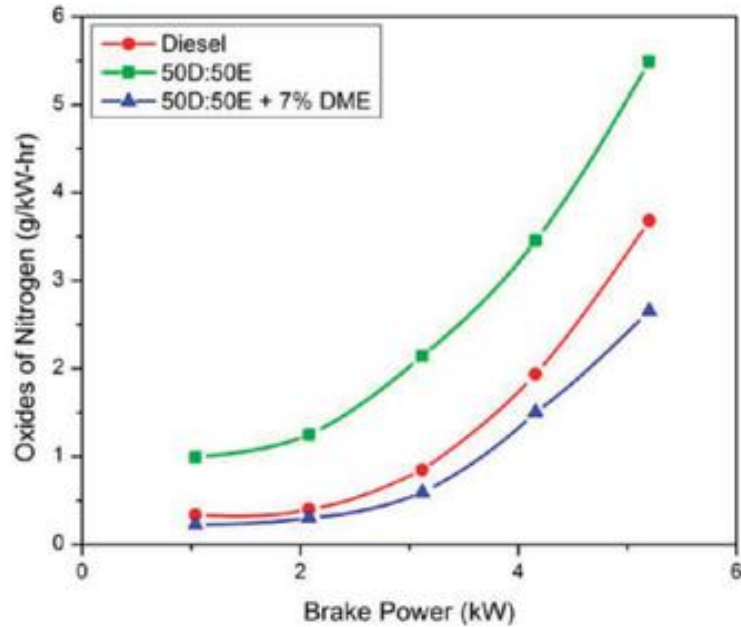


Figure 28. Comparison between diesel, diesel –ethanol blend and the blend of diesel-ethanol and DME in nitric oxides emission [58]

2.4.3 Diesel- Butanol Blend

The aim of this review was to evaluate combustion, performance as well as emission attributes of Diesel engine using Diesel-Butanol blends. In this context, trial examinations were done on a single cylinder four stroke water cooled direct injection Diesel engine utilizing Butanol mixed blended at various volume ratios with Diesel fuel. The Butanol had no stability or solubility issues when mixed with Diesel fuel. As there was most certainly not phase separation in the mixes, no added substance was included. The exploratory examination was finished with four diverse mixes of Butanol on volume premise [B0 (0% Butanol and 100% Diesel), B5 (5% Butanol and 95% Diesel), B10 (10% Butanol and 90% Diesel), B15 (15% Butanol and 85% Diesel) and B20 (20% Butanol and 80% Diesel)] to

concentrate the effect of utilizing Butanol - Diesel mixes on Diesel engine combustion, performance and emissions.

Figure 29 demonstrates the variation of BTE (Brake Thermal Efficiency) at various loads for various mixes of Butanol. BTE increments with an expansion in load for all mixes.

Higher the rate of Butanol in the blend, enhancement of the brake thermal efficiency can be figured out in contrast with pure Diesel fuel. This is because of better combustion on account of the presence of oxygen, which includes higher combustion efficiency. Butanol minimizes the interfacial pressure between at least two communicating immiscible fluids helped the better atomization of fuel, which enhances combustion of Diesel.

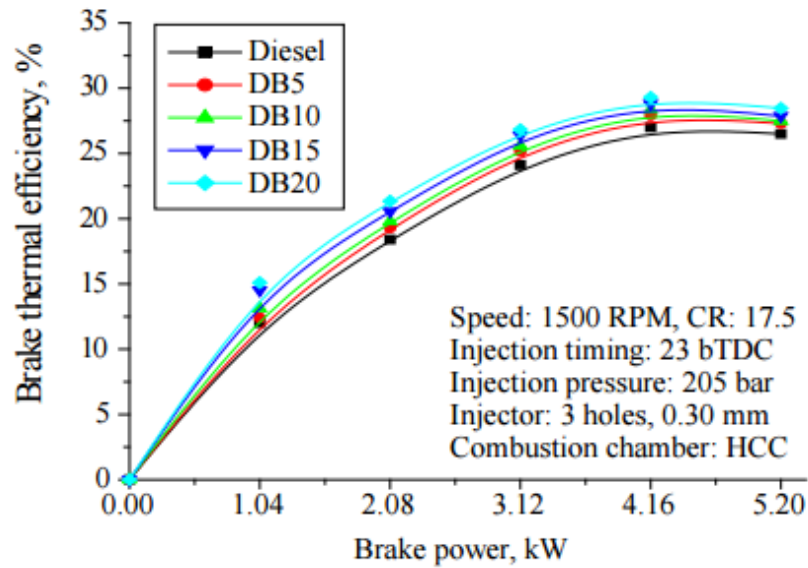


Figure 29. Variation of brake thermal efficiency with load for different butanol blends percentage [59]

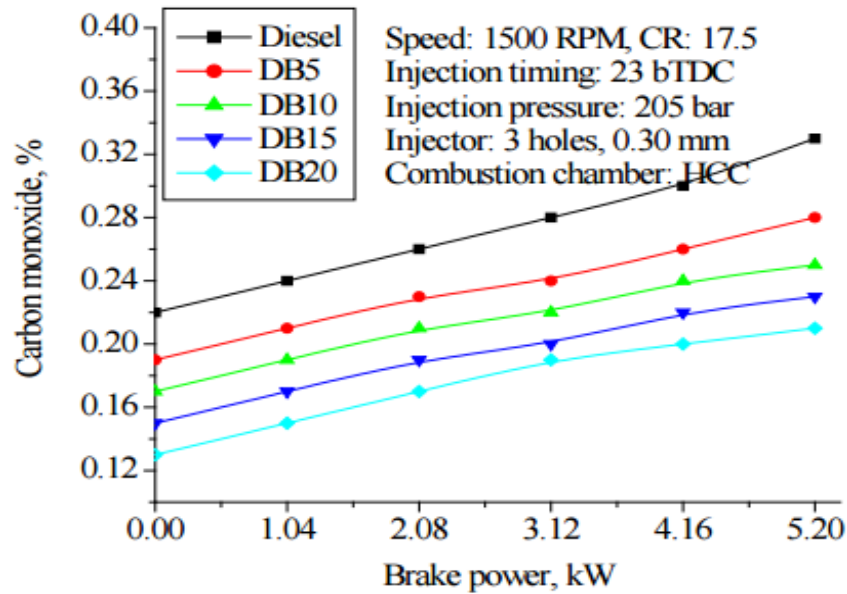


Figure 30. Variation of carbon monoxide emission with load for different butanol blend percentage [59]

Figure 30 demonstrates the variation of CO with load for various rates of Butanol mixes. CO emissions are greater at lower loads. This is because of the reality that its idle latent heat of evaporation is somewhat higher than that of Diesel; as a result there is insufficient vaporization and thus less time to burn fuel totally that outcome in significant increment in CO emanations. At higher loads, enough time accessible for combustion to happen, better blending and inbuilt fuel oxygen which outcomes in total combustion and henceforth marginally diminished the CO emissions, for mixes at high load.

It can be deduced from the previous section that the use of biofuels with diesel in a fuel blend can enhance engine performance and reduces its emission rates. Therefore, in the present study, an experimental investigation is done on a diesel engine using different types

of alternative fuels such as GTL and using corn oil as a biofuel in a blend form with diesel. In addition to that, waste cooking oil has been also utilized to produce biofuels and save the environment from large amounts of hydrocarbon waste.

CHAPTER 3. EXPERIMENTAL SETUP

In this chapter of the report, there will be a detailed discussion about the test rig in section 3.1. After that, this chapter proposes the new induction manifolds that are developed to achieve the goal of this project in section 3.2. In section 3.3, the main physical and chemical properties of the alternative fuels that will be blended with diesel fuel are presented. In the last section of this chapter (3.4) the research methodology that is followed in this project to implement the experiments and get the results is discussed.

3.1 Experimental Method

The experimental tool used for this project is the engine test bed. The aim of this project is to characterize the performance of the engine by using various new induction manifold designs and by altering the type of fuel by using alternative fuels and fuel blends. A schematic diagram for the engine test bed and the measuring devices is shown in Figure 31. This experimental design setup includes the main key features that can be withdrawn out from various experimental setups and it also fulfills the objectives of this project.

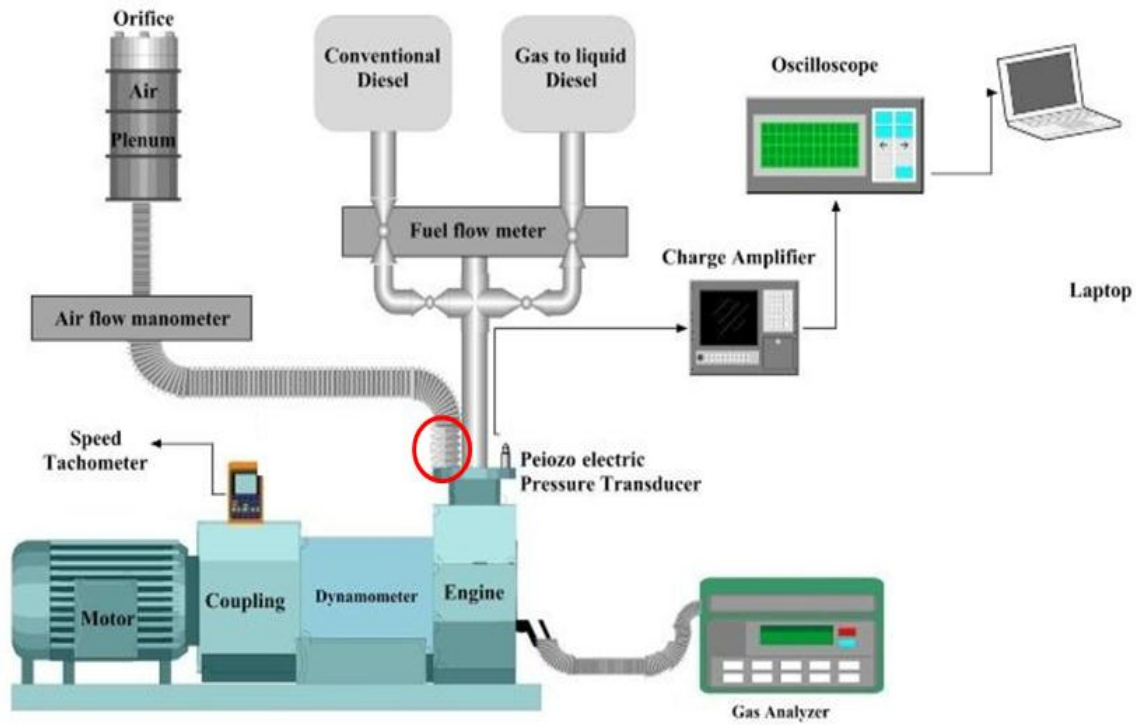


Figure 31. Schematic of the test rig [60]

The engine experiment was performed on a T85D-DIDACTA ITALIA engine test rig that is coupled to an ARONA single cylinder, four stroke, water cooled direct injection compression ignition diesel engine which is coupled to an electric dynamometer with a motor besides a load cell. The mechanical and geometrical specifications of the engine are characterized in Table 6.

Table 6
 Specifications of the used engine

Parameter	Specification
No. Cylinders	single cylinder,4-stroke
Engine Type	Compression ignited
Type of Cooling	Water-Cooled Engine
Bore (m)	0.082 m
stroke (m)	0.068m
Max.Power (H.P.)	6.5 H.P.
Used Fuel	Diesel

The dynamometric unit is mainly a direct current electrical machine that is suitable for working therefore either as a generator or as a motor. Therefore it allows the starting of the endothermic engine and its motoring over. In order to allow for the dynamometer to run, a motor that is connected to the dynamometer through a coupling should be turned on by a switch located on the engine control board. Figure 32 demonstrates how the motor is coupled to the engine through the crankshaft. In this context, the speed of the engine can be tracked out by directing the laser beam of the speed tachometer that uses the pulse counting principle to detect the crank shaft speed. The features and specification for the speed tachometer are covered in Appendix B.

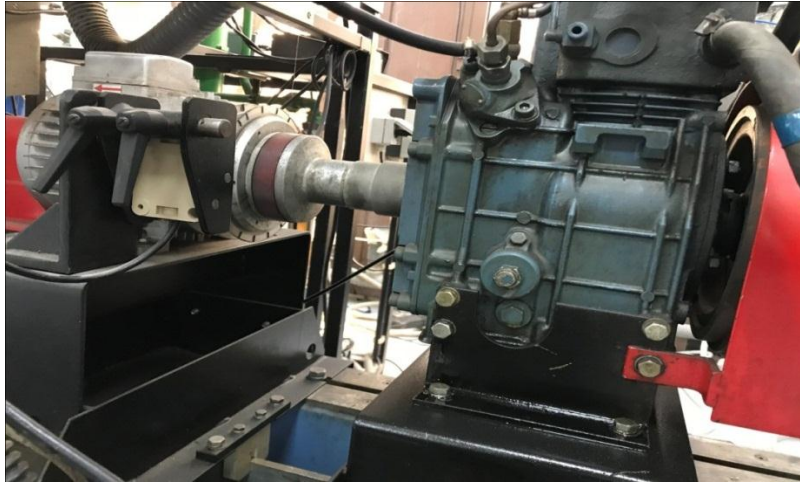


Figure 32. Dynamometer coupled to the engine with the crankshaft

An electrical control board that is installed on the engine test bed accompanies most of the controlling features, buttons and knobs. This board involves the following:

- 1) General switch
- 2) Extractable key switch to activate the feeding of the circuits of the control board
- 3) Spy-pushbutton with red light with functions as a detector of the carried out activation of the electrical circuits and as a safety switch for the feeding circuit.
- 4) Voltmeter for the voltage delivered by the mains
- 5) Commutator to connect the voltmeter 4 with the phases of the feeding network
- 6) Voltmeter for the voltage delivered by the dynamometer
- 7) Ampere meter for the current delivered by the dynamometer
- 8) Selector of the functional conditions of the direct current electrical machine. The horizontal positions of the knob correspond to the condition “motor”. The vertical position (pointer downwards) corresponds to the position “brake”.

- 9) Selector for the electromagnetic couplings: it consents to connect to the dynamometric unit one or the other of the endothermic motors installed on the test stand.
- 10) Potentiometer to regulate the power delivered by the dynamometer in its working condition as a motor.

Figure33 shows how this board does looks like and where each component (1-10) is located on the board looks

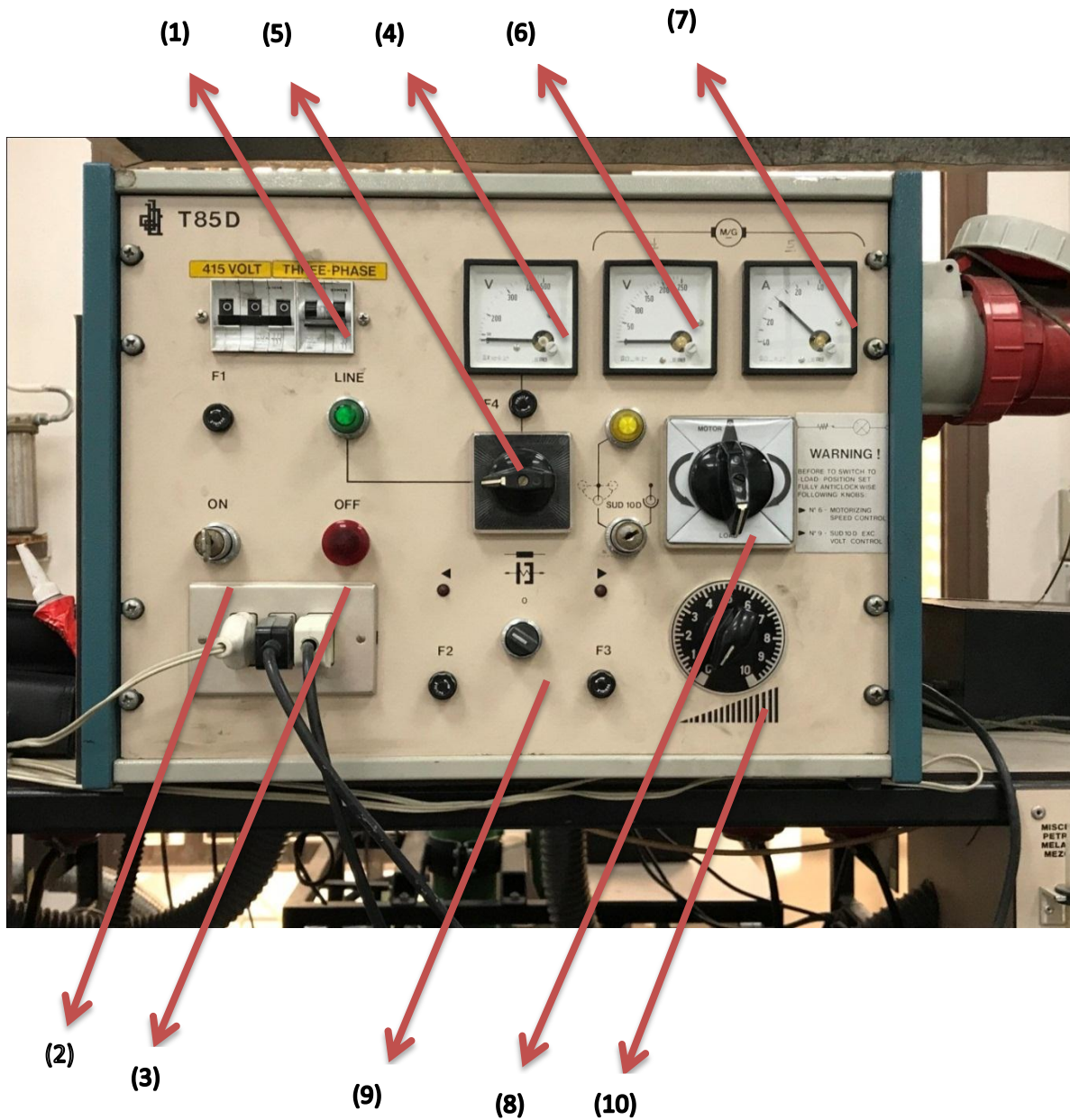


Figure 33. Control board that has the main switches

As it is shown in the schematic of Figure 31, there are two fuel tanks that are used to supply the engine with either Diesel or any alternative fuel by using fuel pipes and controlling valves to control the amount of fuel that is supplied to the engine. Figure 34 shows those two fuel tanks.

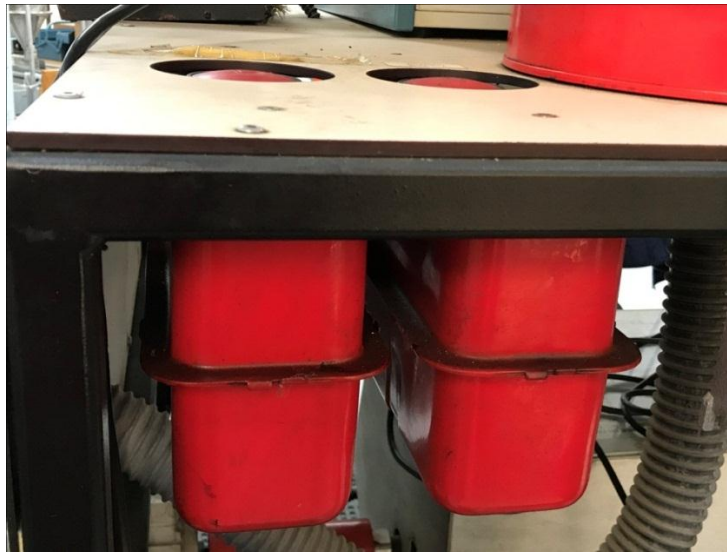


Figure 34. Fuel tanks

As the fuel is being discharged from the tank during the experiment, the fuel flow rate can be measured by using a calibrated burette and a stop watch. Moreover, it is important to note that the volume flow rate can be measured by using a stopwatch and by controlling the fuel supply valve such that it is closed when five cubic centimeters are read on the fuel supplying pipe. Figures 35, 36 demonstrate the fuel calibrated burette and the fuel supply valves respectively.

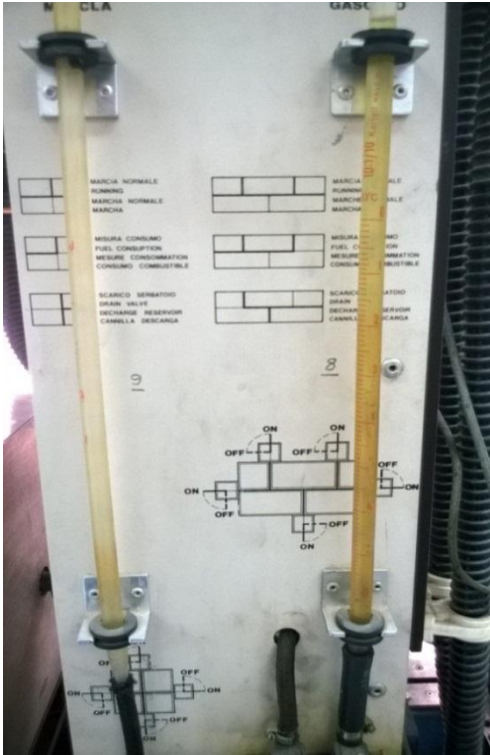


Figure 35. Fuel calibrated burette



Figure 36. Fuel supply valves

It is also visible on the schematic of Figure 31 that there is a tank used to supply the engine with air. Generally, devices such as Nozzles, Venturi meters, and orifice meters are used to measure the air flow rate by creating a reduction in the passage section of the fluid and generate an increase of its speed and therefore a decrease of the pressure in the decreased section. The flow rate is then calculated by an easy way without having to use formulas; however there are diagrams that allow the immediate calculation of the flow rate according to the measured value of ΔP . A typical diagram that allows for the measurement of the air flow rate is attached on Appendix A. The air fuel ratio can be then calculated based on the

knowledge of volume flow rate of the fuel and of the air. Figure 37 demonstrates the air tanks used to supply engine with fresh air. The aim of using these tanks is to damp any fluctuations in the air flow in order to be measured easily.



Figure 37. Air tanks

Before the air enters into the combustion chamber of the engine, it flows through an inclined pipe called the induction manifold (marked with a red circle on the schematic of Figure 31). The existence of this induction manifold is vital for the combustion process since it causes the swirl motion of the air flowing into the engine. As to achieve the purpose of this project, the standard normal induction manifold is not kept on use permanently; however, it is replaced with new manifolds (new induction manifold designs) so the performance of the engine is tested while those new manifolds are kept in use.

The in-cylinder pressure and crank angle position were obtained by AVL QH 33D water cooled piezoelectric pressure transducer and PALAZZOLI digital shaft encoder; the output of the pressure transducer was amplified by an AVL charge amplifier and then the output signals displayed on Instek GDS-3152 Digital Storage Oscilloscope with 150 MHz sampling rate. Hence, the data was recorded and saved as CSV files and transferred for further analysis.

The pressure transducers installed on the engine test bed which is also called a pressure transmitter is mainly a sensor that is used to convert pressure into electrical signal. The pressure transducer sensor is connected to a charge amplifier which is an electronic current integrator that generates a voltage output that is proportional to the integrated value of input current. Finally the oscilloscope is used to preview the output signals on the voltage – time axis. This oscilloscope has a built in data acquisition system that can be used to record a very high number of voltage against time readings and then present them on an excel sheet that has a CSV file format. Later on, a Matlab code is used to convert the Voltage - Time signals to Pressure – Crank angle (θ) signals where the maximum pressure can be then extracted out of these signals with respect to the corresponding crank angle. The features and specification for the oscilloscope and the calibration process for the pressure transducer are covered in Appendix C.

An ENRAC portable emission analyzer with an accuracy of 1 ppm is used to measure exhaust emissions concentrations of CO, CO₂, HC and NO. Thus, the output data were displayed on a laptop with units and recorded as text file. The features and specification for

the gas analyzer and its calibration process are covered in Appendix B. In addition to that, the principle of operation is explained.

An ECO SMOKE 100 smoke meter was used for the measurement of emission level on that Diesel engine by partial flow method using optical based on folded geometry. It has an opacity range between 0 – 99.9% with a resolution of 0.10 % .It has the capability to work on the environmental conditions of the lab and it operates on both AC and DC power supply. The features and specification for the smoke meter and its calibration process are covered in Appendix B. In addition to that, the principle of operation is explained.

An AEROCET 531 is a small portable unit used as a particle counter and a mass concentration detector operated with battery. It can be used in two modes, one of them for counting the number of particles and the other is for detecting the mass concentration of the particles. In the particle counting mode (used in this experiment) it displays on the LCD screen the number of particles detected on the exhaust for the particle sizes $>0.5\mu\text{m}$ and $>5.0\mu\text{m}$ after one minute of operation. In the mass concentration mode, it provides the mass concentration of the particle per cubic meter for the sampled air. It can test particle sizes as fractions of PM1, PM2.5, PM7, PM10 and TSP. The features and specification for the particulate meter and its calibration process are covered in Appendix B. In addition to that, the principle of operation is explained.

K type thermocouples were mounted to measure air inlet, exhaust gases and cooling water temperatures at relevant points. Even though all sensors were provided with its calibration charts and were recalibrated for more accuracy with respect to temperature and pressure lab

working condition that were monitored during the tests. The uncertainties in the measurement and the results are summarized in Table 7.

Table 7

Uncertainty in the readings of measurement and the results

Variable	Uncertainty (%)
Torque (N m)	± 1
Speed (rpm)	± 0.3
Power (kW)	± 0.533
Time (s)	± 1
Fuel volume (cm ³)	± 0.5
bsfc (g/kW.h)	± 0.6
Exhaust gas temperature (C)	± 1.0
Volumetric efficiency (%)	± 0.75
Air flow rate (kg/h)	± 0.125
CO (%)	± 0.1
CO ₂ (%)	± 0.05
HC (%)	± 0.04
NO (ppm)	± 0.26
PM (ppm)	± 10.0
Pressure (bar)	± 0.06

As to make the engine test bid successfully integrated with the project vision that cares about releasing the amount of emission gases, the gas analyzer ,the smoke meter and the particulate meter are connected to the engine exhaust pipe. These three devices have the capability to sense the concentration of the emission gases flowing out of the engine as a result of the combustion process such as Carbon Monoxide, Carbon Dioxide, Oxygen , Hydrocarbons, NOx ,particulate matters and the smoke opacity. The engine experiments were performed at heat engines laboratory in Qatar university and two sets of experiments were performed. In the first set, the engine load was kept constant at 1N.m and the speed was varied. However, in the second set the engine speed was kept constant at 1700 rpm and the load was varied.

3.2 New Induction Manifold Designs

One possible way to increase the engine efficiency, decrease the emission rate and enhance the swirl number in diesel engines is to change the induction manifold design .The new induction manifolds have spiral and helical shapes with two different design parameters configurations inner diameter and outlet port angle which can be varied and is to be set 30° in this experiment. They have been 3D designed using Solid Works and manufactured using Fused Deposition Modeling (FDM) additive manufacturing technology by UPrint SE Plus 3D printer. It has 0.254 layer thickness and 203 x 203 x 152 mm working space. The modified intake manifolds prototypes material has been selected from ABS filament which has a mechanical strength and thermal resistance properties that can withstand high temperature and pressure, however in this experiment it is only used to withstand only low temperature as the new intake manifold receives only air at atmospheric condition. In this

section, three different designs for the intake manifold are proposed which have the same outlet angle ($=30^0$) and different inner diameters. The selection for 30^0 as the manifold outlet angle is justified by the reason that it gives the highest turbulence kinetic energy at low valve lifts as it was discussed in the literature review chapter. Swirl number which is a parameter that is used to characterize the level of swirl depends mainly on geometrical factors such as the inner and outer diameter. Also; it depends on the radial velocity as well as the entrance velocity. As this parameter is important for design selection, it will be calculated for every intake manifold in companion. The equation that is used for calculating the swirl number is [61]:

$$S = \frac{2}{3} \left[\frac{1 - \left(\frac{d_h}{d_o}\right)^3}{1 - \left(\frac{d_h}{d_o}\right)^2} \right] \tan \alpha \quad \text{Eq. 1}$$

Where,

d_h : is the hub or inner diameter of swirler

d_o : is the outer diameter of the swirler

α is the inlet angle to the swirler

First New Induction Manifold

Figure 38 illustrates the design of the first new engine's intake manifold where the inlet diameter is kept the same as the standard intake manifold, however the outlet angle is varied which will make an effect in swirl motion of the incoming air flow, hence this is expected to enhance engine performance and reduce its emission rates because of the better mixing between fuel and air in the combustion chamber. The swirl number was calculated for this intake manifold to be equals to 1.155.



Figure 38. Induction manifold with one normal manifold diameter and 30° outlet angle

Second New Induction Manifold

Figure 39 illustrates the design of another new induction manifold where the inlet diameter is doubled (x2) and the outlet angle is 30° . The swirl number was calculated for this new manifold and was found to be equals to 1.25.



Figure 39. Induction manifold with doubled inlet diameter and 30 degrees outlet angle

Third New Induction Manifold

Figure 40 illustrates the design of the third new induction manifold where the inlet diameter is tribled (x3) and the outlet angle is set to be 30°.The swirl number was calculated for this new manifold and was found to be 1.34.






Figure 40. Induction manifold with tribled inlet diameter and 30 degress outlet angle

To summarize, Table 8 compares between the normal and the three new manifolds.

Table 8

Comparison between the normal and the three new manifolds

Intake Manifold type	Outlet angle	Intake manifold name	Total length and volume cm/cm³	Geometrical swirl number	Image
Standard	0	Normal	21.1 cm/ 186.7 cm ³	0.091	
Helical- spiral with 1D helical diameter	30	m(D,30)	33.5 cm / 200.12 cm ³	1.155	
Helical- spiral with 2D helical diameter	30	m(2D,30)	64.4 cm/ 385.21 cm ³	1.25	

Helical-

spiral

with 3D 30

m(3D,30)

95.86 cm/

1.34

573.53 cm³

helical

diameter



3.3 Fuels Used

The waste cooking oil was supplied from Qatar university food court campus. Biodiesel prepared and characterized by chemical engineering department at Qatar university according to the ASTM standards. It was prepared at the lab temperature of 25 °C; sodium hydroxide catalyst percentage by weight of waste vegetable oil 0.5-0.6 %; stirring time 30 minutes and 50% excess of methanol with NaOCH₃ were optimum conditions..

The corn oil was prepared manually in the laboratory with very much caution. The corn seeds were shelled with a seed mill at first, and then dropped in water to get rid of any remaining shell. After that, they were collected and heated on the oven at a temperature of 300 C for around 30 minutes. The heated corn seeds were then allowed to cool and finally boiled in a water container. The resultant corn oil was stored on a container for later use.

The main physical and chemical properties for those alternative fuels are listed on Table 9, in addition to the properties of diesel fuel.

Table 9

Main properties of fuels used

Properties	Diesel	GTL	Corn oil	Waste cooking oil
H/C Ratio	2.125	2.1-2.15	1.9-2.0	1.82
Approx. Formula	$C_{16} H_{34}$	$C_{16} H_{34}$	$C_{16} H_{34} O_2$	$C_{17} H_{31} O_2$
Density at 15C (kg/m ³)	866	760	915	863
Flash Point (C)	55	77	270	160
Cetane No.	55	70	37.6	32.5
Calorific Value (MJ/kg)	44.3	47.3	36.3	42.7
Viscosity (mm ² /s)	5.2	2	45	4.9
Distillation Temperature (C)	190-200	190	160	316

In this experiment, two fuels (Diesel and GTL) are used in the pure form and there are four fuels in the blended form. Those fuel blends were prepared at the laboratory in Qatar university and their physical and chemical properties were analyzed and determined there by chemical specialists. The first fuel blend is the mixture between diesel and GTL with a 50% to 50% volume composition. This fuel blend will be indicated to by the symbol 'DG' in Table 18 and in the results chapter. The second fuel blend is the mixture between diesel and waste cooking oil by a 50 to 50 volume percentage and is indicated to with a symbol

‘DW’. The third fuel blend is the mixture between diesel, GTL and waste cooking oil by an equivalent volume percentage in which each fuel contributes to 33.33 % from the fuel blend. This fuel blend will be indicated to by the symbol ‘DGW’. The last fuel blend is the mixture between diesel, GTL, corn oil and waste cooking oil by a volume percentage composition of 25% for each and will be indicated to by the symbol ‘DGWW’. Table 10 summarizes the main properties for those fuel blends.

Table 10
Main properties of used fuel blends

Property	DG	DW	DGW	DGWW
H/C Ratio	2.138	1.973	2.032	2.024
Density at 15C (kg/m ³)	813	865	830	851
Flash Point (C)	66	108	97.3	141
Cetane No.	62.5	43.8	52.5	48.8
Calorific Value (MJ/kg)	45.8	43.5	44.8	42.7
Viscosity (mm ² /s)	3.60	5.05	4.03	14.3
Distillation Temperature (C)	195	258	235	217

3.4 Research Methodology

As this research is based on using alternative induction manifold designs and alternative fuels that can replace the standard normal manifold and the conventional Diesel fuel, consequently it is expected that the methodology on this project is based on the first place in fabricating new manifold designs and preparing those alternative fuels in the laboratory for test or bringing them from external resources.

The engine test bed should be supplied with all measuring devices that will be used on the experiments and their connections to the engine must be correct. Some of these devices are used for sensing and measuring the emission gases coming out of engine exhaust such as the smoke meter, the gas analyzer and the particulate meter. Others are used for sensing the engine operating status such as the speed tachometer which is used to detect the speed of the engine's crankshaft. In addition to the speed tachometer, the oscilloscope is connected to the engine using a pressure transducer where it can generate waves for the voltage against the time. After that, these waves are transformed and plotted as graphs which present the value of cylinder inner pressure at each crank angle position using calibration equations.

The new induction manifolds have spiral and helical shapes with two different design parameters configurations inner diameter and outlet port angle that is set to be 30°. The inner diameter is varied to be doubled (x2) and tripled (x3) and to be used with this new outlet angle (30 degree). These new manifolds have been 3D designed using Solid Works and manufactured using Fused Deposition Modeling (FDM) additive manufacturing technology by UPrint SE Plus 3D printer. It has 0.254 layer thickness and 203 x 203 x 152

mm working space. The modified intake manifolds prototypes material has been selected from ABS filament which has a mechanical strength and thermal resistance properties. Once all induction manifolds are 3D printed, all fuels are prepared and the measuring devices are connected to the engine, the experiment can be initiated. Measuring devices such as the smoke meter, the gas analyzer are connected to the computer by a USP inlet port as their measured data can be viewed on the computer screen using data acquisition method. Other measurements such as the fuel (or air) flow rate, the exhaust temperature, dynamometer supplied voltage and current can be taken manually. For the oscilloscope, waveforms can be saved on the device and then transferred to a flash memory where they are transformed into pressure-crank angle waves using Matlab software and built in functions on Excel.

The analysis process for data can then be initiated where the comparison between all manifolds as well as used fuels can be made based on the plots and graphs that compare between them. Finally, the manifold that caused the most reduction on the engine exhaust gases rates and raised its performance the most can be compared with all types of fuels that were involved on this experiment where a clear conclusion can be then deduced.

CHAPTER 4. RESULTS AND DISCUSSION

This chapter expresses the data that was acquired and measured during the experimental setup in the form of analytical plots and graphs. These graphs are studied and analyzed as they present and compare the differences in performance between the tested manifolds and fuels. In addition, this chapter discusses the behavior of these graphs and find a relation between interrelated components. Section 4.1 discusses the effect of varying induction manifold design on the performance and the emission rates of the engine, however in section 4.2 the discussion is about the effect of using different fuel blends in the performance and emission criteria.

4.1 Effect of Induction Manifold Design

This section compares between normal manifold and another three new manifolds. The test will involve combustion, performance and emission characteristics using six different types of fuels. Also, the test will be performed at two stages. At the first stage, the engine load will be kept constant at 1N.m and the engine speed will be varied gradually. However, at the second stage, the engine speed will be kept constant at 1700rpm and the engine load will be varied gradually.

4.1.1 Combustion Characteristics

1) P- θ Diagram

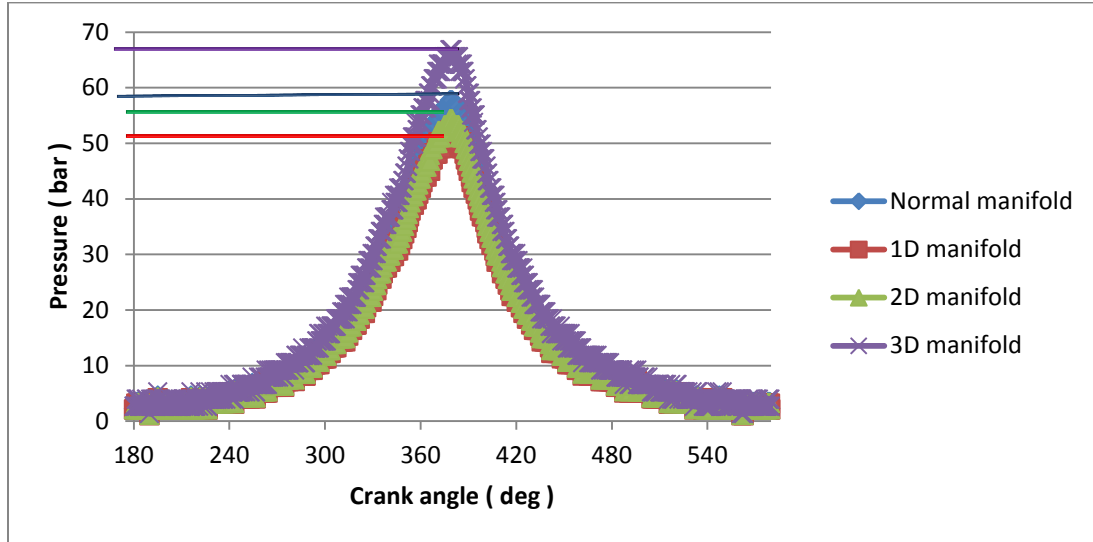


Figure 41. Pressure vs. crank angle position for normal and new manifolds using GTL fuel at a speed=1700rpm and a load =1N.m

The plot of pressure vs. crank angle position is drawn for the range of crank rotation from top dead center (360°) to a period after combustion (aTDC) during the expansion stroke (390°). While studying the plots, the main emphasis was on the period between the start of injection (SOI) and the start of combustion (SOC) and then the peak of the curve achieved at SOI and SOC in which both are expressed in terms of crank angle position before the top dead center (bTDC) since the top dead center (TDC) is the moment at which the piston reaches the highest level in the cylinder at the end of compression.

Figure 41 compares the pressure rise between the normal manifold and the three new manifolds. It is noticed that the pressure rise in case of 3D manifold is the highest. The reason behind that can be explained by the fact which emphasizes that increasing the intake manifold volume causes the air pressure level in the intake manifold to rise, thus incoming air flow into the cylinder will be more pressurized than normal manifold. However, the pressure rise in case of 1D and 2D manifold is very close to the normal manifold as the intake manifold volume is very much similar [62]. Another reason is that increasing the manifold volume enhances the pulsating nature of the intake airflow through manifold. This may develop resonances in the airflow at certain speeds, which improve the engine performance [62, 63].

2) The maximum pressure raise rate (dP-dθ) @ constant load

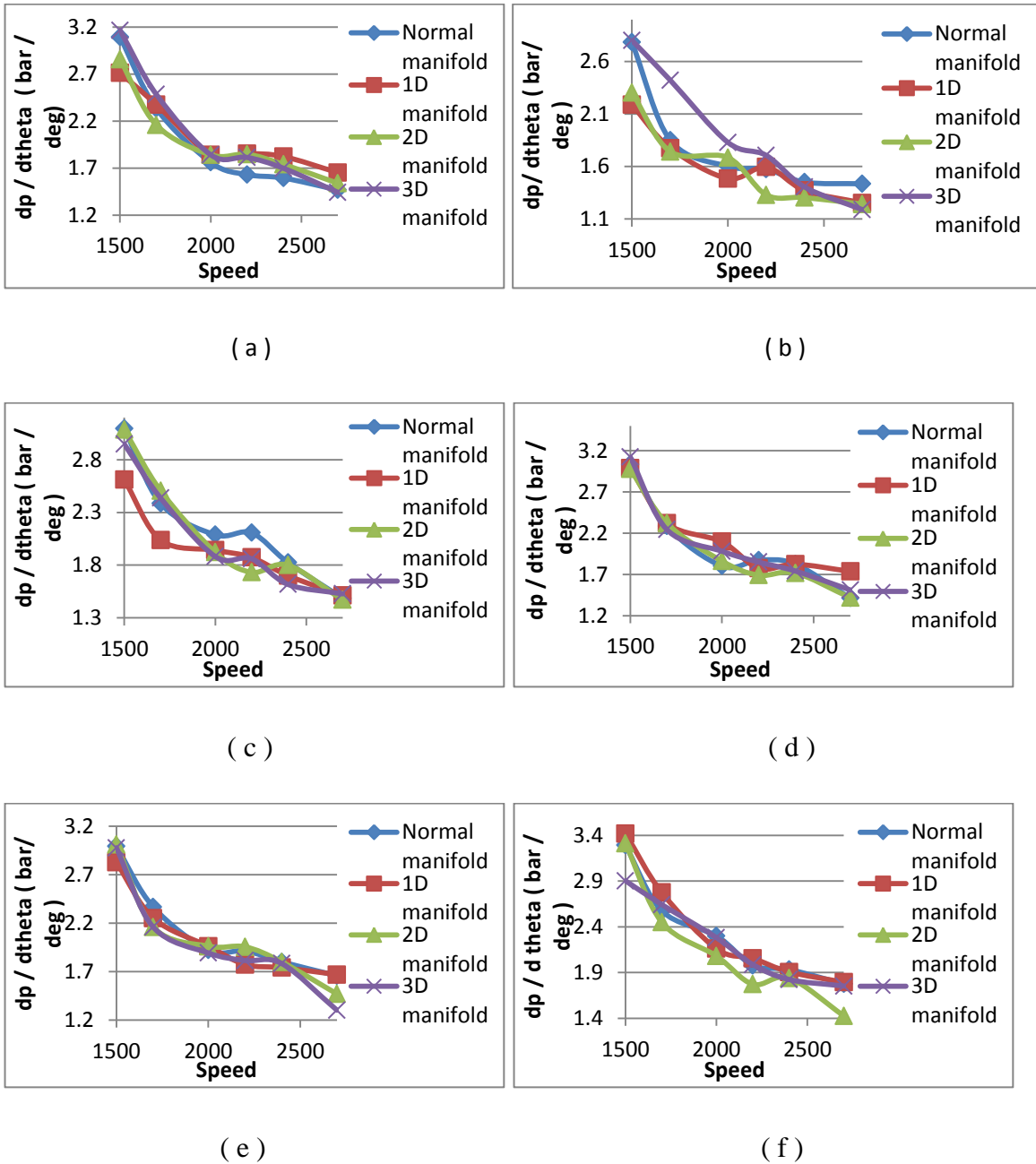


Figure 42. Dp/dθ vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Figure 42 indicates the maximum pressure raise rate with different engine speeds and different fuel blends. In compression ignition engine during combustion process, fuel is injected few degrees (23° bTDC in the engine under study) before top dead center and before the starting of combustion, the droplets of fuel injected mix up with air and takes heat from the compressed air inside the cylinder and burn when reaches to self-ignition temperature of the fuel. During the process of start of injection (SOI) to initiation of combustion the significant number of fuel particle accumulated in the engine cylinder and if the delay in ignition is longer, then large number of particles burns together and leads to rate of pressure rise and $dp/d\theta$ is large. In case of shorted delay gradual burning of small group of fuel particles leads to small peak for $dp/d\theta$ curves.

It can be noticed from Figure 42 that the maximum pressure raise rate is less when using the 1D and 2D manifolds in comparison with the normal manifold as a result of better air-fuel mixing generated by the swirl motion. In addition, this can be related to the high turbulence kinetic energy produced from those manifolds [64]. However, the use of the 3D manifold has caused the maximum pressure raise rate to increase as a result of its increased volume which has caused the incoming air to be more pressurized [62, 63]. The use of the 1D and 2D manifolds instead of the normal manifold can decrease the maximum pressure raise rate by an average percentage of 7%. However, the use of the 3D manifold has decreased the maximum pressure raise rate by about 3%.

Moreover, it can be also observed that the rate of change in pressure is less in case of using GTL fuel as it can be observed in Figure (42 b) because it has the highest cetane number leading to a less ignition delay period. Also, the use of diesel-GTL fuel blend in Figure (42

c) was found to be effective in decreasing maximum pressure raise rate due to the enhanced cetane number of the mixture. However, the addition of waste cooking oil to that blend in Figure (42 e) or for diesel fuel alone in Figure (42 d) has caused some reduction in the maximum pressure raise rate due its very low cetane number (=32.5). In the same concept, the addition of corn oil to the conventional fuels has also decreased the maximum pressure raise rate and causes longer ignition delay periods.

3) Peak pressure Vs. Engine Speed @ constant load

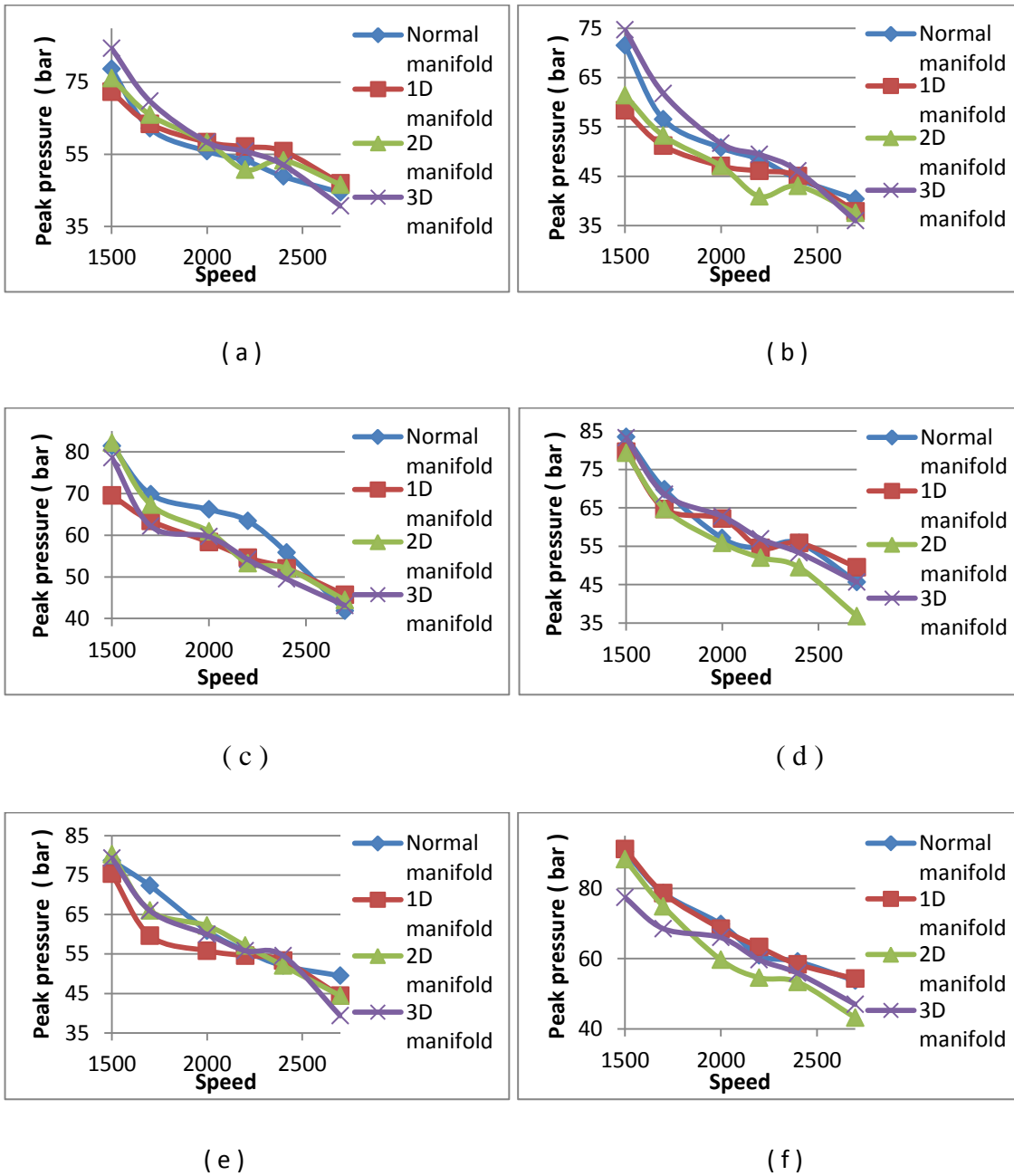


Figure 43. Peak pressure vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

In a compression ignition engine, the combustion characteristics of the fuel and the engine performance and emission depend upon the ignition delay period. The longer the delay period, the higher is the rate of combustion and the higher is the resulting pressure rise. The rise in pressure inside the cylinder is attributed to the rate of combustion as well as the ignition delay period of the fuel, which in turn depends on the cetane index. The higher the value of the cetane index of the fuel, the shorter is the ignition delay period.

Figure 43 demonstrates the variation of the in-cylinder peak pressure with engine speed for the six fuel blends. As it can be deduced, the peak pressure decreases with increasing the engine speed because as the engine speed increases, the mass flow rate of air increases which causes faster and better mixing between fuel and air leading to a faster combustion. Also, it is clear that there is only a slight reduction in the pressure when using the 1D and 2D new manifolds instead of the normal manifold, which indicates that ignition delay is less and hence the in-cylinder peak pressure is reduced. However, the use of the 3D manifold has caused the in-cylinder peak pressure to increase as a result of its increased volume which has caused the incoming air to be more pressurized [62, 63]. The use of the 1D and 2D manifolds instead of the normal manifold can decrease the maximum cylinder pressure by an average percentage of 10%. However the use of the 3D manifold can result in 7% increment of in-cylinder peak pressure.

The values of peak pressure at different engine speeds were the lowest when using GTL fuel as it has the highest cetane number (=70) which causes shorter ignition delay and less peak pressure as shown in Figure (43 b). Also, Diesel and GTL fuel blend is considered to be a good solution for reducing the pressure rise in the cylinder when compared with the

use of pure diesel as Figure (43c) indicates. The cetane number was measured for this blend to be equals to 62.5 which is considered to be high with respect to other used fuel blends.

The waste cooking oil which has a very low cetane number (= 32.5) caused the pressure rise in the cylinder to be higher when it was blended with diesel and GTL as shown in Figure (43 d) and Figure (43 e) . The cetane number was measured for those fuel blends to be 43.8 and 52.5 respectively. The use of corn oil in companion with waste cooking oil in one fuel blend which is the case in Figure (43 f) has further increased the pressure rise in the cylinder as the corn oil has also a low cetane number (=37.6) .

2) The maximum pressure raise rate ($dp/d\theta$) @ constant speed

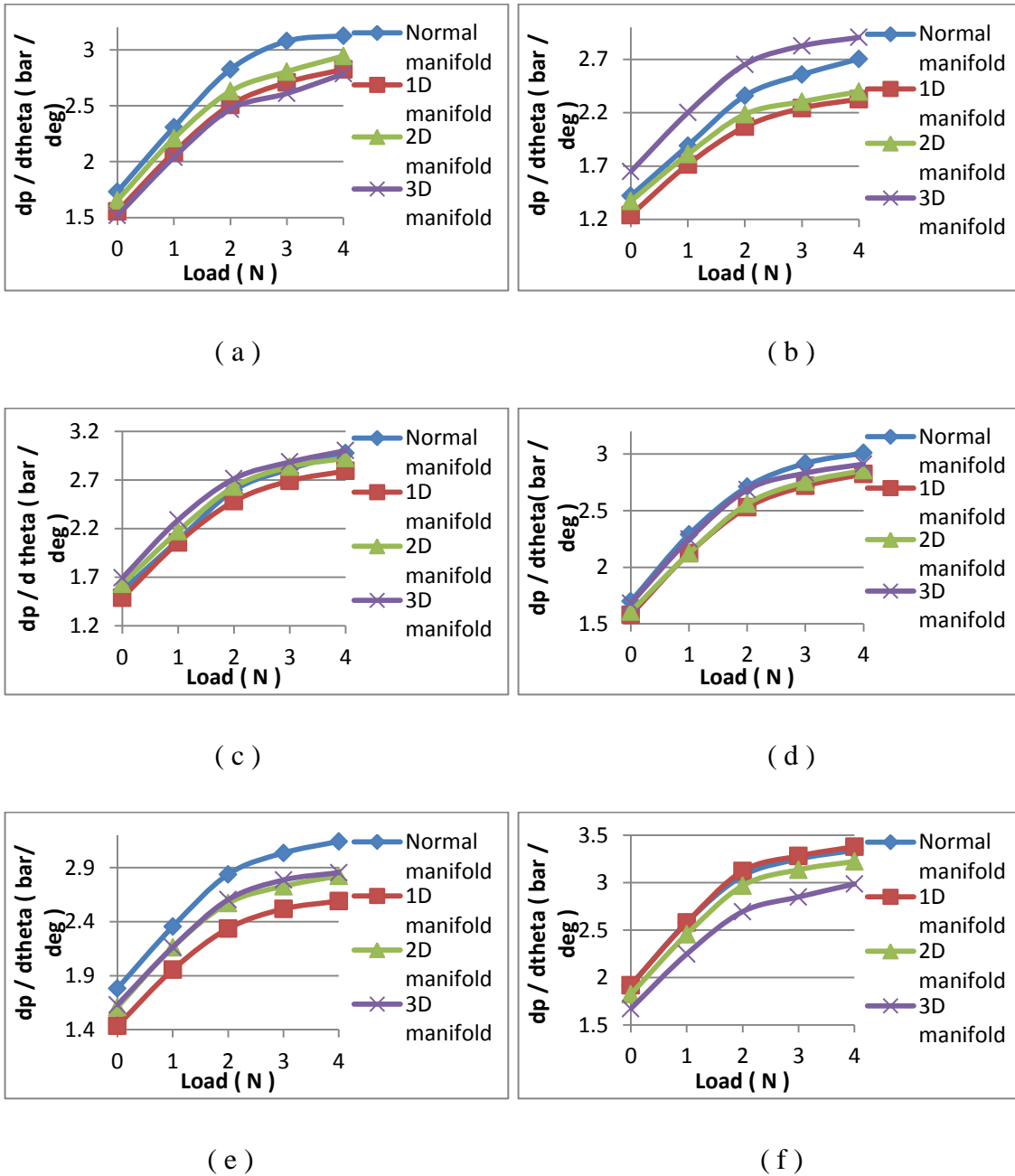


Figure 44. $Dp/d\theta$ vs. engine Load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

A higher value for $dp/d\theta$ means that there is longer delay period and less efficient engine combustion. It can be observed from Figure 44 that the rate of change in pressure with respect to the change in crank angle is less when using the 1D and 2D manifold in comparison with the normal manifold as a result of better air-fuel mixing generated by the swirl motion. However, the use of the 3D manifold has caused the maximum pressure rise rate to increase slightly because of the more air pressure which results due to its increased volume [62, 63]. The use of the 1D and 2D manifolds instead of the normal manifold can decrease the maximum pressure raise rate by an average percentage of 5%. However, the use of the 3D manifold can result in 3% decrement of in-cylinder peak pressure

The maximum pressure raise rate is found to be less when GTL fuel is added to diesel fuel in one fuel blend as Figure (44 c) demonstrates. This can be related to the higher cetane number (=62.5) that results when using GTL with diesel in one fuel blend rather than using it purely (cetane number = 55 for diesel fuel). The use of waste cooking oil in fuel blends is not favorable for reducing the maximum pressure raise rate as it has a very low cetane number (= 32.5) which causes the maximum pressure raise rate in the cylinder to be higher as shown in Figure (44 d) and Figure (44 e). The use of corn oil in companion with waste cooking oil in one fuel blend with the conventional fuels which is the case of Figure (44 f) has further increased the maximum pressure raise rate in the cylinder as the corn oil has also a low cetane number (=37.6) .

3) Peak pressure Vs. Engine Load @ constant speed

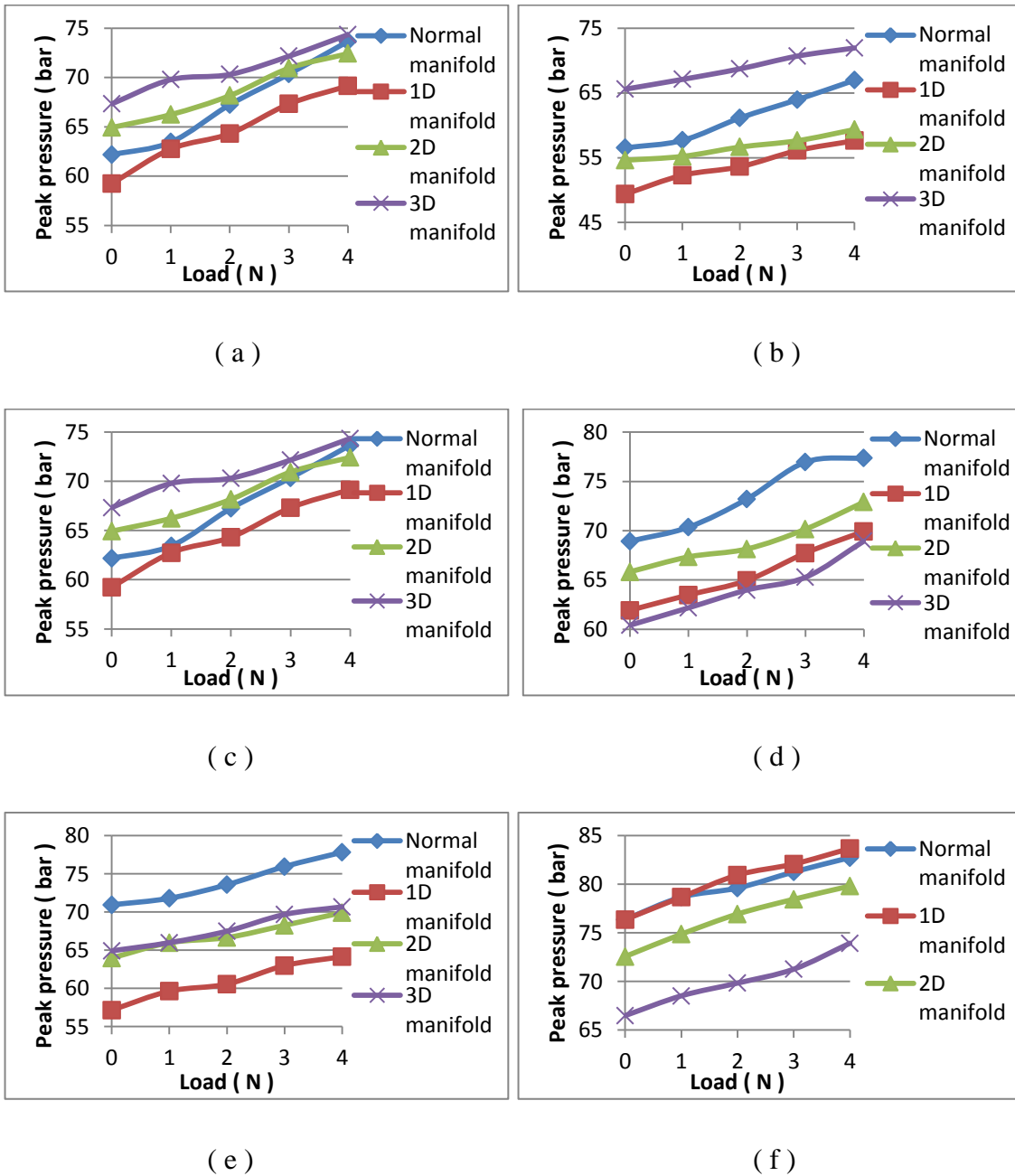


Figure 45. Peak pressure vs. engine load for : (a) diesel fuel, (b) GTL fuel, (c) diesel-GTL blend, (d) diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Figure 45 demonstrates the variation of the in-cylinder peak pressure with load for six different types of fuels. As it can be observed, the peak pressure increases with increasing the engine load. The reason behind that is that the mixing charge becomes richer at higher engine loads resulting in more fuel burning; consequently the in-cylinder temperature and pressure both become higher.

Also, it is clear that using the 1D and 2D manifold causes the peak pressure to be reduced more in comparison with normal manifold because of the better mixing generated as the result of the swirl motion which enhanced the combustion characteristics. However, the peak pressure was higher when using the 3D manifold because of the much more material (or volume) addition in comparison with the normal manifold design. The use of the 1D and 2D manifolds instead of the normal manifold can decrease the maximum cylinder pressure by an average percentage of 5% and 8% respectively. However, the use of the 3D manifold can result in 1% decrement of in-cylinder peak pressure.

Further notice is that a lower peak pressure is expected when using a fuel with a higher cetane number and this was the case in Figure (45 b) as the GTL fuel has the highest cetane number (=70) amongst other fuels which causes the ignition delay period to be less. The use of GTL fuel with diesel fuel in one fuel blend has also resulted in a reduction in the maximum in-cylinder pressure as can be shown in Figure (45 c). The reason behind that can be related to the higher cetane number of diesel-GTL fuel blend compared to pure diesel fuel which has decreased the ignition delay period resulting in a reduction of the amount of fuel burned during the premixed combustion phase. The cetane number was measured for

this blend to be equal to 62.5 which is considered to be higher than the cetane number of other used fuel blends.

The waste cooking oil which has a very low cetane number (= 32.5) caused the pressure rise in the cylinder to be higher when it was blended with diesel and GTL as shown in Figure (45 d) and Figure (45 e) in which the cetane number was measured for those fuel blends to be 43.8 and 52.5 respectively. The use of corn oil in companion with waste cooking oil in one fuel blend with the conventional fuels which is the case of Figure (45 f) has further increased the pressure rise in the cylinder as the corn oil has also a low cetane number (=37.6) .

Also, the high calorific value of GTL fuel (=47.3 MJ/kg) has resulted in better engine combustion characteristics in terms of in-cylinder peak pressure and the maximum in-cylinder pressure rise rate in comparison to other used fuels which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equals to 42.7 MJ/kg.

Furthermore, GTL fuel has the lowest kinematic viscosity (=2 mm^2/s) in comparison to other used fuel blends. The kinematic viscosity was measured for other fuels at STP and was found to be equals to 5.2 mm^2/s for diesel fuel, 4.9 mm^2/s for waste cooking oil and 45 mm^2/s for corn oil. The very high viscosity of corn oil explains why the maximum cylinder pressure readings were the highest in Figure (45 f). As the fuel is less viscous, the fuel droplets become smaller which enhance the air-fuel mixing and thus enhances the engine combustion characteristics and vice versa.

4.1.2 Engine Performance

1) Brake Specific Fuel Consumption @ constant load

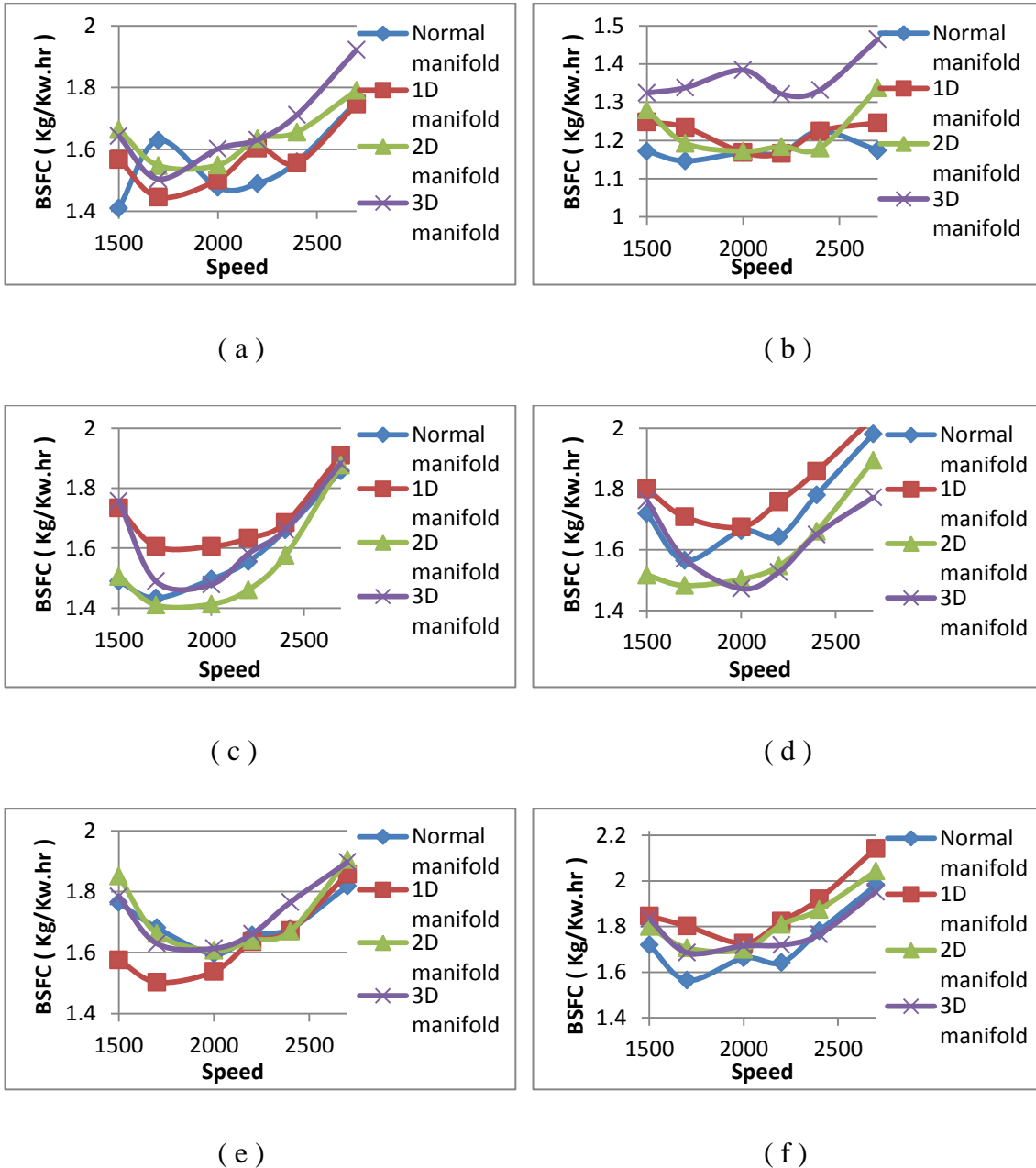


Figure 46. Brake specific fuel consumption vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Figure 46 shows the relation between brake specific fuel consumption with respect to engine speed for six different types of fuel. Bsfcr is an indication of the quality of air-fuel mixing and the rate of fuel consumption. It can be seen that there is a slight improvement in engine fuel consumption with the use of the 1D and 2D manifolds. This enhancement in bsfcr can be seen for most of the cases above. This enhancement in bsfcr indicates that the use of 1D and 2D manifolds can enhance the quality of air-fuel mixing which results in better combustion and less fuel consumption. However, the use of the 3D manifold does not give lower bsfcr readings than the normal manifold. Main reason for that is the increase in the intake manifold pressure. Due to this increase, more air entered into the cylinder in such a way that the relative air ratio causes more fuel consumption [62]. The use of the 1D and 2D manifolds is better in reducing the brake specific fuel consumption by an average percentage of 5% and 7% respectively at a constant engine load. However the use of the 3D manifold can result in 8% increment of bsfcr.

The high calorific value for GTL fuel (=47.3 MJ/kg) has resulted in less fuel consumption as shown in Figure (46 b) in comparison to other used fuels which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equals to 42.7 MJ/kg. As the fuel calorific value becomes higher, less amount of fuel has to be burned in order to produce the same amount of combustion energy and thus more fuel is saved.

Furthermore, GTL fuel has the lowest kinematic viscosity ($=2 \text{ mm}^2/\text{s}$) in comparison to other used fuel blends which has resulted in lower fuel consumption rates. The kinematic viscosity was measured for other fuels at STP and was found to be equals to $5.2 \text{ mm}^2/\text{s}$ for

diesel fuel, $4.9 \text{ mm}^2/\text{s}$ for waste cooking oil and $45 \text{ mm}^2/\text{s}$ for corn oil. The very high viscosity of corn oil explains why the brake specific fuel consumption readings were the highest in Figure (46 f). As the fuel is less viscous, the fuel droplets become smaller which enhance the air-fuel mixing and thus decreases the amount of fuel consumption.

2) Volumetric Efficiency @ constant load

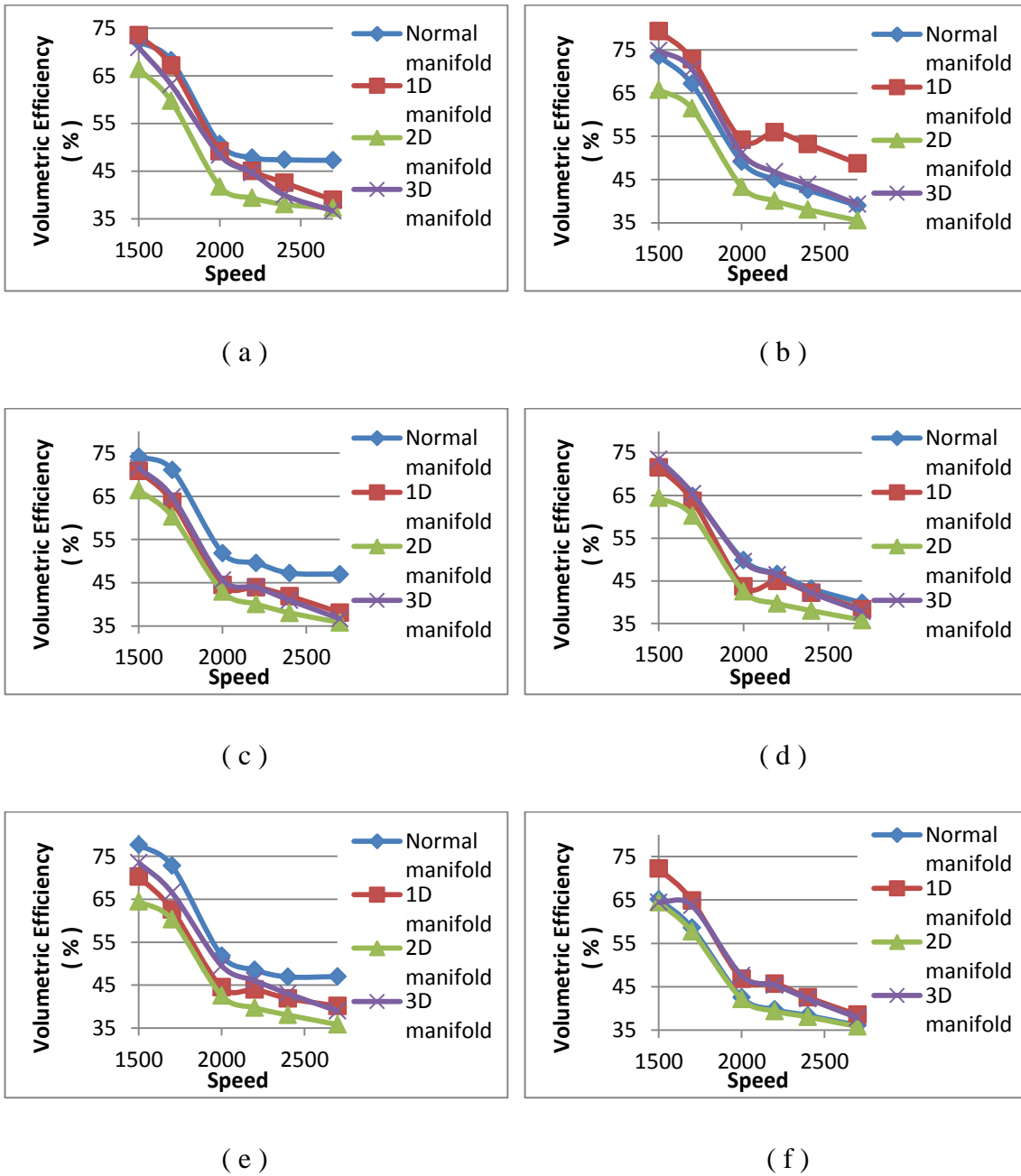


Figure 47. Volumetric efficiency vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Volumetric efficiency is a measure of how good the engine is at receiving in air, and anything that reduces the flow of air into the engine will lower the volumetric efficiency. The two biggest reasons for poor volumetric efficiency at high engine speeds are frictional flow losses and choked flow. As it can be noticed in Figure 47, the volumetric efficiency readings for the 1D manifold are almost similar or higher than the normal manifold in most of the cases due to the pulsating behavior that is developed with the increase in length and volume of the induction manifold. However, the use of the 2D and 3D manifolds has resulted in lower volumetric efficiency. This can be referred to the high reduction in volumetric efficiency when using those new manifolds due to their high air flow resistance. The use of 1D manifold instead of the normal manifold can enhance the volumetric efficiency by an average percentage of 10 %. However the use of the 2D and 3D manifolds can result in 10% and 3% decrement in volumetric efficiency respectively.

The use of waste cooking oil and corn oil in fuel blends is found to be effective for increasing the engine volumetric efficiency. This can be referred to the high oxygen content inherited in the waste cooking oil and corn oil in comparison to conventional fuels diesel and GTL. Consequently, less amount of air needed to initiate the combustion and the volumetric efficiency reading gets higher as it can be shown in Figure (47 d) through Figure (47 f).

3) Exhaust Temperature @ constant load

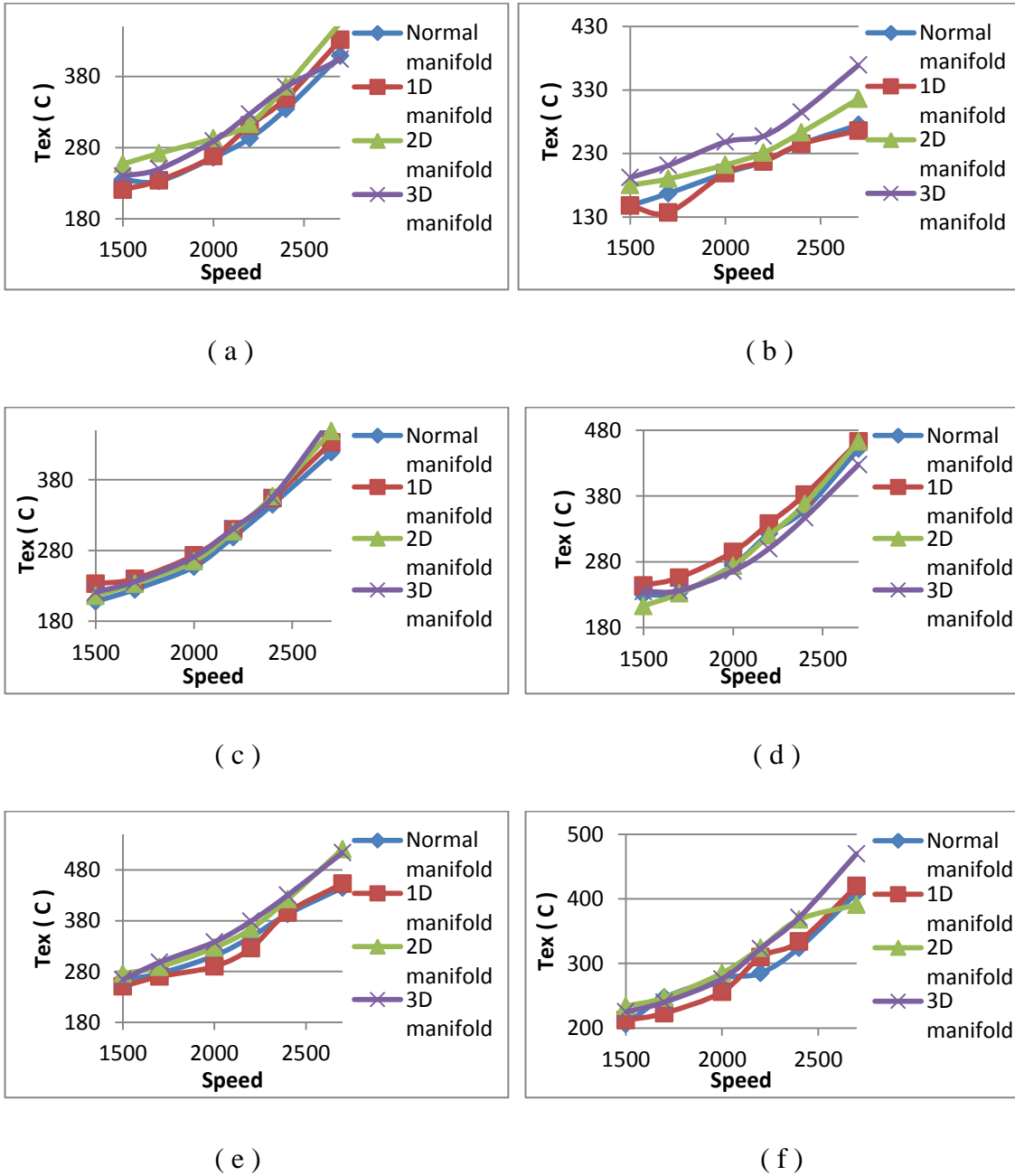


Figure 48. Exhaust temperature vs. engine speed for:(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Figure 48 depicts the variation of exhaust gas temperature for the normal manifold and the three new manifolds for different fuels and engine speeds. Exhaust gas temperature indicates that the heat in the cylinder is converted into work. It is also an indication of the complete combustion inside the cylinder. At various speed conditions, it is observed that the exhaust gas temperature increases with speed because the mixing charge becomes richer with speed which causes more fuel to be burned. As noticed, the exhaust gas temperature is slightly higher for the 1D manifold and the 3D manifold in most of the cases which indicates that the new manifolds do enhance the mixing quality and generate combustion that is more complete. The use of the 1D and 3D manifolds instead of the normal manifold can increase the exhaust gas temperature by an average percentage of 12% and 6% respectively. However the use of the 2D manifold can result in 7% decrement in the exhaust temperature.

GTL fuel has the lowest kinematic viscosity ($=2 \text{ mm}^2/\text{s}$) in comparison to other used fuel blends and this has raised the air-fuel mixing quality and resulted in a more complete combustion and higher exhaust temperature reading as noticed in Figure (48 b). The kinematic viscosity was measured for other fuels at STP and was found to be equals to $5.2 \text{ mm}^2/\text{s}$ for diesel fuel, $4.9 \text{ mm}^2/\text{s}$ for waste cooking oil and $45 \text{ mm}^2/\text{s}$ for corn oil. The very high viscosity of corn oil explains why the exhaust temperature readings were the highest in Figure (48 f). However, the use of other fuel blends that has almost a similar viscosity to diesel fuel has given very close exhaust temperature readings to it as it can be noticed from Figure (48 c) through Figure (48 e).

Also, the high calorific value of GTL fuel ($=47.3$ MJ/kg) has resulted in better engine combustion and higher exhaust temperature readings in comparison to other used fuels which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equal to 42.7 MJ/kg.

1) Brake Specific Fuel Consumption @ constant speed

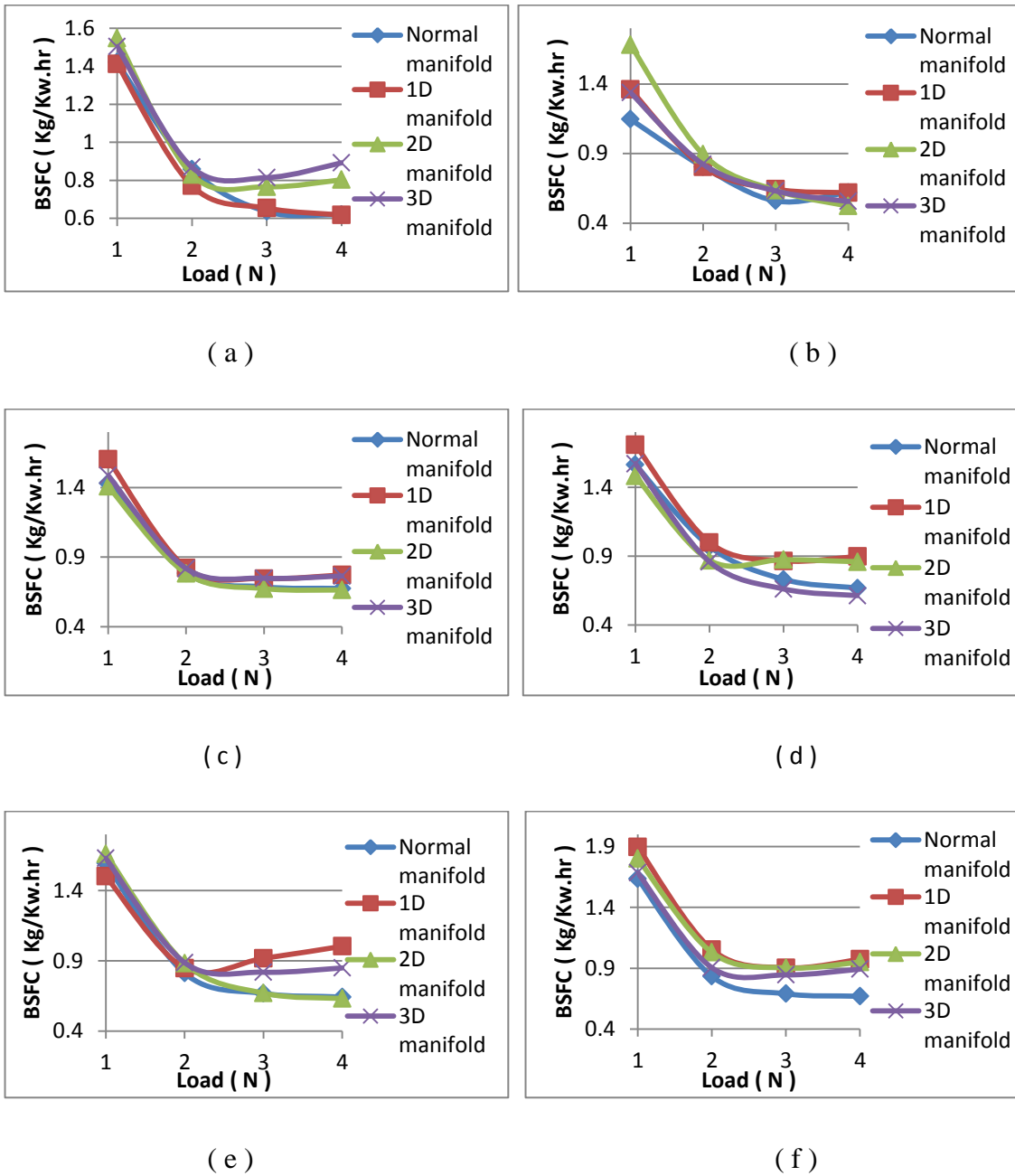


Figure 49. Brake specific fuel consumption vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

The bsfc is a measure of engine efficiency and it indicates for a less fuel consumption as it decreases more. In fact, bsfc and engine brake thermal efficiency are inversely related, so that the lower the bsfc the higher is the thermal efficiency of the engine. The variation of brake specific fuel consumption at different loads for the normal manifold and the three new manifolds is shown in Figure 49. It can be seen that there is a slight improvement in engine fuel consumption with the use of the 1D manifold in most of the cases. This enhancement in bsfc indicates that the use of 1D manifold can enhance the quality of air-fuel mixing which results in better combustion and less fuel consumption. It is also observed that the use of the 2D manifold gives slightly higher bsfc which indicates that it can also give good air-fuel mixing and results in low fuel consumption. However, the use of the 3D manifold was found to cause low quality combustion and high fuel consumption. The use of the 1D instead of the normal manifold can reduce the brake specific fuel consumption by an average percentage of 5%. However the use of 2D and 3D manifolds can result in 21% and 28% increment in bsfc.

The high calorific value for GTL fuel ($=47.3$ MJ/kg) has resulted in less fuel consumption as shown in Figure (49 b) in comparison to other used fuels which has lower calorific value. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equals to 42.7 MJ/kg. As the fuel calorific value becomes higher, less amount of fuel has to be burned in order to produce the same amount of combustion energy.

Furthermore, GTL fuel has the lowest kinematic viscosity ($=2$ mm²/s) in comparison to other used fuel blends which has resulted in lower fuel consumption rates. The kinematic viscosity was measured for other fuels at STP and was found to be equals to 5.2 mm²/s for

diesel fuel, 4.9 mm²/s for waste cooking oil and 45 mm²/s for corn oil. The very high viscosity of corn oil explains why the brake specific fuel consumption readings were the highest in Figure (49 f). As the fuel is less viscous, the fuel droplets become smaller which enhance the air-fuel mixing and thus decreases the amount of fuel consumption.

2) Volumetric Efficiency@ constant speed

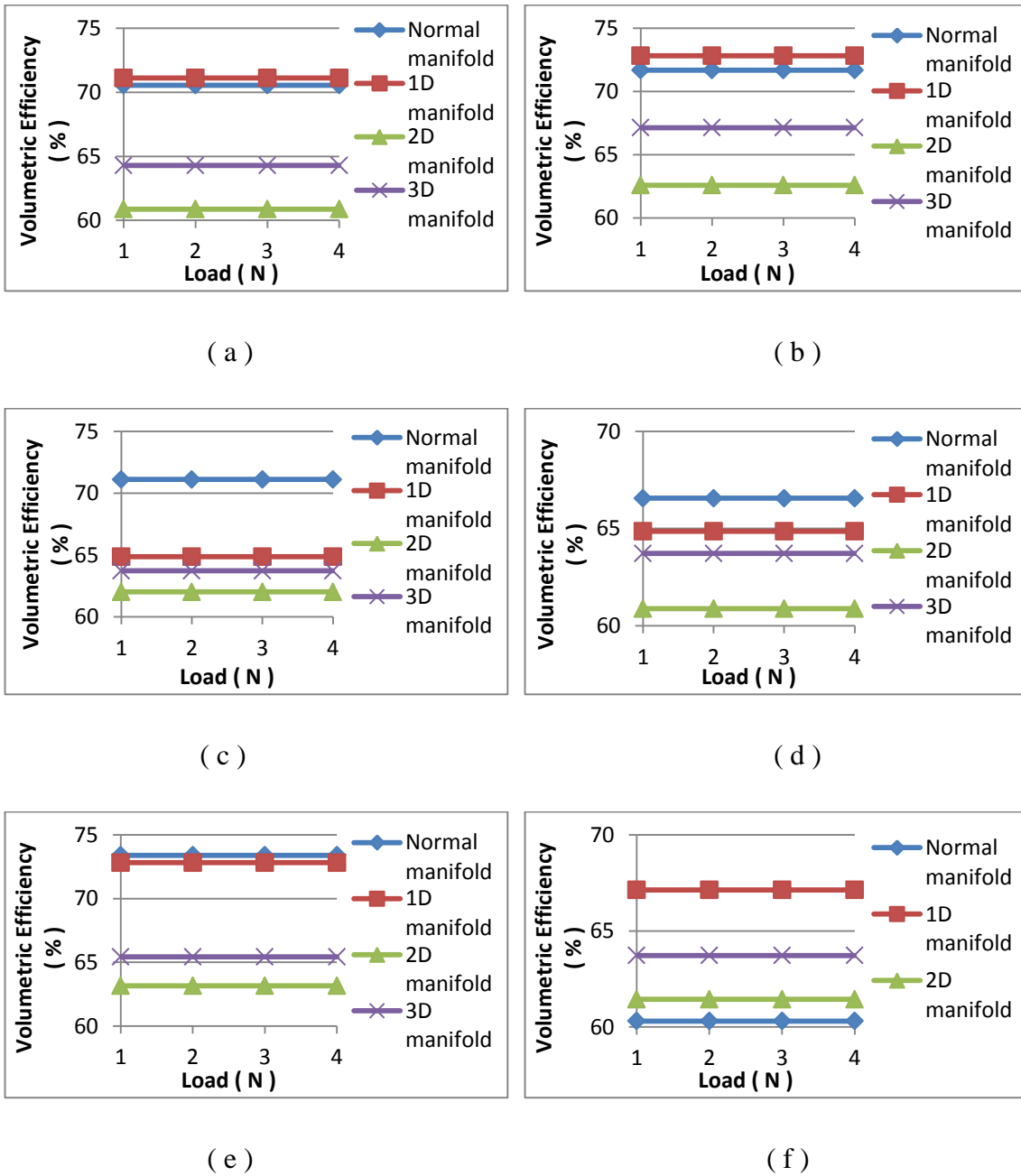


Figure 50. Volumetric efficiency vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

As it can be noticed in Figure 50 that the volumetric efficiency readings for the 1D manifold are almost similar or higher than the normal manifold in most of the cases due to the pulsating behavior that is developed with the increase in length and volume of the induction manifold. However, the use of the 2D and 3D manifolds has resulted in lower volumetric efficiency. This can be referred to the high reduction in volumetric efficiency when using those new manifolds due to their high air flow resistance. The use of 1D manifold instead of the normal manifold can enhance the volumetric efficiency by an average percentage of 4%. However the use of the 2D and 3D manifolds can result in 11% and 8% decrement in volumetric efficiency respectively.

It can be also observed that the volumetric efficiency readings in Figure (50 e) and Figure (50 f) are the highest. The most probable reason for that is the high oxygen content inherited in the waste cooking oil and corn oil, thus less amount of air needed to initiate the combustion and the volumetric efficiency readings get higher in comparison to conventional fuels diesel and GTL that are poor of oxygen content as shown in Figure (50 a) through Figure (50 c).

3) Exhaust Temperature @ constant speed

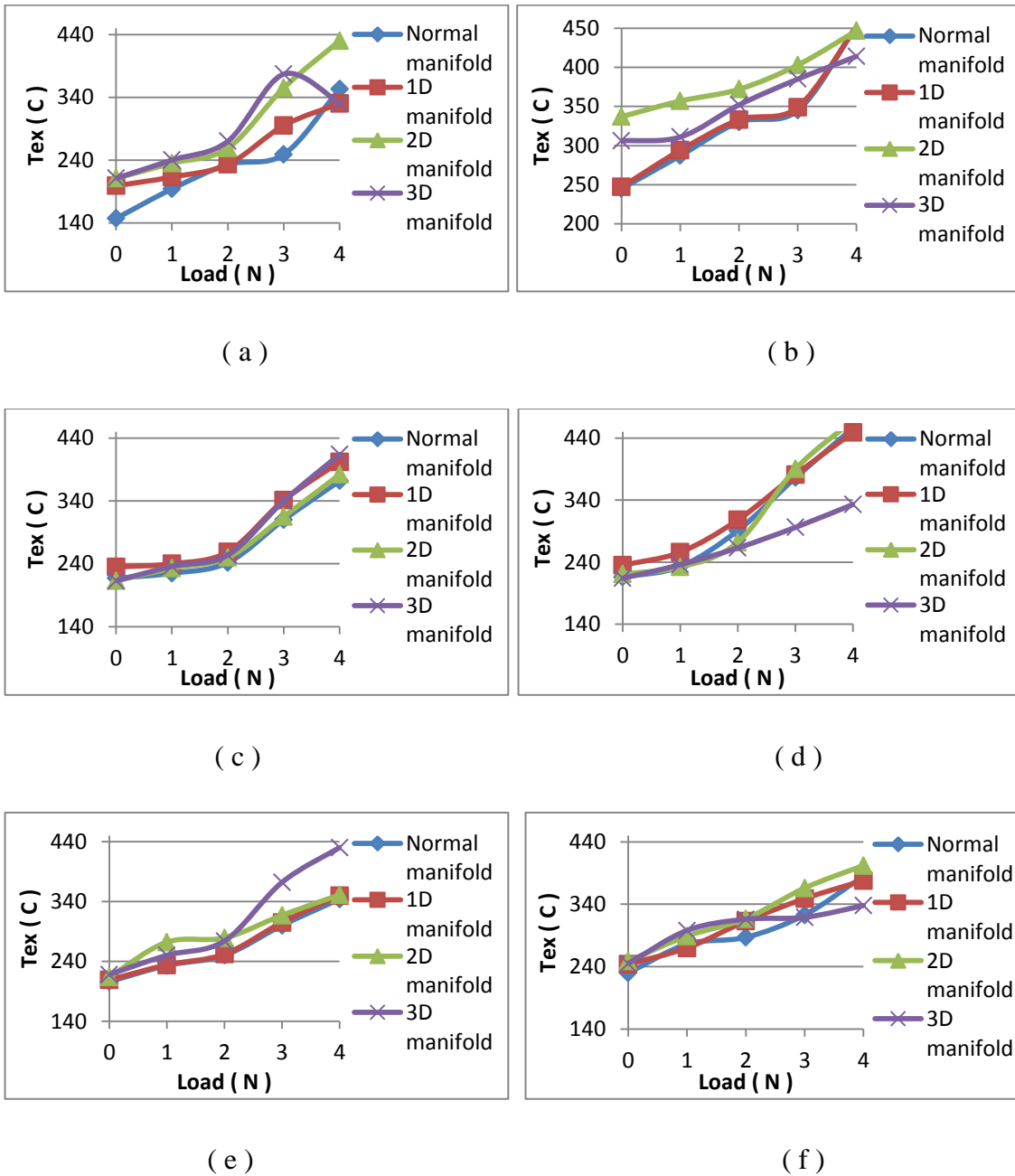


Figure 51. Exhaust temperature vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Figure 51 depicts the variation of exhaust gas temperature for the normal manifold and the three new manifolds at different engine loads and using different types of fuels. Exhaust gas temperature indicates that the heat in the cylinder is converted into work. At various load conditions, it is observed that the exhaust gas temperature increases with load because more fuel is burnt to meet the power requirement. As it can be noticed from Figure 51 that the exhaust gas temperature is higher for the 1D manifold and the 3D manifold in most of the cases which indicates that the new manifolds do enhance the mixing quality and generate combustion that is more complete. The use of the 1D and 3D manifolds instead of the normal manifold can increase the exhaust gas temperature by an average percentage of 7%. However the use of the 2D manifold can result in 11% decrement in the exhaust temperature.

GTL fuel has the lowest kinematic viscosity ($=2 \text{ mm}^2/\text{s}$) in comparison to other used fuel blends and this has raised the air-fuel mixing quality and resulted in a more complete combustion and higher exhaust reading as noticed in Figure (51 b) and also when it was blended with diesel fuel in Figure (51 c). The kinematic viscosity was measured for other fuels at STP and was found to be equals to $5.2 \text{ mm}^2/\text{s}$ for diesel fuel, $4.9 \text{ mm}^2/\text{s}$ for waste cooking oil and $45 \text{ mm}^2/\text{s}$ for corn oil. The very high viscosity of corn oil explains why the exhaust temperature readings were the highest in Figure (51 f). However, the use of other fuel blends that has almost a similar viscosity to diesel fuel has given very close exhaust temperature readings to it as it can be noticed from Figure (51 d) and Figure (51 e).

Also, the high calorific value for GTL fuel ($=47.3 \text{ MJ/kg}$) has resulted in better engine combustion and higher exhaust temperature readings in comparison to other used fuels

which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equal to 42.7 MJ/kg.

4.1.3 Engine Emissions

1) Carbon Monoxide @constant load

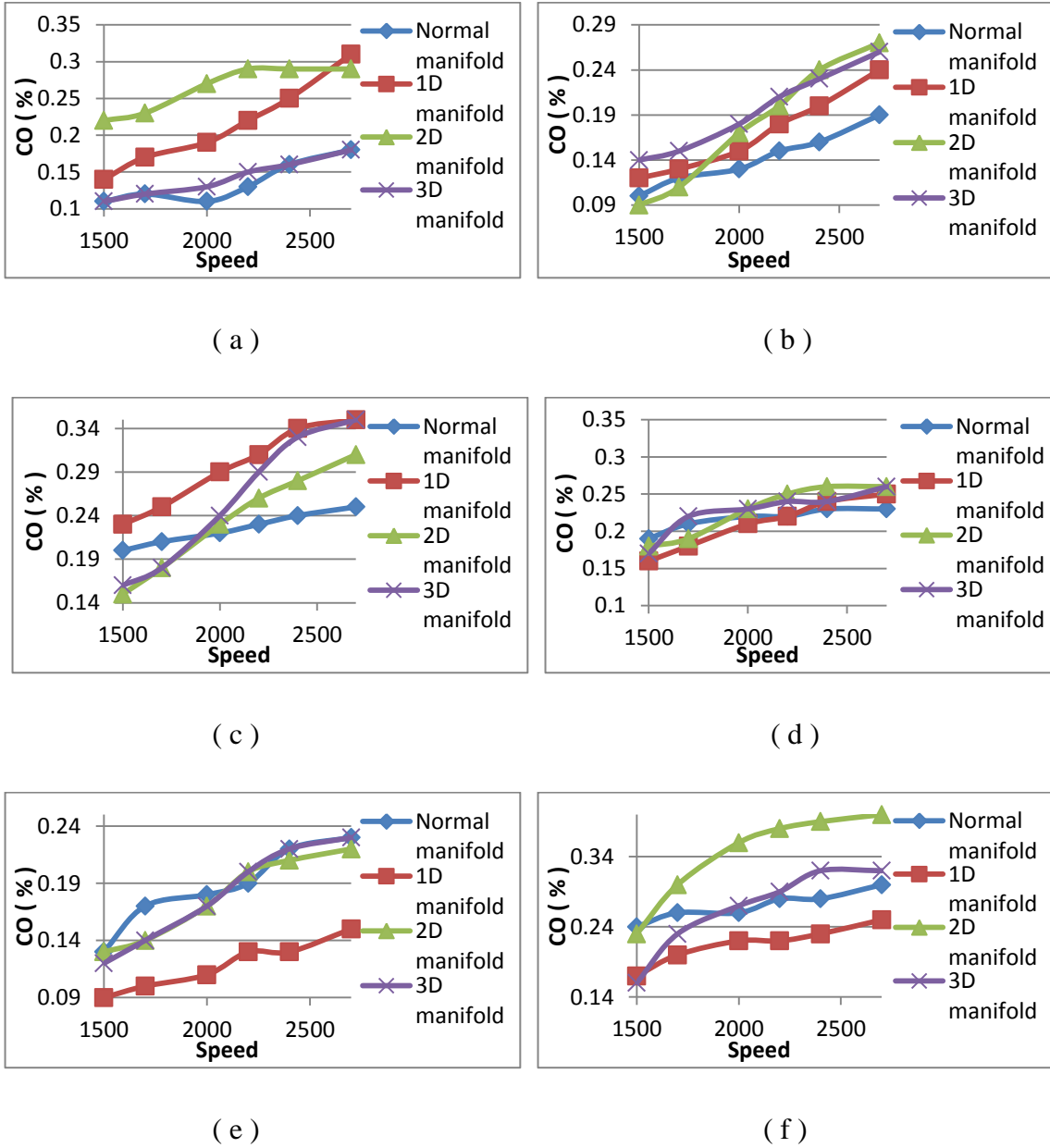


Figure 52. Carbon monoxide vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

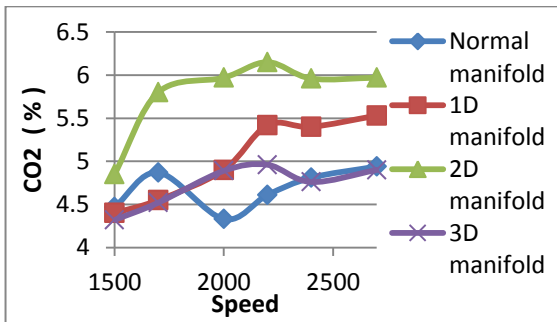
Carbon monoxide occurs only in the engine exhaust and it is resulted as the product of incomplete combustion. Higher CO emission rates are expected at higher engine speeds as the air-fuel mixture becomes richer and the air to fuel ratio decreases more [65]. Moreover, at higher engine speeds there will be less time available for air-fuel mixing and an increased quantity of injected fuel which further increases CO emission. As noticed in Figure 52, the use of 1D manifold has decreased carbon monoxide emission in most of the cases which indicates that it gives a better air-fuel mixing and results in a more complete combustion as a result of swirl motion and the high TKE that enhances the quality of air-fuel mixing. However, the use of 2D and 3D manifolds has reduced CO emission rates only at low engine speeds but was found not to be effective at higher engine speeds where it has increased CO emission rates more than the normal manifold.

Also, it can be clearly noticed that the CO emission rates are the lowest in Figure (52 b) and Figure (52 e). In Figure (52b), the use of GTL fuel which has the highest cetane number (=70) amongst other used fuels has resulted in better combustion characteristics and a more complete combustion. Also, GTL low viscosity played a major role in enhancing the air-fuel mixing which has resulted in a higher quality of air-fuel mixing and thus less CO emission. The use of 1D manifold instead of normal manifold can decrease the carbon monoxide emission by an average percentage of 12%. However the use of 2D and 3D manifolds can result in 31% and 38% increment of CO emission at low engine speeds.

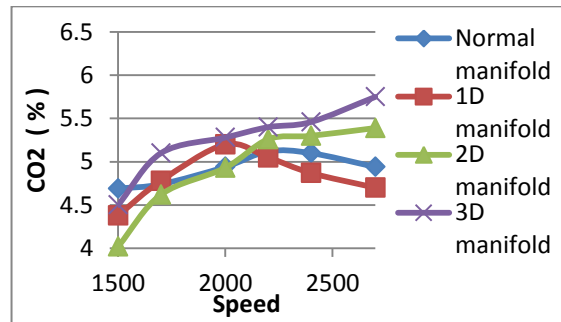
The use of diesel - GTL - waste cooking oil fuel blend in Figure (52 e) was also found to be effective in decreasing CO emission. The reason behind that can be referred to the high oxygen content that is inherited in waste cooking oil molecules, thus more oxygen is

available in the air-fuel mixture and the charge becomes leaner. Consequently, CO emission rate decreases. Also, the addition of waste cooking oil to pure diesel in a 50 to 50 volume percentage has recorded a remarkable reduction in CO as Figure (52 d) indicates. Although the corn oil has high oxygen content, however its very high viscosity has affected the mixing quality more and a poor air-fuel mixing was obtained. This fact interprets why the CO emission in Figure (52 f) was higher than other used fuels.

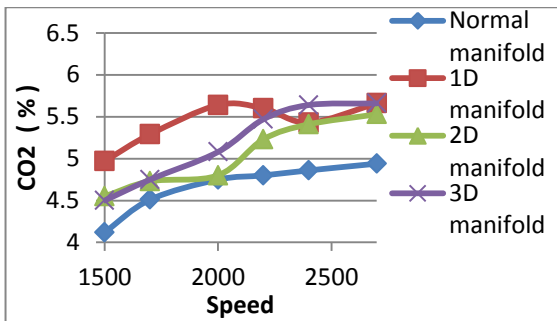
2) Carbon Dioxide @constant load



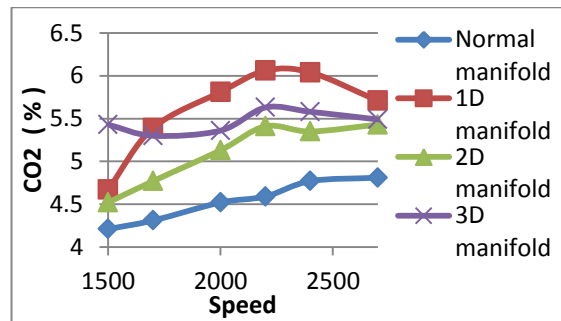
(a)



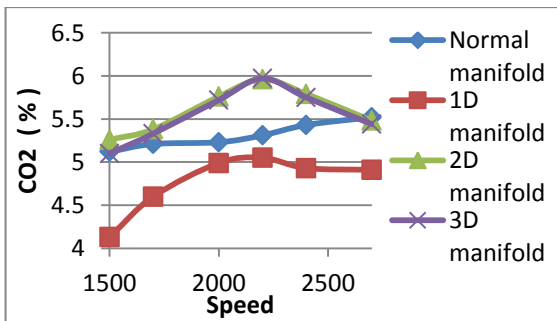
(b)



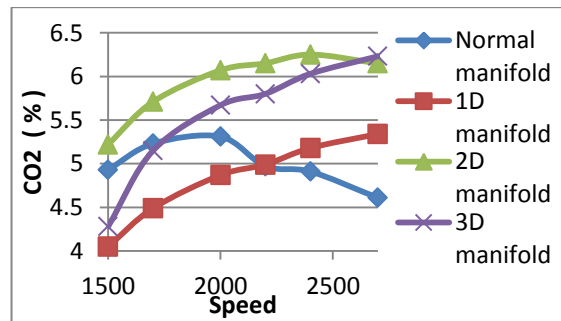
(c)



(d)



(e)



(f)

Figure 53. Carbon dioxide vs. engine speed for : (a) diesel fuel , (b) GTL fuel , (c) diesel-GTL blend, (d) diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

The production of carbon dioxide in the exhaust tailpipe is a function of Hydrogen to Carbon ratio of the fuel and it depends also on the chemical conversion of CO into CO₂. It can be clearly noticed in all the graphs that CO₂ is increasing as the engine speed increases. The reason behind that is that the equivalence ratio is getting more as the engine speed increases, thus a greater amount of CO is allowed to convert into CO₂. [66]

As noticed in Figure 53, the use of 1D manifold has decreased carbon dioxide emission in most of the cases which indicates that it gives a better air-fuel mixing and results in a more complete combustion as a result of swirl motion and the high TKE that enhances the quality of air-fuel mixing. However, the use of 2D and 3D manifolds was found not to be effective in decreasing CO₂ emission rates which indicates that it results in a lower air-fuel mixing quality. This can be referred to the increased volume of the intake manifold in which the incoming air becomes more pressurized and consumes a higher amount of fuel to match the required equivalence ratio, hence the mixture becomes richer and more CO and CO₂ are produced in the exhaust [62, 63]. The use of 1D manifold instead of normal manifold can decrease the carbon dioxide emission by an average percentage of 18% at a constant engine load. However the use of 2D and 3D manifolds can result in 10% and 9% increment of CO₂ emission.

The use of diesel and GTL in the pure as in Figure (53 a) and Figure (53 b) or in the blended form as in Figure (53 c) has resulted in moderate CO₂ emission rates as they have a high Hydrogen to Carbon ratio (=2.125 for diesel and 2.15 for GTL). However, the addition of waste cooking oil to diesel-GTL fuel blend in Figure (53 e) and further corn oil

to that fuel blend in Figure (53 f) has resulted in more reduction of CO₂ emission as those biofuels have a high amount of oxygen content inherited in their molecules. Further notice that the use of waste cooking oil with diesel in one fuel blend is also found to be effective in decreasing CO₂ emission, more specifically at high engine speeds (above 2100 rpm).

3) Hydrocarbons @constant load

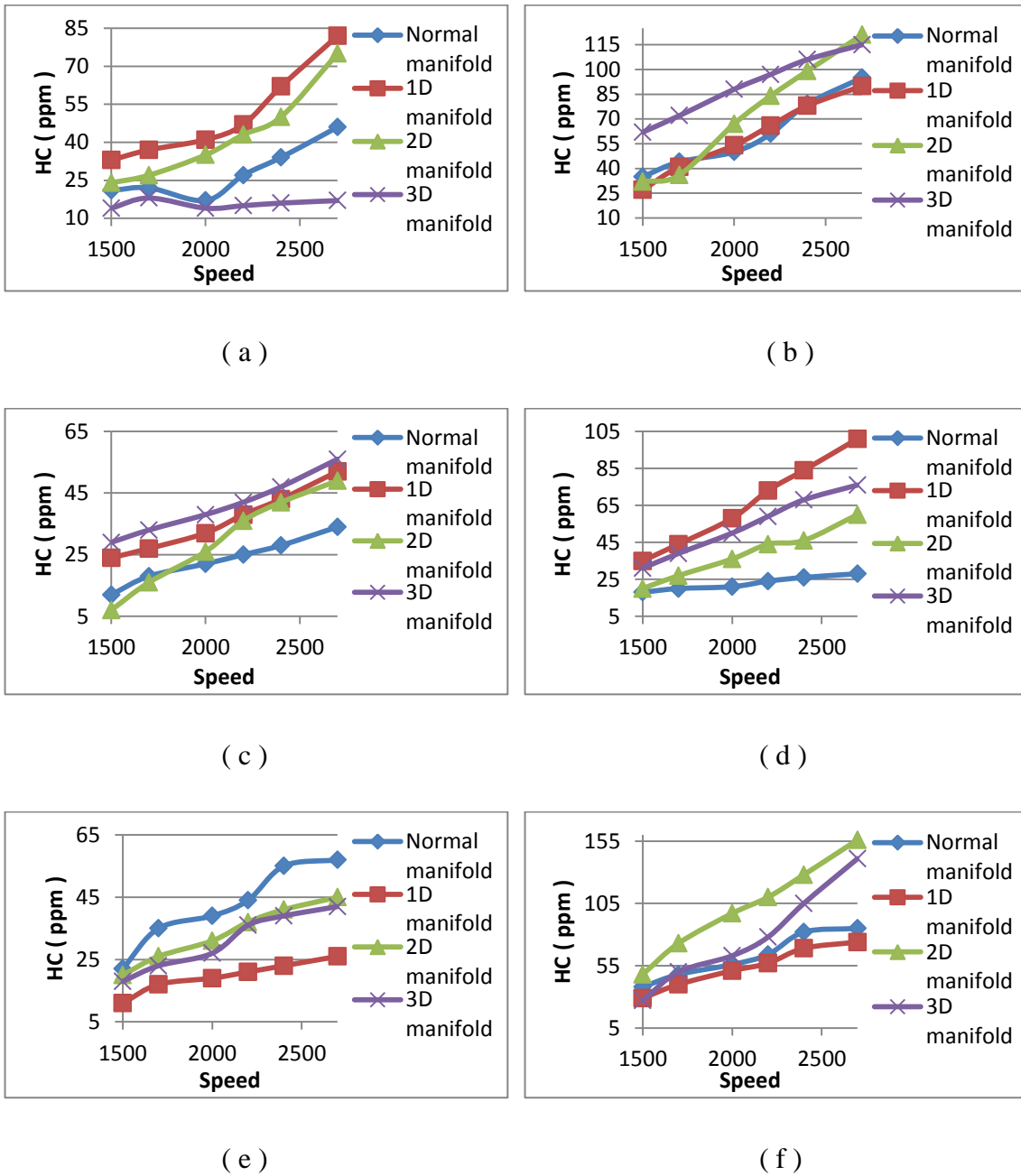


Figure 54. Hydrocarbons emission vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Figure 54 demonstrates the relation between unburned hydrocarbon concentrations with speed increment for six different types of fuels. Unburned HC results mainly because of incomplete combustion of fuel and air and from mixtures that are too lean to combust. One more source for HC formation is when HC becomes in a direct contact with the walls of the combustion chamber and becomes quenched, more specifically during the cold start period.

It can be observed from Figure 54 that the use of 1D manifold has decreased CO emission in most of the cases. This can be related to the better air-fuel mixing generated as a result of the swirl motion which caused the combustion to be more complete. However, the use of 2D and 3D manifolds was found not to be effective in decreasing HC emission. This conclusion indicates that the use of 2D and 3D manifolds causes the mixing charge to be too lean for combustion and thus more HC is produced at a constant engine load operation. The use of 1D manifold instead of normal manifold can decrease hydrocarbon emission by an average percentage of 19%. However the use of 2D and 3D manifolds can result in 44% and 68% increment in HC emission.

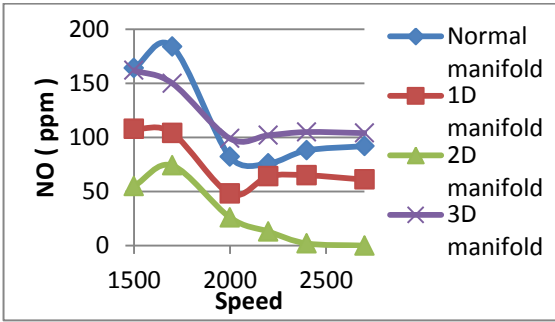
It can be clearly noticed that the use of diesel-waste cooking fuel blend in Figure (54 d) is found to be effective at low engine speeds due to the high distillation temperature of waste cooking oil (around 318 C) which results in less volatile fuel blend ,thus less amount of fuel droplets are quenched and HC emission is reduced.

The very low viscosity of GTL fuel in Figure (54 b) in comparison to other used fuels caused the HC emission to be higher because HC results from fuel that over penetrates and wets the cylinder walls during the ignition delay period .As the fuel is less viscous; it is more able to penetrate the cylinder walls and thus results in more HC emission. The use of

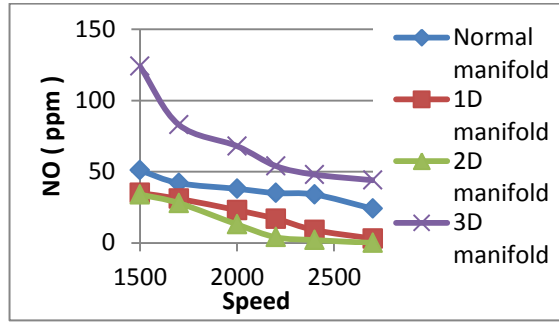
diesel-GTL fuel blend in Figure (54 c) has decreased HC emission by a considerable amount. This may be referred to the improved cetane number of the mixture where less amount of fuel penetrates the cylinder walls due to the shortened ignition delay period. However, the addition of waste cooking oil to this blend in Figure (54 e) was found to be very effective in reducing HC emission due to the improved distillation temperature of the mixture, thus less amount of fuel is quenched during the engine early operation.

Nevertheless, the addition of corn oil to that blend in Figure (54 f) has raised HC emission in comparison to diesel fuel because of its very high viscosity leading to a more incomplete combustion and thus more HC emission. In addition to that, the corn oil has a very low distillation temperature (=160 C) which causes more evaporation of fuel during the cold start period and more fuel is quenched resulting in more HC emission.

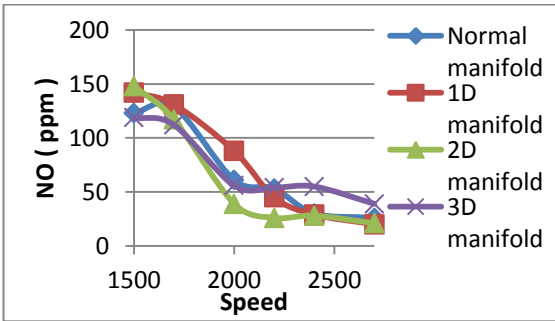
4) Nitric Oxide @constant load



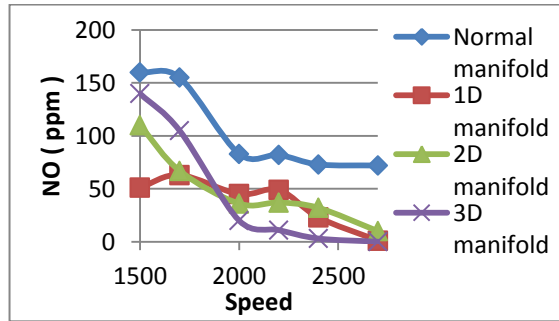
(a)



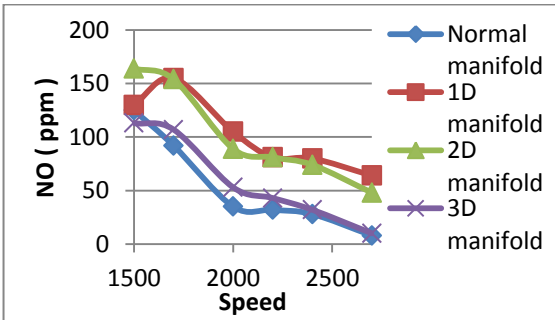
(b)



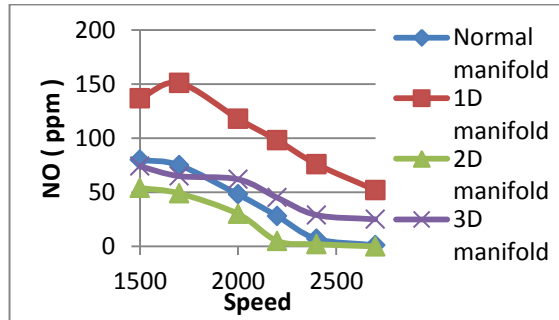
(c)



(d)



(e)



(f)

Figure 55. Nitric oxide emission vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Nitric oxide (NO) is formed during the combustion of oil by two mechanisms; high-temperature thermal fixation of molecular oxygen (O₂) and nitrogen (N₂) present in the combustion air and, second, reaction of atmospheric oxygen with nitrogen-containing compounds in the fuel. NO is the major component in the NO_x emission that is why the focus is mainly about it. As the engine operates at higher speed, NO emission becomes lower. The reason behind that is that the flame temperature decreases with more engine speed which results in less NO emission.

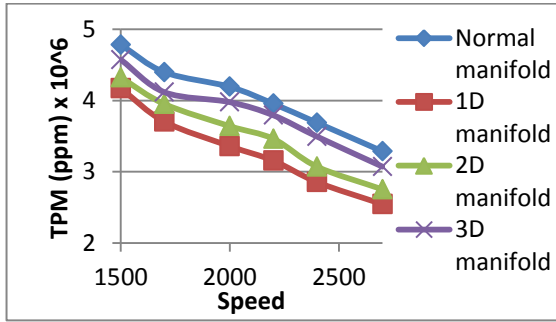
It can be noticed from Figure 55 that the use of 1D and 2D manifolds has decreased NO emission in most of the cases. This can be related to the lower in-cylinder peak pressure and maximum pressure raise rate which is obtained when using the 1D and 2D manifolds. Consequently, both the ignition delay period and the flame temperature are less. However, the use of 3D manifold was found not to be effective in decreasing NO emission as the ignition delay period and the flame temperatures are higher due to its increased volume which has resulted in more cylinder pressure [62]. The use of 1D and 2D manifolds instead of normal manifold can decrease Nitric oxide emission by an average percentage of 32% and 43% respectively. However the use of the 3D manifold can result in 41% increment of NO emission.

The values of NO emission at different engine speeds were lowest when using GTL fuel as it has the highest cetane number (=70) which causes shorter ignition delay and less peak pressure as shown in Figure (55 b). Also, diesel and GTL fuel blend is considered to be a good solution for reducing NO emission in the cylinder when compared with the use of

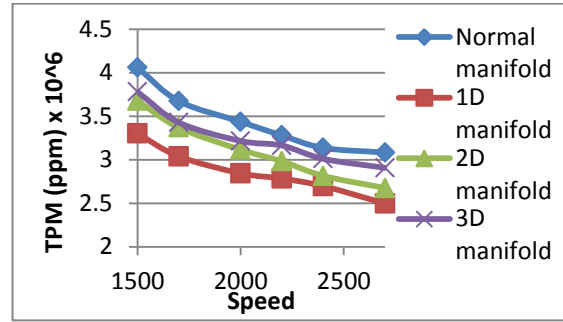
pure diesel as Figure (55 c) indicates. The cetane number was measured for this blend to be equals to 62.5 which is considered to be high with respect to other used fuel blends.

The waste cooking oil which has a very low cetane number (= 32.5) caused the NO emission to be higher when it was blended with diesel and GTL as shown in Figure (55 d) and Figure (55 e) in which the cetane number was measured for those fuel blends to be 43.8 and 52.5 respectively. In the same principle, the addition of corn oil to the aforementioned fuel blend in Figure (55 f) has also resulted in higher NO emission rates due to its low cetane number.

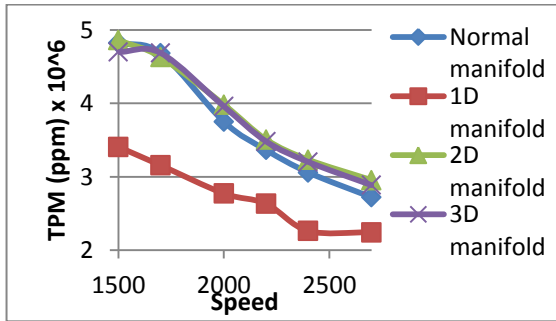
5) Total Particulate Matters @constant load



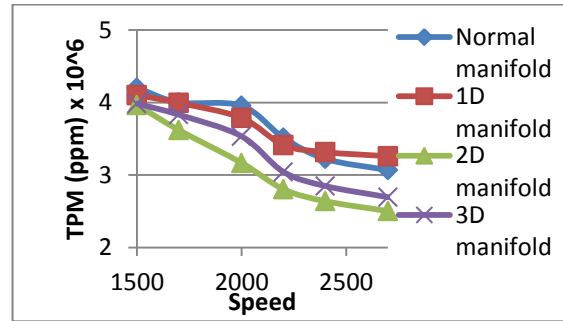
(a)



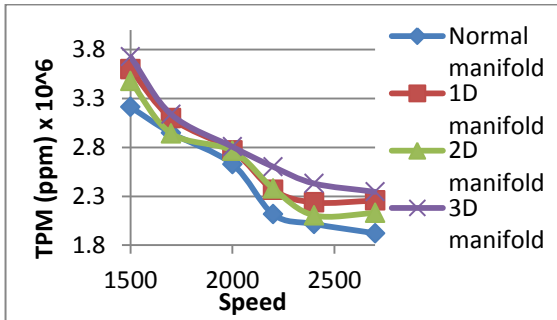
(b)



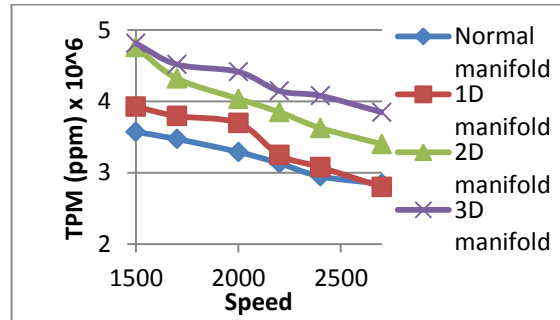
(c)



(d)



(e)



(f)

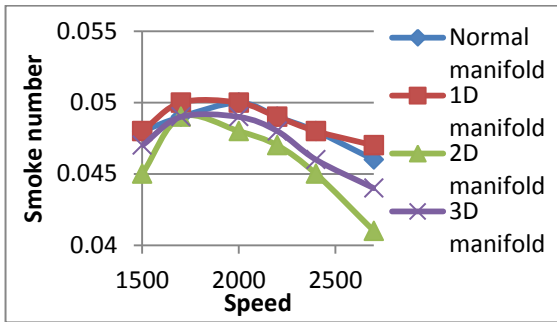
Figure 56. Particulate emission vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Incomplete combustion is the main reason of the particulate matters emission in diesel engine. Particulate matters are formed mainly in the fuel rich regions during the diffusion burning period. As it can be noticed in Figure 56, the total amount of different size particles detected in the engine exhaust decreases with increasing engine speed. The reason behind that is that better turbulence effect is obtained at higher engine speeds in which the extent of complete combustion is improved [67].

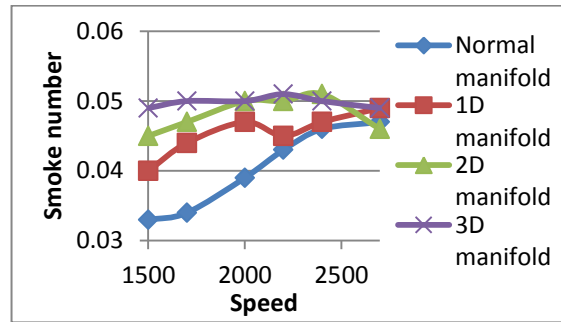
It can be observed from Figure 56 that the use of 1D manifold has decreased CO emission in almost every case. This can be related to the high TKE generated as a result of using a helical and spiral inner diameter and varying the outlet angle to be 30° instead of zero as it is in the standard normal manifold. For the same reason, it was figured out that the use of 2D and 3D manifolds was found to be effective in decreasing PM emission but with fewer amounts than the 1D manifold. This conclusion indicates that the use of 1D manifold is more preferable than the use of 2D and 3D manifolds for decreasing PM emission as it gives higher TKE. The use of the 1D manifold can decrease the PM emission by an average percentage of 15% and the use of 2D and 3D manifolds can also decrease the PM emission by an average percentage of 8% at a constant engine load.

Also, it can be observed that using pure GTL fuel as in Figure (56 b) or mixing it with diesel in one fuel blend as in Figure (56 c) can highly decrease the PM emission as it has a very low –sulfur and aromatics content in comparison to diesel fuel. The addition of a biofuel like the corn oil or using waste cooking oil in fuel blends with the conventional fuels (diesel and GTL) is considered to be a good solution for decreasing PM emission as those alternative fuel blends contains less amount of sulfur, thus less amount of PM are produced in comparison to diesel fuel as can be observed in Figure (56 e) and Figure (56 f).

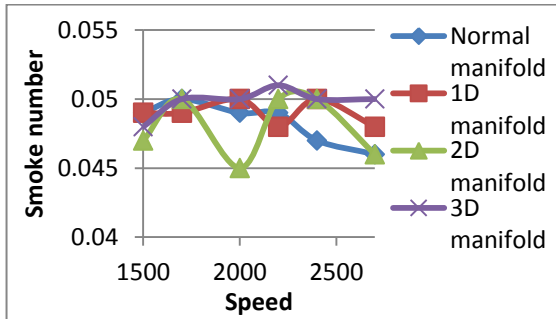
6) Smoke Number @constant load



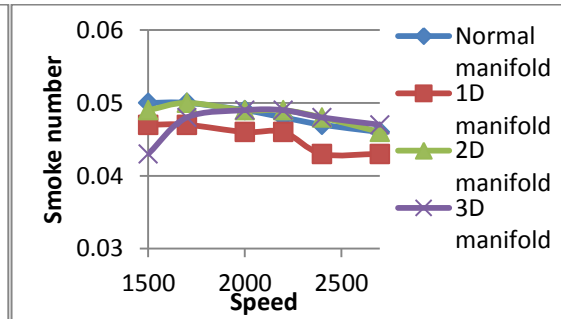
(a)



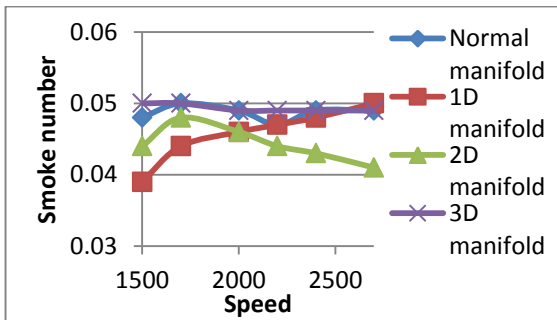
(b)



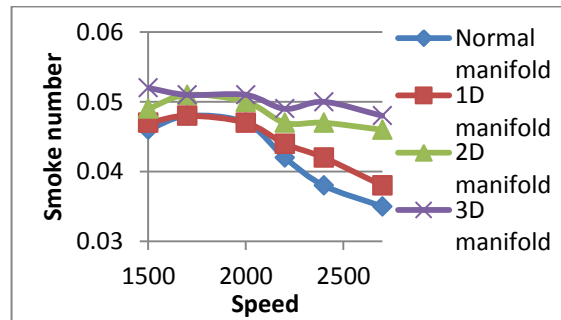
(c)



(d)



(e)



(f)

Figure 57. Smoke emission vs. engine speed for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

The black smoke produced by a fuel is composed of carbon particles released by the thermal cracking of the large hydrocarbon fuel molecules. This process occurs on the rich side of the flame front during the diffusion combustion phase as the PM does. Also, Smoke production is increased at elevated temperatures and this explains why the smoke production in Figure 57 increases with engine speed. Another possible reason is that higher engine speeds lead to a shorter residence time of gases in the combustion chamber.

As observed in Figure 57, the use of 1D manifold has recorded lower readings for the smoke emission in comparison with the normal manifold. This can be referred to the increase in volumetric efficiency that is caused when using the 1D manifold, thus the mixing charge becomes leaner and less amount of smoke can be detected in the exhaust. However, the use of 2D and 3D manifolds was not effective in decreasing smoke production rates as those manifolds can result in lower volumetric efficiencies than the normal manifold. The use of the 1D manifold can decrease the smoke concentration by an average percentage of 5%. However the use of 2D and 3D manifold can result in 2% and 4% increment in smoke concentration.

The use of waste cooking oil and corn oil in one fuel blend with conventional fuels (diesel and GTL) as in Figure (57 d) through Figure (57 f) has decreased the smoke production slightly. The reason behind can be referred to their high oxygen content which plays a major role in reducing soot formation and in soot oxidation [68]. Vice versa, the use of conventional fuels has given almost similar smoke readings as can be shown in Figure (57 a) through Figure (57 c).

1) Carbon Monoxide @constant speed

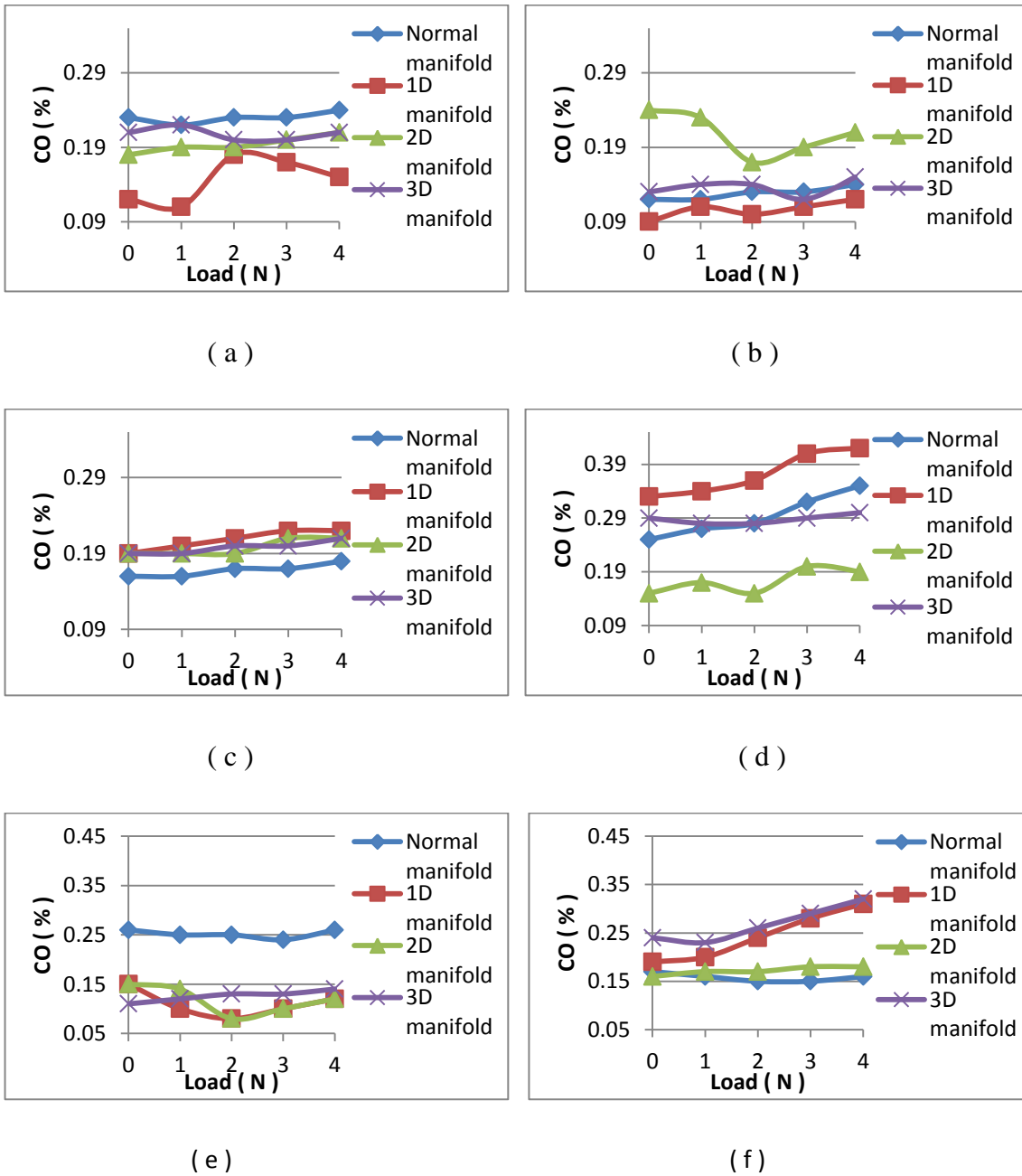


Figure 58. Carbon monoxide vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Carbon monoxide occurs only in the engine exhaust and it results as the product of incomplete combustion. Generally CO emission increases with increasing the load because CO is produced from burning rich air-fuel mixture. At a constant speed engine, the air flow rate is kept constant, however the fuel amount is increasing with load and thus the mixing charge becomes richer. As noticed in Figure 58, the use of 1D and 2D manifolds has decreased carbon monoxide emission in most of the cases which indicates that it gives a better air-fuel mixing and results in a more complete combustion as a result of swirl motion and the high TKE that enhances the quality of air-fuel mixing. However, the use of the 3D manifold has almost given similar readings or slightly less than the normal manifold. This can be referred to the large increment in the manifold volume which played a major role in weakening the TKE which results in less complete combustion. The use of 1D and 2D manifolds instead of normal manifold can decrease the carbon monoxide emission by an average percentage of 48% and 21% at a constant engine speed operation. However the use of 3D manifold can result in 17% increment of CO emission.

Also, it can be clearly noticed that the CO emission rates are the lowest in Figure (58 e) and Figure (58 f). The reason behind that can be referred to the high oxygen content that is inherited in waste cooking oil and corn oil molecules, thus more oxygen is available in the air-fuel mixture and the charge becomes leaner. Consequently, CO emission rate decreases. In the same principle, the use of diesel with waste cooking oil in one fuel blend as in Figure (58 d) was also found to be effective in decreasing CO emission. However, the use of conventional fuels (diesel and GTL) alone without the biofuels has resulted in higher CO emission rates as shown in Figure (58 a) through Figure (58 c).

2) Carbon Dioxide @constant speed

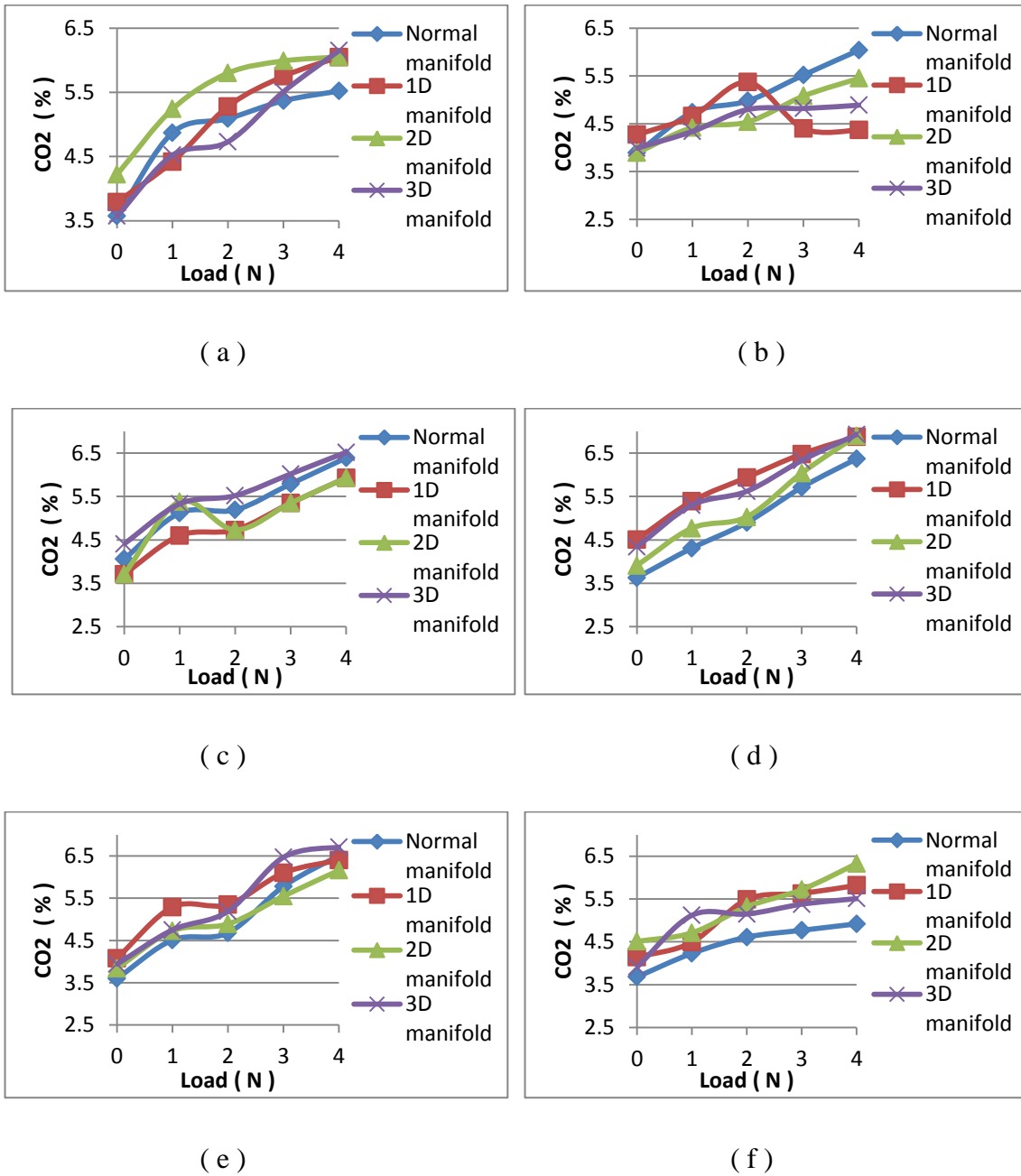


Figure 59. Carbon dioxide vs. engine load for: (a) diesel fuel, (b) GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Figure 59 depicts the relation between Carbon Dioxide emission rates with load increment for six different types of fuel at a constant engine speed operation. It can be clearly noticed in all the graphs that CO₂ readings are increasing as the engine load increases. The reason behind that is that the mixing charge becomes richer as the load increases which allows for more CO and CO₂ to be produced in the exhaust.

As noticed in Figure 59, the use of 1D and 2D manifolds has decreased carbon dioxide emission in most of the cases which indicates that it gives a better air-fuel mixing and results in a more complete combustion as a result of swirl motion and the high TKE that enhances the quality of air-fuel mixing. Also, it is essential to note that the use of 1D manifold has decreased the CO₂ emission rates more than the 2D manifold which proves that the 1D manifold gives higher TKE and leads to a more complete combustion than the 2D manifold. However, the use of the 3D manifold has given almost similar or higher amount of CO₂. This can be referred to the increased volume of the intake manifold in which the incoming air becomes more pressurized and consumes a higher amount of fuel to match the required equivalence ratio, hence the mixture becomes richer and more CO and CO₂ are produced in the exhaust [62, 63]. The use of 1D and 2D manifold instead of normal manifold can decrease the carbon dioxide emission by an average percentage of 19% and 7% respectively at a constant engine speed operation. However the use of 3D manifold can result in 31% increment of CO₂ emission.

The use of diesel and GTL in the pure form has resulted in high CO₂ emission rates as they have low oxygen content in their chemical structure as can be shown in Figure (59 a) through Figure (59 c). However, the addition of waste cooking oil to diesel-GTL fuel blend

in Figure (59 e) and further corn oil to that blend in Figure (59 f) has resulted in a high reduction of CO₂ emission as those biofuels have a high amount of oxygen content inherited in their molecules. In the same principle, it was expected that the addition of waste cooking oil to diesel has reduced CO₂ emission rate more than using pure diesel as noticed in Figure (59 c).

3) Hydrocarbons @constant speed

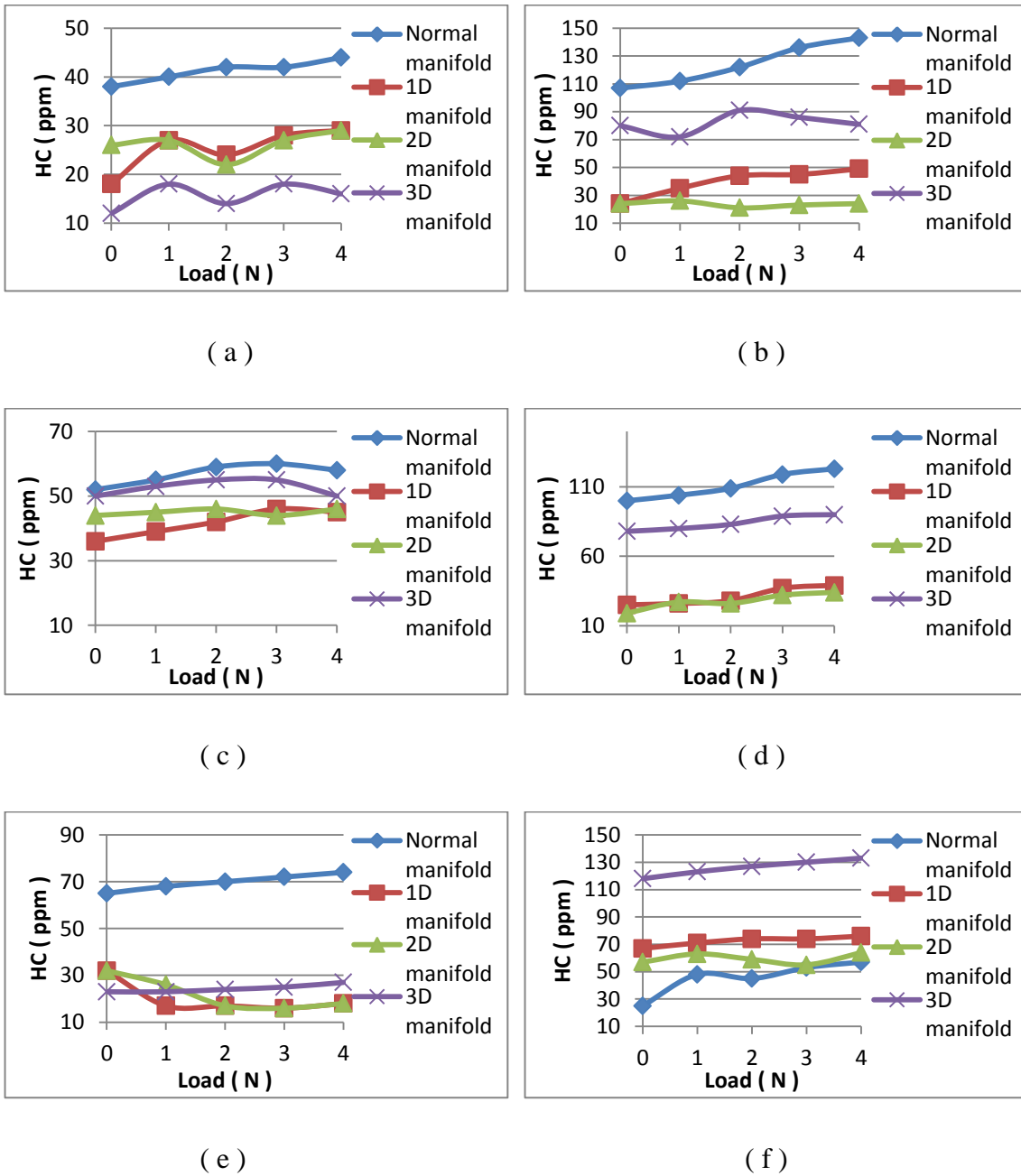


Figure 60. Hydrocarbons emission Vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Hydrocarbon emissions arise on the one hand from HC that are too lean to combust, and on the other from fuel that over penetrates and wets the cylinder wall during the ignition delay period. It can be observed in Figure 60 that the use of 1D and 2D manifolds has decreased CO emission in most of the cases. This can be related to the better air-fuel mixing generated as a result of the swirl motion which caused the combustion to be more complete. However, the use 3D manifold has given almost similar readings for HC emission or slightly less than the normal manifold. This conclusion proves that the use of the new induction manifolds is effective in obtaining more complete combustion and less HC emission rates. The use of 1D and 2D manifolds instead of normal manifold can decrease hydrocarbon emission by an average percentage of 52% and 61% respectively. However the use of 3D manifold can result in 20% increment in HC emission.

It can be clearly noticed that the use of diesel-waste cooking fuel blend in Figure (60 d) has resulted in the lowest HC readings. The reason behind that is the high distillation temperature of waste cooking oil (around 318 C) which results in less volatile fuel blend ,thus less amount of fuel droplets are quenched during the cold start period and HC emission is reduced.

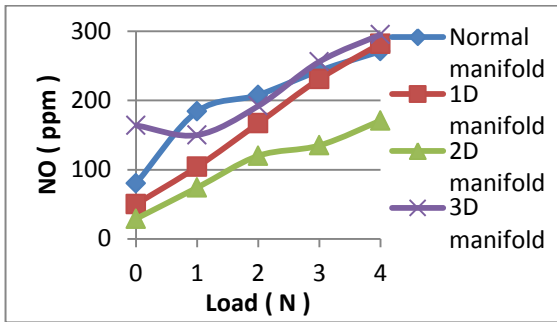
The very low viscosity of GTL fuel in comparison to other used fuels caused the HC emission to be higher in Figure (60 b) because HC results from fuel that over penetrates and wets the cylinder walls during the ignition delay period .As the fuel is less viscous; it is more able to penetrate the cylinder walls and thus results in more HC emission. However, the use of diesel-GTL fuel blend in Figure (60 c) has decreased HC emission partially. This

may be referred to the improved cetane number of the mixture where less amount of fuel penetrates the cylinder walls due to the shortened ignition delay period.

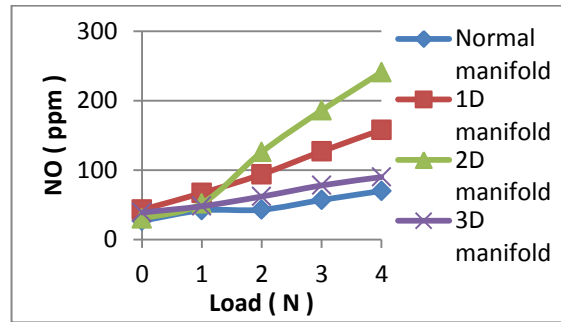
The addition of waste cooking oil to this blend in Figure (60 e) was found to be very effective in reducing HC emission due to the improved distillation temperature of the mixture, thus less amount of fuel is quenched during the engine early operation.

Nevertheless, the addition of corn oil to that blend has raised HC emission significantly in comparison to diesel fuel because of its very high viscosity leading to a more incomplete combustion and thus more HC emission. In addition to that, the corn oil has a very low distillation temperature (=160 C) which causes more evaporation of fuel during the cold start period and more fuel is quenched resulting in more HC emission.

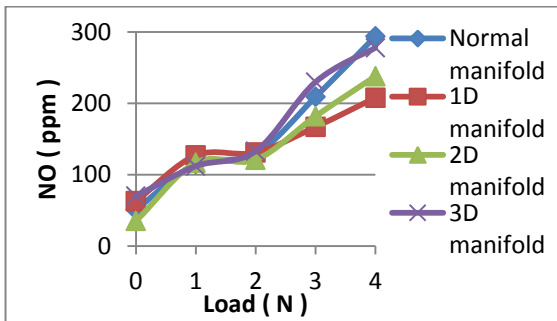
4) Nitric Oxide @constant speed



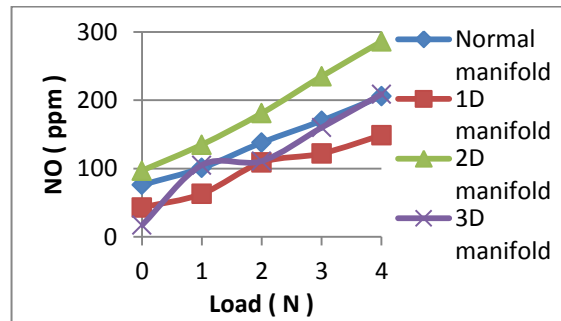
(a)



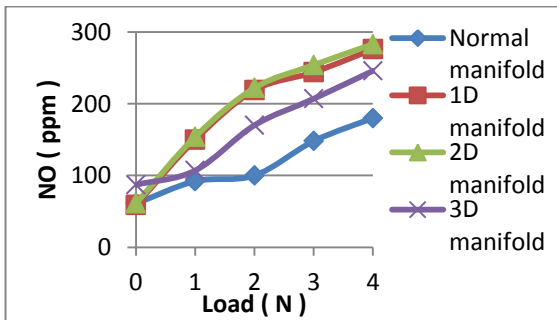
(b)



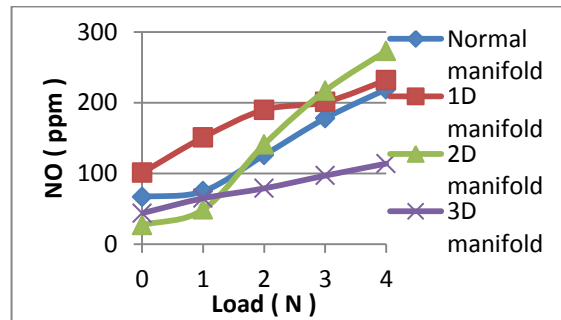
(c)



(d)



(e)



(f)

Figure 61. Nitric oxide emission vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

Figure 61 depicts the relation between the emission of Nitric oxide and engine load. NO_x results from reaction of nitrogen and oxides at relatively high temperature. NO is the major component in the NO_x emission that is why the focus is mainly about it. As the engine operates at higher loads, NO emission becomes higher. The reason behind that is that the flame temperature gets higher with more engine load which results in more NO emission.

It can be noticed from Figure 61 that the use of 1D and 2D manifolds has decreased NO emission in most of the cases. This may be related to the lower in-cylinder peak pressure and maximum pressure raise rate which is obtained when using those manifolds.

Consequently, both the ignition delay period and the flame temperature are less. However, the use of the 3D manifold has given almost similar or slightly less readings as it causes more in-cylinder peak pressure and thus higher flame temperatures. The use of 1D and 2D manifolds instead of normal manifold can decrease Nitric oxide emission by an average percentage of 32% and 35% respectively. However the use of the 3D manifold can result in 10% increment of NO emission.

The values of NO emission at different engine speeds were lowest when using GTL fuel as it has the highest cetane number (=70) which causes shorter ignition delay and less peak pressure as shown in Figure (61 b). Also, diesel and GTL fuel blend is considered to be a good solution for reducing NO emission in the cylinder when compared with the use of pure diesel as Figure (61 c) indicates. The cetane number was measured for this blend to be equals to 62.5 which is considered to be high with respect to other used fuel blends.

The waste cooking oil which has a very low cetane number (= 32.5) caused the NO emission to be higher when it was blended with diesel and GTL as shown in Figure (61 d)

and Figure (61 e) in which the cetane number was measured for those fuel blends to be 43.8 and 52.5 respectively. The addition of corn oil further to that blend has decreased NO emission slightly as the corn oil has a low cetane number (=37.6) as demonstrated in Figure (61 f).

5) Total Particulate Matters @constant speed

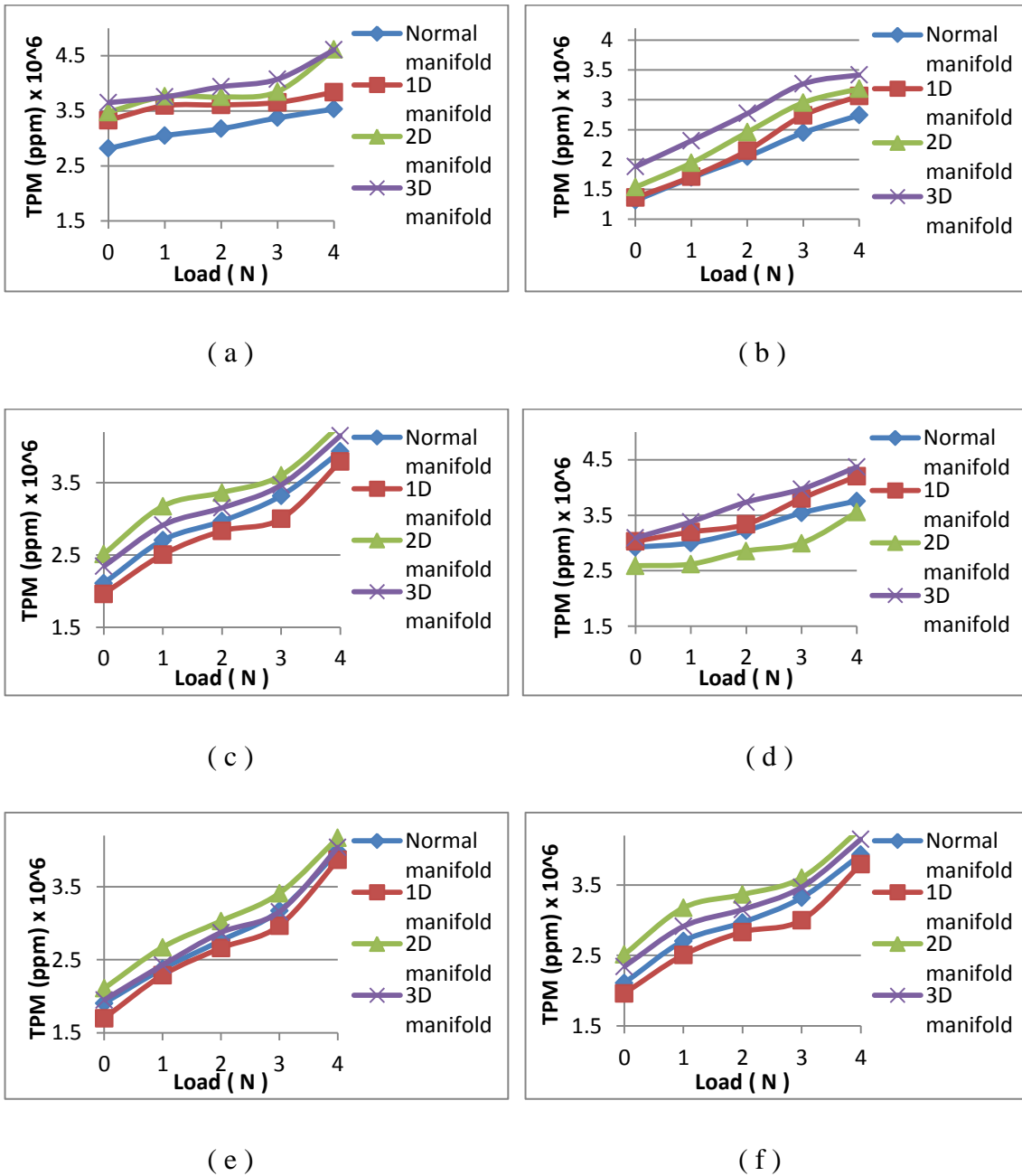
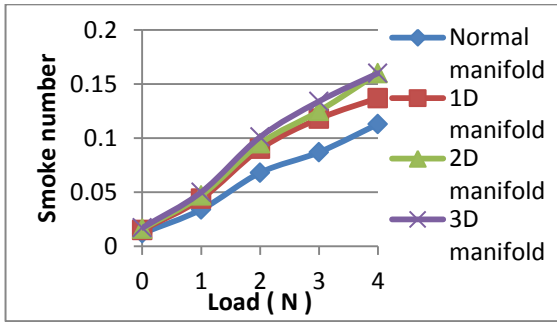


Figure 62. Particulate emission vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

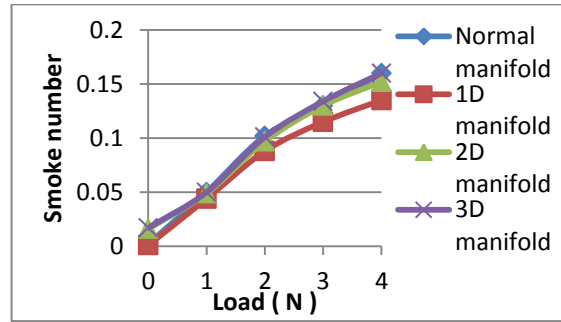
Figure 62 demonstrates the relation between particulate matters emission and engine load for six different types of fuels. PM is formed mainly in the fuel rich regions during the combustion period. Thus, as the load increases more the mixing charge becomes richer and a higher amount of PM is allowed to be emitted. Also, PM is produced from combustion that is incomplete. It can be observed from Figure 62 that the use of 1D manifold can decrease the PM emission more than the use of normal manifold. This can be referred to the swirl motion caused by the use of 1D manifold which enhances the air-fuel mixing quality and generates a high TKE. However, the use of 2D and 3D manifolds was found not to be effective due to increased volume of the intake manifold which causes the incoming air to be more pressurized and tends to demote the air-fuel mixing quality [62, 63]. The use of the 1D and 2D manifolds can decrease PM emission by an average percentage of 5% and 8% respectively. However, the use of 3D manifold can increase the PM emission by an average percentage of 20% at a constant engine speed operation.

Also, it can be observed that using pure GTL fuel as in Figure (62 b) or mixing it with diesel in one fuel blend as in Figure (62 c) can highly decrease the PM emission as it has a very low –sulfur and aromatics content in comparison to diesel fuel. Also, the addition of biofuels like the corn oil or using waste cooking oil in fuel blends with the conventional fuels (diesel and GTL) is considered to be a good solution for decreasing the PM emission as those alternative fuel blends contains less amount of sulfur, thus less amount of PM are produced in comparison to diesel fuel as can be observed in Figure (62 d) through Figure (62 f).

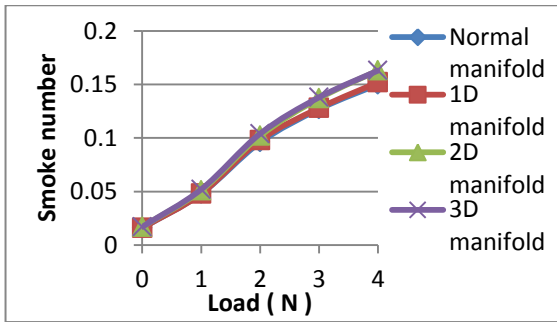
6) Smoke Number @constant speed



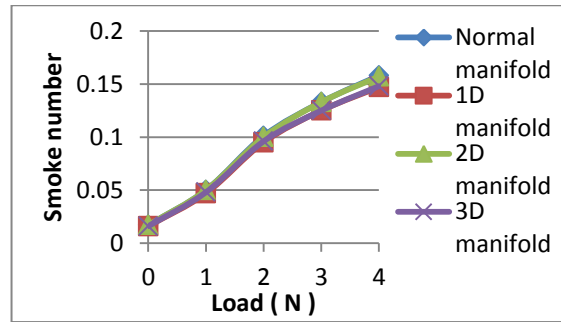
(a)



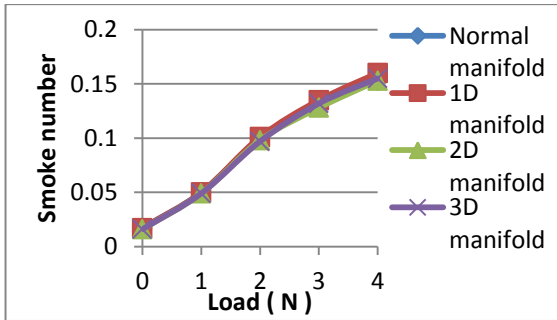
(b)



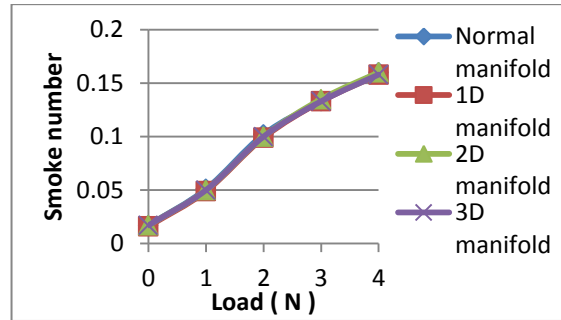
(c)



(d)



(e)



(f)

Figure 63. Smoke emission Vs. engine load for :(a) diesel fuel ,(b)GTL fuel ,(c)diesel-GTL blend,(d)diesel-waste cooking oil blend (e) diesel-GTL-waste cooking oil blend (f) diesel-GTL-waste cooking oil-corn oil blend

The black smoke is produced from the fuel carbon particles that are released by the thermal cracking of the large hydrocarbon fuel molecules. Higher loads imply higher fuel consumptions and that is why the smoke number in Figure 63 is significantly increasing with load. As observed in Figure 63, the use of 1D manifold has recorded slightly lower readings for the smoke emission in comparison with the normal manifold. This can be referred to the increase in volumetric efficiency in case of using the 1D manifold, thus the mixing charge becomes leaner and less amount of smoke can be detected in the exhaust. However, the use of 2D and 3D manifolds was not effective in decreasing smoke production rates as those manifolds can result in lower volumetric efficiencies than the normal manifold. The use of the 1D manifold can decrease the smoke concentration by an average percentage of 3%. However the use of 2D and 3D manifold can result in 1% increment in smoke concentration.

The use of waste cooking oil and corn oil in one fuel blend with conventional fuels (diesel and GTL) as in Figure (63 d) through Figure (63 f) has decreased the smoke production slightly. The reason behind can be referred to their high oxygen content which plays a major role in reducing soot formation and in soot oxidation [68]. Vice versa, the use of conventional fuels has given almost similar smoke readings as can be shown in Figure (63 a) through Figure (63 c).

4.2 Effect of Using Fuel Blends

This section compares between diesel fuel and another five different fuels .Of those fuels, GTL fuel is the only one used in the pure form and the remaining are all fuel blends. The test will involve combustion, performance and emission characteristics using the 1D manifold in the constant load case and the normal manifold in the constant speed case .The selection of those manifolds is based on their optimum results in the previous section.

4.2.1 Combustion Characteristics

1) P- θ Diagram

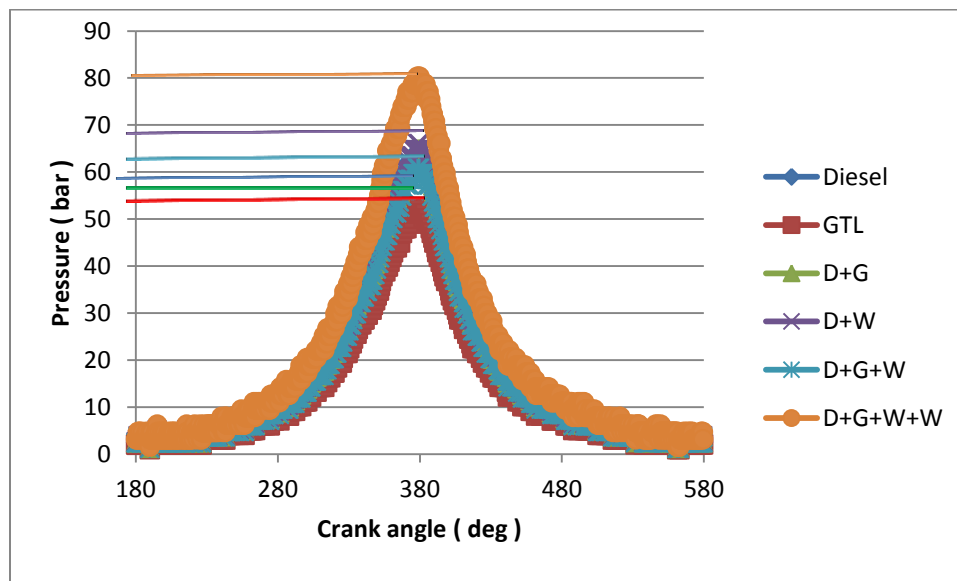


Figure 64. Pressure vs. crank angle position for six different types of fuels using the 1D manifold at a speed=1700rpm and a load =1N.m

The plot of pressure vs. crank angle position is drawn for the range of crank rotation from top dead center (360°) to a period after combustion (aTDC) during the expansion stroke (390°). While studying the plot, the main emphasis was on the period between the start of injection (SOI) and the start of combustion (SOC) and then the peak of the curve achieved at SOI and SOC in which both are expressed in terms of crank angle position before the top dead center (bTDC) since the top dead center (TDC) is the moment at which the piston reaches the highest level in the cylinder at the end of compression.

Figure 64 compares the pressure rise for six different types of fuels at an engine speed of 1700 rpm and an engine load of 1N.m using the 1D manifold. It is clearly noticed that the use of GTL fuel has resulted in the least pressure accumulation in the cylinder during the compression stroke and combustion period. The reason behind that can be referred to the high cetane number of GTL in comparison with other used fuel blends, hence a shorter ignition delay period was obtained and the pressure rise was the least. The use of GTL has resulted in 19% reduction in pressure rise values at different crank angle positions.

Diesel and GTL fuel blend can also be considered a good solution for reducing the pressure rise in the cylinder and minimizing the ignition delay period in comparison to pure diesel. The cetane number was measured for this blend to be equal to 62.5, however the cetane number for pure diesel is 55 and this explains the difference between both fuels in pressure rise readings. The use of diese-GTL blend has minimized the pressure rise by an average percentage of 4% at the specified engine operation conditions.

The waste cooking oil which has a very low cetane number (= 32.5) caused the pressure rise in the cylinder to be higher when it was blended with diesel alone or with diesel and

GTL and an observable increment in pressure rise was detected for both fuel blends. The use of those fuel blends has resulted in about 7% in-cylinder pressure rise in comparison to diesel fuel.

For the same reason, the addition of corn oil further to diesel-GTL-waste cooking oil blend has resulted in about 17% in-cylinder pressure rise. The cetane number was measured for the aforementioned fuel blend to be equal to 48.8 which is less than the cetane number of pure diesel (=55).

Constant Load

2) The maximum pressure raise rate (dP-dθ)

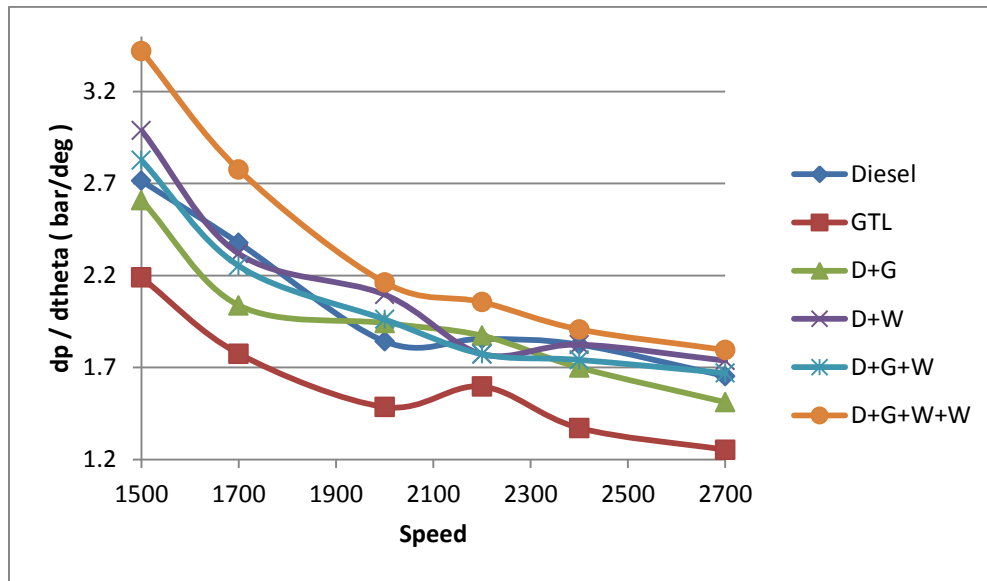


Figure 65. Dp/dθ vs. engine speed for different types of fuel using the 1D manifold

Figure 65 indicates the maximum pressure raise rate with different engine speeds and different fuel blends. It is clearly noticed that the use of GTL fuel has resulted in the least maximum pressure raise rate. This gives an indication that the use of GTL results in a shorter ignition delay period and thus fewer amounts of fuel droplets are burned which results in a less maximum pressure raise rate in comparison with diesel fuel. The reason behind that can be referred to the higher cetane number of GTL in comparison with diesel fuel and the other used fuel blends as this fuel property plays a vital role in enhancing engine combustion characteristics. The use of GTL fuel instead of diesel fuel can result in 19% reduction in maximum pressure raise rate values at different engine speeds.

Diesel and GTL fuel blend has also minimized the maximum pressure rise rate by an average percentage of 5% due to its higher cetane number. The cetane number was measured for this fuel blend to be equal to 62.5, however the cetane number for pure diesel is 55 and this explains the difference between both fuels in maximum pressure rise rate readings.

The waste cooking oil which has a very low cetane number (= 32.5) caused the maximum pressure rise rate in the cylinder to be slightly higher when it was blended with diesel alone or with diesel and GTL. The use of those fuel blends has resulted in about 2% in-cylinder maximum pressure rise rate in comparison with diesel fuel. For the same reason, the addition of corn oil further to diesel-GTL-waste cooking oil blend has resulted in about 14% in-cylinder maximum pressure rise rate. The cetane number was measured for the aforementioned fuel blend to be equal to 48.8 which is less than the cetane number of pure diesel (=55).

3) Peak Pressure

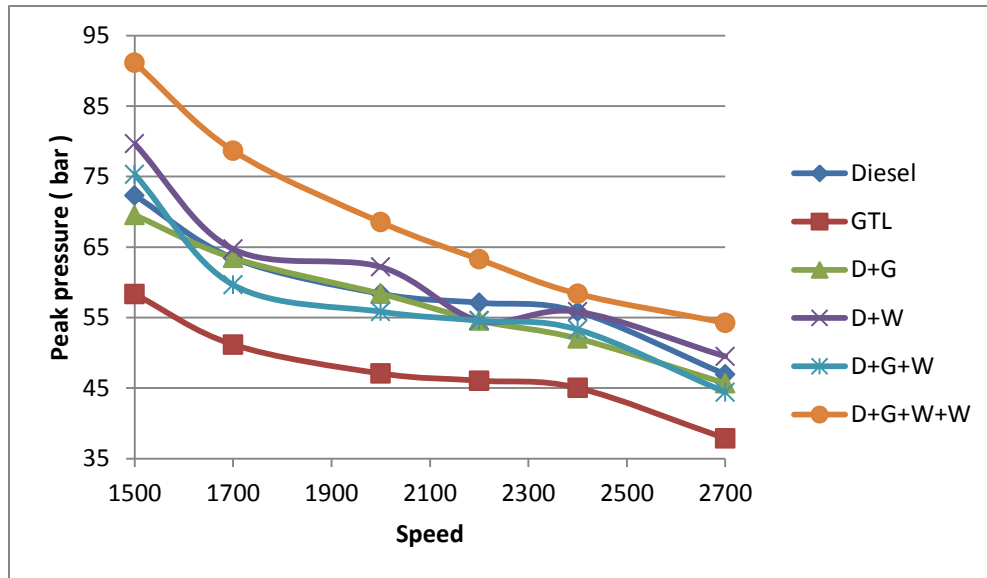


Figure 66. Peak pressure vs. engine speed for different types of fuel using the 1D manifold

In a compression ignition engine, the combustion characteristics of the fuel and the engine performance and emission depend upon the ignition delay period. The longer the delay period, the higher is the rate of combustion and the higher is the resulting pressure rise. The rise in pressure inside the cylinder is attributed to the rate of combustion as well as the ignition delay period of the fuel, which in turn depends on the cetane index. Moreover, the fuel viscosity and its calorific value can also play a major role in determining its in-cylinder peak pressure.

As it can be noticed in Figure 66, the lowest peak pressure readings were obtained in case of using GTL fuel. This can be referred to its high cetane number in comparison with diesel fuel and the other used fuel blends. The use of GTL fuel instead of diesel fuel can result in

18% in-cylinder peak pressure at various engine speeds. The use of GTL fuel with diesel fuel in one fuel blend has resulted in a slight reduction in the in-cylinder peak pressure readings of about 4%. The reason behind that can be related to the higher cetane number of diesel-GTL fuel blend compared to pure diesel fuel which has decreased the ignition delay period resulting in a reduction of the accumulated fuel in the cylinder prior to combustion period.

The waste cooking oil which has a very low cetane number (= 32.5) caused the peak pressure in the cylinder to be higher when it was blended with diesel and GTL. The use of those fuel blends has resulted in about 10% increment of in-cylinder peak pressure in comparison with diesel fuel. In the same principle, the addition of corn oil further to the conventional fuels and the waste cooking oil has resulted in about 17% increment of in-cylinder peak pressure. The cetane number was measured for the aforementioned fuel blend to be equal to 48.8 which is less than the cetane number of pure diesel (=55).

Also, the high calorific value for GTL fuel (=47.3 MJ/kg) has resulted in better engine combustion characteristics in terms of maximum in-cylinder pressure and in-cylinder pressure rise rate in comparison to other used fuels which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for the corn oil equals to 42.7 MJ/kg.

Furthermore, GTL fuel has the lowest kinematic viscosity ($=2 \text{ mm}^2/\text{s}$) in comparison to other used fuel blends. The kinematic viscosity was measured for other fuels at STP and was found to be equals to $5.2 \text{ mm}^2/\text{s}$ for diesel fuel, $4.9 \text{ mm}^2/\text{s}$ for waste cooking oil and $45 \text{ mm}^2/\text{s}$ for corn oil. The very high viscosity of corn oil explains why the maximum

cylinder pressure readings were the highest .As the fuel is less viscous, the fuel droplets become smaller which enhances the air-fuel mixing and thus improves the engine’s combustion characteristics.

Constant Speed

2) dP-dθ Diagram

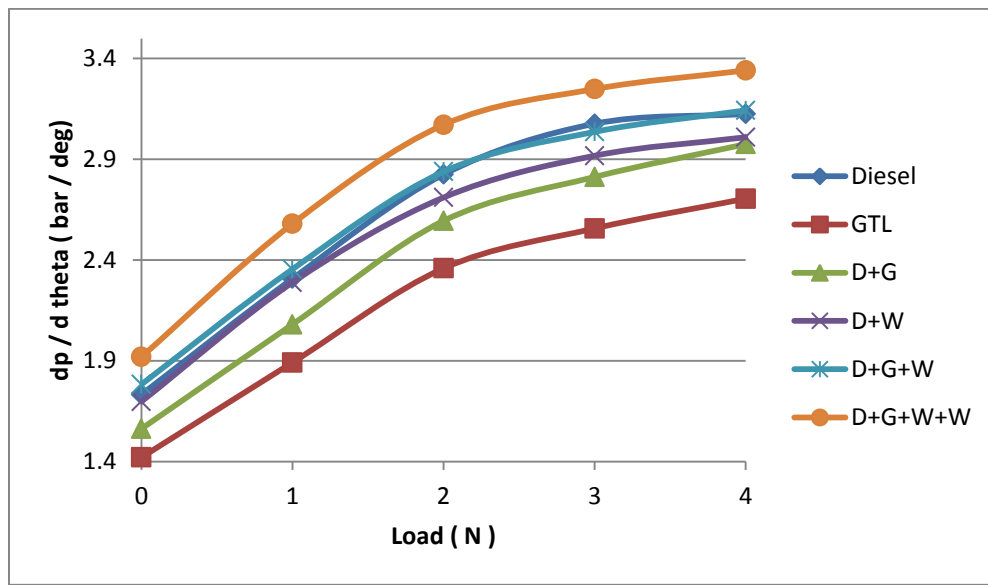


Figure 67. Dp/dθ Vs. engine load for different types of fuel using the normal manifold

Figure 67 indicates the maximum pressure raise rate at different engine loads and for different fuel blends. It is clearly noticed that the use of GTL fuel has resulted in the least maximum pressure raise rate. This gives an indication that the use of GTL results in a shorter ignition delay period and thus fewer amounts of fuel droplets are burned which results in a less maximum pressure raise rate in comparison with diesel fuel. The reason

behind that can be referred to the higher cetane number of GTL in comparison with diesel fuel and the other used fuel blends as this fuel property plays a vital role in enhancing engine combustion characteristics. The use of GTL fuel instead of diesel fuel can result in 18% reduction in maximum pressure rise rate values at different engine speeds.

The use of diesel - GTL fuel blend has also minimized the maximum pressure rise rate by an average percentage of 7% due to its higher cetane number. The cetane number was measured for this blend to be equal to 62.5, however the cetane number for pure diesel is 55 and this explains the difference between both fuels in maximum pressure rise rate readings. The waste cooking oil which has a very low cetane number (= 32.5) caused the maximum pressure rise rate in the cylinder to be slightly lower when it was blended with diesel alone or with diesel and GTL. Although the cetane number of those fuel blends is less than diesel fuel (43.8 and 52.5 respectively), however their lower viscosity played a major role in obtaining lower maximum pressure rise rate readings of about 4% and 2% respectively due to the enhanced combustion quality.

The addition of corn oil further to the conventional fuels and the waste cooking oil has resulted in about 9% increment of in-cylinder maximum pressure rise rate. The cetane number was measured for the aforementioned fuel blend to be equal to 48.8 which is less than the cetane number of pure diesel (=55) and this explains why the readings were higher for that case. In addition to that, the corn oil has a very high viscosity which results in a lower combustion quality.

3) Peak Pressure

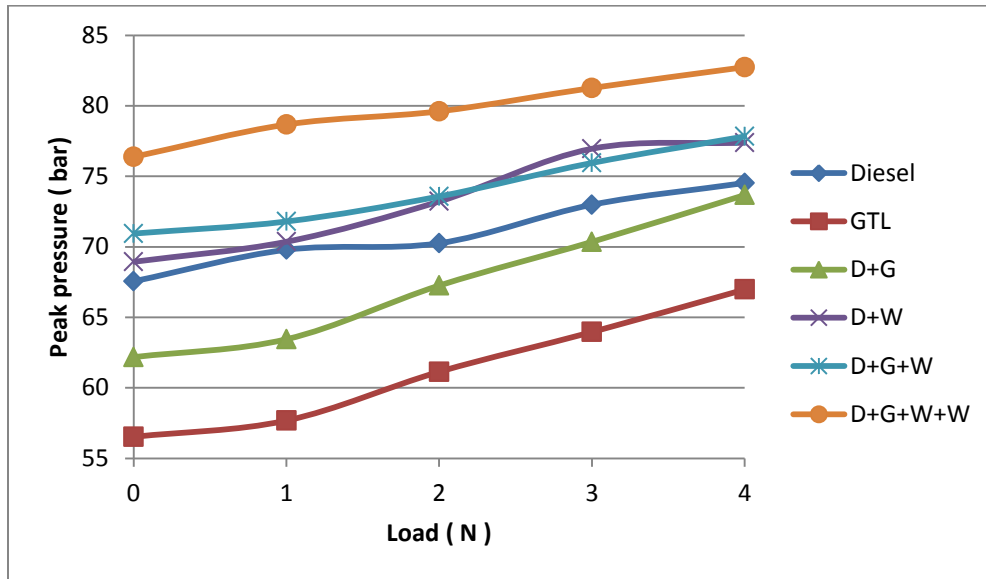


Figure 68. Peak pressure Vs. engine load for different types of fuel using the normal manifold

The rise in pressure inside the cylinder is attributed to the rate of combustion as well as the ignition delay period of the fuel, which in turn depends on the cetane index. The higher the value of the cetane index of the fuel, the shorter is the ignition delay period.

Figure 68 demonstrates the variation of the in-cylinder peak pressure with load for six different types of fuel. As it can be noticed in Figure 68, the lowest peak pressure readings were obtained in case of using GTL fuel. This can be referred to its high cetane number in comparison with diesel fuel and the other used fuel blends. The use of GTL fuel has result in about 11% in-cylinder peak pressure reduction at different engine loads.

The use of GTL fuel with diesel fuel in one fuel blend has resulted in a slight reduction in the in-cylinder peak pressure readings of about 5%. The reason behind that can be related to the higher cetane number of diesel-GTL fuel blend compared to pure diesel fuel which has decreased the ignition delay period resulting in a reduction of the accumulated fuel in the cylinder prior to combustion period.

The waste cooking oil which has a very low cetane number (= 32.5) caused the peak pressure in the cylinder to be higher when it was blended with diesel or with diesel and GTL. The use of those fuel blends has resulted in about 4% increment of in-cylinder peak pressure in comparison with diesel fuel. For the same reason, the addition of corn oil further to diesel-GTL-waste cooking oil fuel blend has resulted in about 13% increment of in-cylinder peak pressure. The cetane number was measured for the aforementioned fuel blend to be equal to 48.8 which is less than the cetane number of pure diesel (=55).

Also, the high calorific value for GTL fuel (=47.3 MJ/kg) has resulted in better engine combustion characteristics in terms of in-cylinder peak pressure and the maximum in-cylinder pressure rise rate in comparison to other used fuels which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equals to 42.7 MJ/kg.

Furthermore, GTL fuel has the lowest kinematic viscosity ($=2 \text{ mm}^2/\text{s}$) in comparison to other used fuel blends. The kinematic viscosity was measured for other fuels at STP and was found to be equals to $5.2 \text{ mm}^2/\text{s}$ for diesel fuel, $4.9 \text{ mm}^2/\text{s}$ for waste cooking oil and $45 \text{ mm}^2/\text{s}$ for corn oil. The very high viscosity of corn oil explains why the maximum cylinder pressure readings were the highest. As the fuel is less viscous, the fuel droplets

become smaller which enhances the air-fuel mixing and thus improves the engine's combustion characteristics.

4.2.2 Engine Performance

Constant Load

1) Brake Specific Fuel Consumption

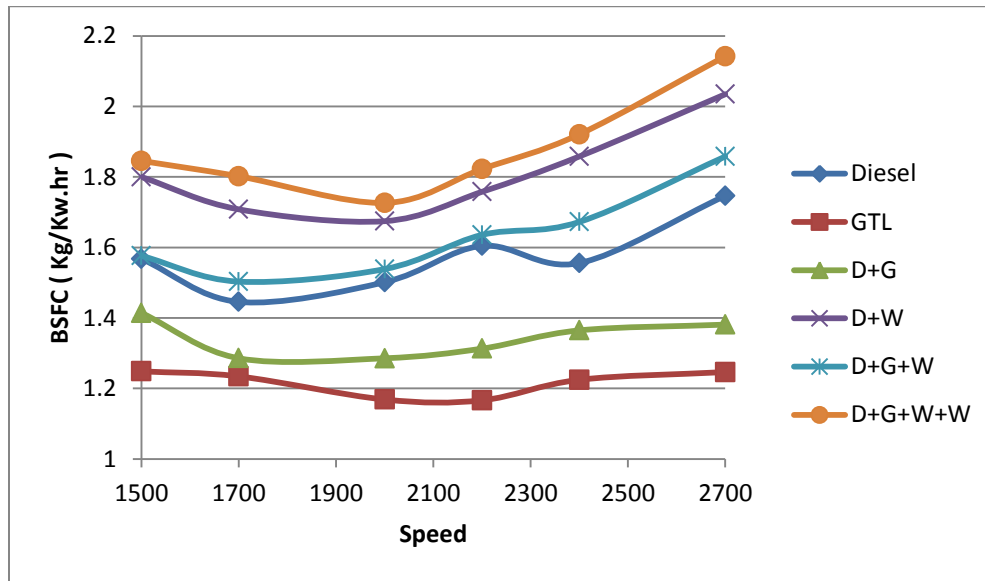


Figure 69. Brake specific fuel consumption vs. engine speed for different types of fuel using the 1D manifold

The bsfc is a tool used for measuring engine efficiency and fuel economy. Figure 69 shows the relation between the bsfc with respect to engine speed for six different types of fuels.

The high calorific value of GTL fuel (=47.3 MJ/kg) has resulted in less fuel consumption in comparison to other used fuels which have lower calorific values. The calorific value for

diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equals to 42.7 MJ/kg. As the fuel calorific value becomes higher, less amount of fuel has to be burned in order to produce the same amount of combustion energy. The use of GTL fuel has resulted in 20 % decrement in bsfc in comparison to diesel fuel.

Furthermore, the high cetane number of GTL has caused less amount of fuel to be burned during the combustion phase and this has also been considered one of the reasons to the enhanced bsfc with the use of GTL. The use of GTL fuel with diesel fuel in one fuel blend was also effective in decreasing bsfc by about 10% due to the enhancement of the fuel blend's calorific value. The calorific value was measured for this fuel blend and was found to be equal to 45.8 MJ/kg.

However, the use of diesel-waste cooking oil fuel blend has resulted in 15% increment in bsfc due to the lower heating value of the fuel blend in comparison with pure diesel fuel. The calorific value was measured for this fuel blend and was found to be equal to 43.5 MJ/kg. The addition of GTL for that fuel blend has resulted in only 4% increment in bsfc, however the addition of corn oil further for that mixture has resulted in about 17% increment in comparison with diesel fuel due to its low calorific value. The calorific value was measured for this fuel blend and was found to be equal to 42.7 MJ/kg. The low cetane number for waste cooking oil and corn oil has also been considered a reason for increasing bsfc values as this caused more fuel to be burned during the combustion period and thus more fuel was consumed.

2) Volumetric Efficiency

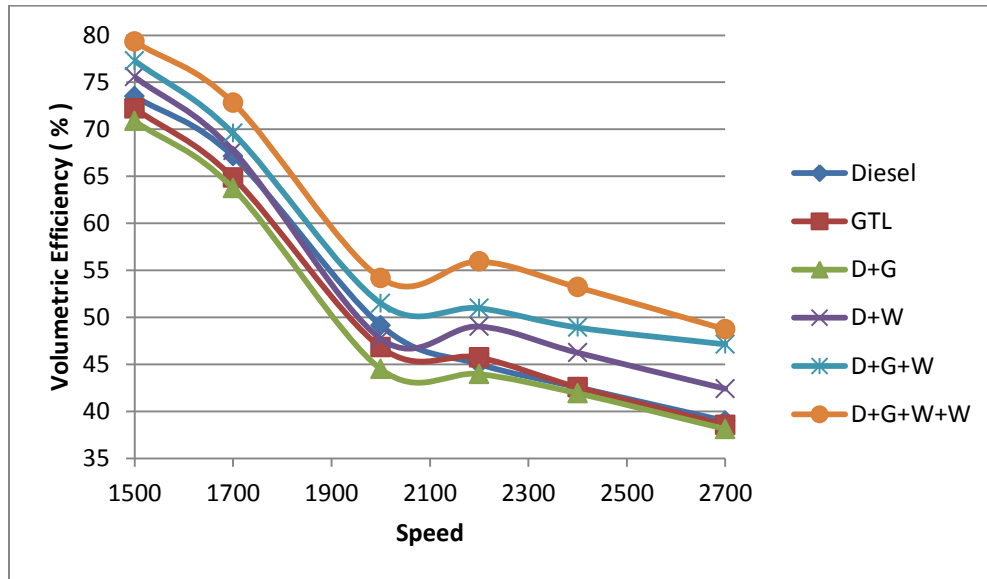


Figure 70. Volumetric efficiency Vs. engine speed for different types of fuel using the 1D manifold

Figure 70 depicts the relation between the volumetric efficiency and engine speed for six different types of fuels. Volumetric efficiency is a measure of how good the engine is at receiving in air, and anything that reduces the flow of air into the engine will lower the volumetric efficiency. The two biggest reasons for poor volumetric efficiency at high engine speeds are frictional flow losses and choked flow.

As it can be noticed in Figure 70, the volumetric efficiency readings in the fuel blends where the waste cooking oil and corn oil have been used are the highest. This can be referred to the high oxygen content inherited in the waste cooking oil and corn oil in comparison to conventional fuels diesel and GTL. Consequently, less amount of air needed to initiate the combustion and the volumetric efficiency reading get higher. The addition of

waste cooking oil to diesel in a 50 to 50 percent volume composition has resulted in 2% increment in volumetric efficiency. However, the addition of corn oil further for that blend has resulted in 8% volumetric efficiency's increment.

The use of pure GTL or the addition of GTL to diesel had almost no effect in the volumetric efficiency readings because those conventional fuels are poor of oxygen content. Nevertheless, the addition of waste cooking oil to that blend in 33.3 % volume composition has resulted in 5% increment in volumetric efficiency.

3) Exhaust Temperature

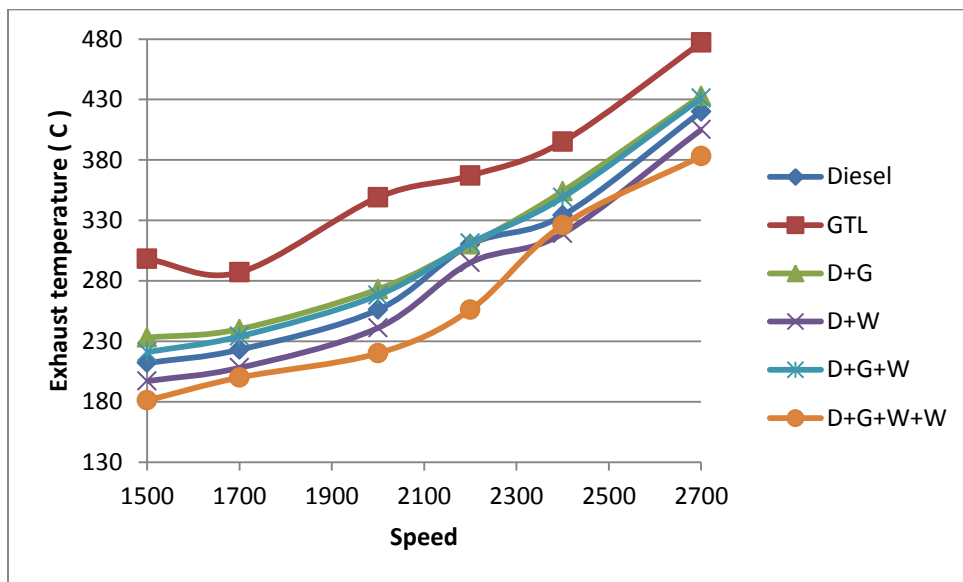


Figure 71. Exhaust temperature vs. engine speed for different types of fuel using the 1D manifold

Figure 71 depicts the variation of exhaust gas temperature readings with speed for six different types of fuels. Exhaust gas temperature indicates that the heat in the cylinder is converted into work and the combustion is more complete. At various speed conditions, it is observed that the exhaust gas temperature increases with speed because the mixing charge becomes richer with speed which causes more fuel to be burned.

GTL fuel has the lowest kinematic viscosity ($=2 \text{ mm}^2/\text{s}$) in comparison to other used fuel blends and this has raised the air-fuel mixing quality and resulted in a more complete combustion and thus higher exhaust temperature readings. The use of GTL has caused 37% increment in exhaust temperature in comparison to diesel fuel. The use of GTL fuel with diesel fuel in one fuel blend has resulted in a slight increment in the in-exhaust temperature readings of about 10% due to the enhancement of the fuel viscosity. The kinematic viscosity was measured for this fuel blend at STP and was found to be equal to $3.6 \text{ mm}^2/\text{s}$.

The kinematic viscosity was measured for the other used fuels at STP and was found to be equals to $5.2 \text{ mm}^2/\text{s}$ for diesel fuel, $4.9 \text{ mm}^2/\text{s}$ for waste cooking oil and $45 \text{ mm}^2/\text{s}$ for corn oil. The addition of waste cooking oil to diesel has resulted in about 7% reduction in the exhaust temperature readings due to its lower viscosity and its high oxygen content which caused the mixing charge to be leaner, thus less fuel is burned and the exhaust temperature is reduced. However, the use of GTL with diesel and waste cooking oil fuel blend has raised the exhaust temperature readings in about 4% due to the enhancement of the fuel viscosity. The kinematic viscosity was measured for this fuel blend at STP and was found to be equal to $4 \text{ mm}^2/\text{s}$.

The very high viscosity of corn oil explains why the exhaust temperature readings were the lowest as this has caused the mixing quality to be lower and the combustion is less complete. The use of this fuel blend has caused 15 % decrement in exhaust temperature with in comparison to diesel fuel. Also, the high calorific value of GTL fuel (=47.3 MJ/kg) has resulted in better engine combustion and higher exhaust temperature readings in comparison to other used fuels which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equal to 42.7 MJ/kg.

Constant Speed

1) Brake Specific Fuel Consumption

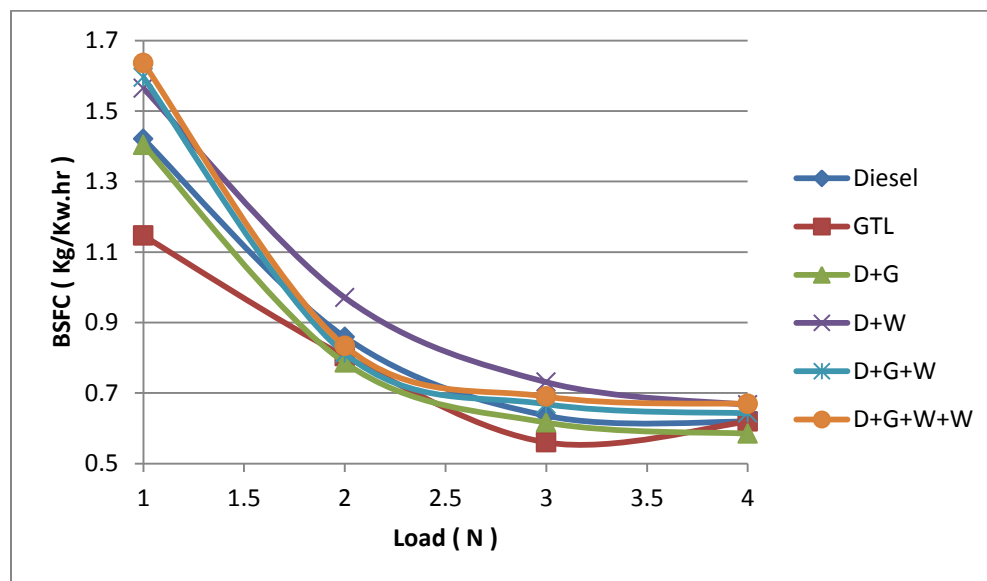


Figure 72. Brake specific fuel consumption vs. engine load for different types of fuel using the normal manifold

The bsfc is used as an indication for measuring engine efficiency and fuel economy. The least is the bsfc, the higher is the thermal efficiency and the more the fuel is economical. Figure 72 shows the relation between the bsfc with respect to engine load for different types of fuels. As it can be clearly noticed, the bsfc readings for GTL are the lowest. The high calorific value for GTL fuel (=47.3 MJ/kg) has resulted in less fuel consumption in comparison to other used fuels which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equals to 42.7 MJ/kg. As the fuel calorific value becomes higher, less amount of fuel has to be burned in order to produce the same amount of combustion energy. The use of GTL has resulted in 12 % decrement in bsfc in comparison to diesel fuel.

Furthermore, the high cetane number of GTL caused less amount of fuel to be burned during the combustion phase and this has also been considered one of the reasons to the enhanced bsfc with the use of GTL. The use of GTL fuel with diesel fuel in one fuel blend was also effective in decreasing bsfc by about 7% due to the enhancement of the fuel blend's calorific value. The calorific value was measured for this fuel blend and was found to be equal to 45.8 MJ/kg.

However, the use of diesel-waste cooking oil fuel blend has resulted in 13% increment in bsfc due to the lower heating value of the fuel blend in comparison with pure diesel fuel. The calorific value was measured for this fuel blend and was found to be equal to 43.5 MJ/kg. The addition of GTL for that fuel blend has resulted in 12% increment in bsfc, however the addition of corn oil further for that mixture has resulted in about 15% increment in comparison with diesel fuel due to its low calorific value. The calorific value

was measured for this fuel blend and was found to be equal to 42.7 MJ/kg. The low cetane number for waste cooking oil and corn oil has also been considered a reason for decreasing bsfc values as this caused more fuel to be burned during the combustion period and thus more fuel was consumed.

2) Volumetric Efficiency

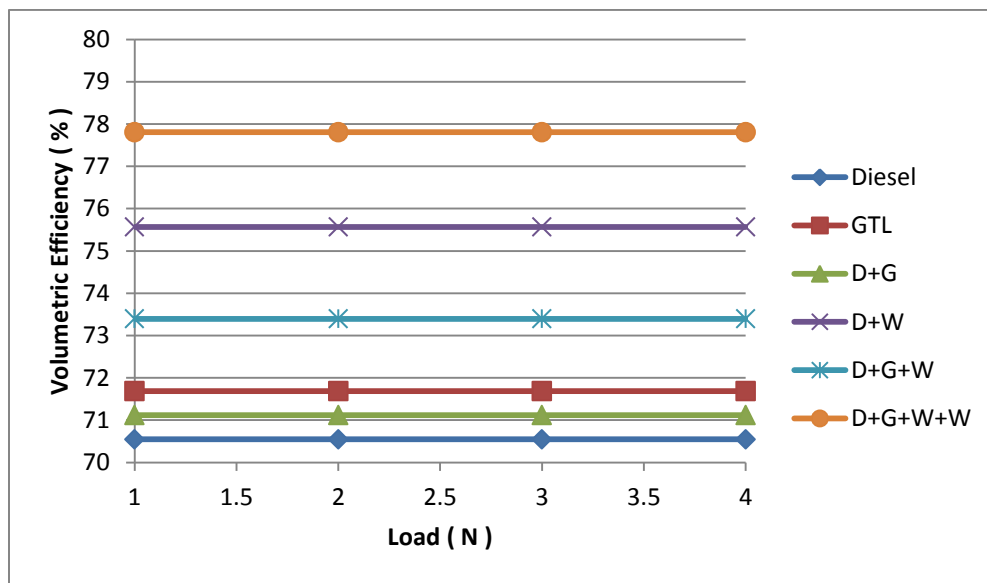


Figure 73. Volumetric efficiency vs. engine load for different types of fuel using the normal manifold

Figure 73 depicts the relation between the volumetric efficiency and engine load for six different types of fuel. Volumetric efficiency is a measure of how good the engine is at receiving in air, and anything that reduces the flow of air into the engine will lower the

volumetric efficiency. The two biggest reasons for poor volumetric efficiency are frictional flow losses and choked flow.

As it can be noticed in Figure 73, the volumetric efficiency readings in the fuel blends where the waste cooking oil and corn oil have been used are the highest. This can be referred to the high oxygen content inherited in the waste cooking oil and corn oil in comparison to conventional fuels diesel and GTL. Consequently, less amount of air needed to initiate the combustion and the volumetric efficiency reading get higher. The addition of waste cooking oil to diesel in a 50 to 50 percent volume composition has resulted in 7% increment in volumetric efficiency.

The use of pure diesel or diesel-GTL fuel blend has only increased the volumetric efficiency marginally because those conventional fuels are poor of oxygen content. The use of pure GTL has increased the volumetric efficiency by only 2% and the use of diesel-GTL fuel blend has resulted in only 1% increment in the volumetric efficiency. However, the addition of waste cooking oil further for diesel-GTL fuel blend has resulted in 5% volumetric efficiency's increment. Also, the addition of corn oil further for that mixture has enhanced the volumetric efficiency the most and was found to cause 10% increment in the readings in comparison to pure diesel.

3) Exhaust Temperature

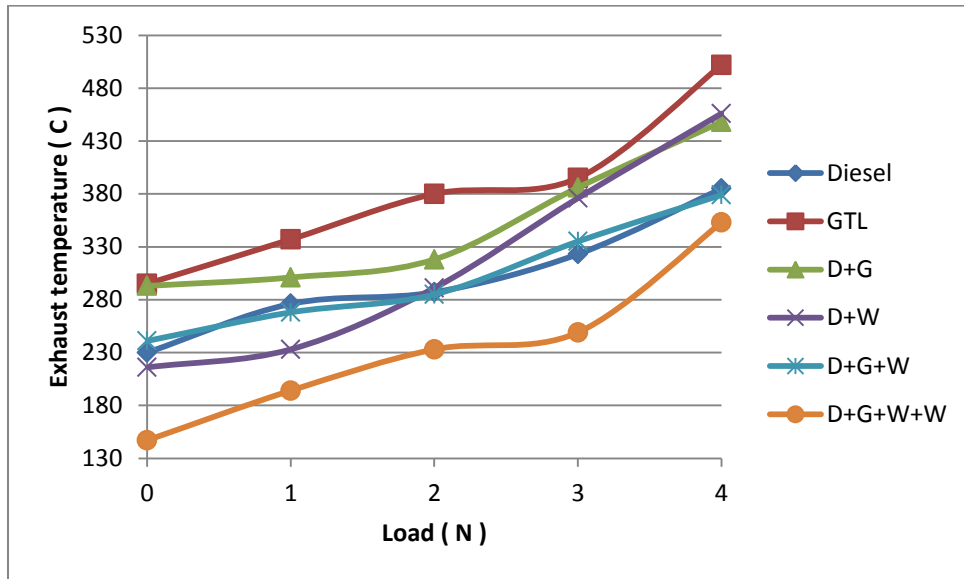


Figure 74. Exhaust temperature vs. engine load for different types of fuel using the normal manifold

Figure 74 depicts the variation of exhaust gas temperature with load for six different types of fuels. A higher exhaust temperature reading indicates that the combustion is more complete. At various load conditions, it is observed that the exhaust gas temperature increases with load because the mixing charge becomes richer as the load increases which causes more fuel to be burned.

GTL fuel has the lowest kinematic viscosity ($=2 \text{ mm}^2/\text{s}$) in comparison to other used fuel blends and this has raised the air-fuel mixing quality and resulted in a more complete combustion and higher exhaust temperature readings. The use of GTL fuel has caused 26% increment in exhaust temperature with respect to diesel fuel. The use of GTL fuel with

diesel fuel in one fuel blend has resulted in a slight increment in the in-exhaust temperature readings of about 16% due to the enhancement of the fuel viscosity. The kinematic viscosity was measured for this fuel blend at STP and was found to be equal to $3.6 \text{ mm}^2/\text{s}$.

The kinematic viscosity was measured for other fuels at STP and was found to be equals to $5.2 \text{ mm}^2/\text{s}$ for diesel fuel, $4.9 \text{ mm}^2/\text{s}$ for waste cooking oil and $45 \text{ mm}^2/\text{s}$ for corn oil. The addition of waste cooking oil to diesel was only effective at high engine loads (above 2N.m) and was inefficient beyond that load in reducing CO emission. The reason behind that is that the air fuel ratio becomes lower at high engine loads, however the existence of waste cooking oil in the blend can compensate the reduction of air that happens at high engine loads and thus the combustion is more complete. The use of diesel- waste cooking oil can increase the exhaust temperature by an average percentage of 16% at high engine loads.

The use of GTL with diesel -waste cooking oil fuel blend has raised the exhaust temperature readings in about 5% due to the enhancement of the fuel viscosity. The kinematic viscosity was measured for this fuel blend at STP and was found to be equal to $4 \text{ mm}^2/\text{s}$. The very high viscosity of corn oil explains why the exhaust temperature readings were the lowest as this has caused the mixing quality to be lower and the combustion is less complete. The use of this fuel blend has caused 36 % decrement in exhaust temperature in comparison to diesel fuel.

Also, the high calorific value for GTL fuel ($=47.3 \text{ MJ/kg}$) has resulted in better engine combustion and higher exhaust temperature readings in comparison to other used fuels

which have lower calorific values. The calorific value for diesel fuel equals to 44.3 MJ/kg, for waste cooking oil equals to 36.3 MJ/kg and for corn oil equal to 42.7 MJ/kg.

4.2.3 Engine Emissions

Constant Load

1) Carbon Monoxide

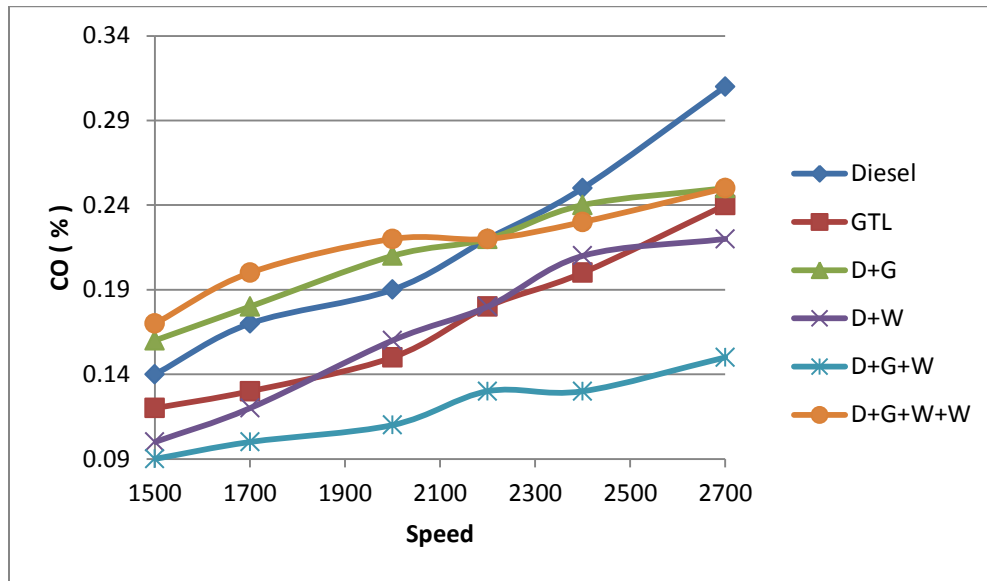


Figure 75. Carbon monoxide emission vs. engine speed for different types of fuel using the 1D manifold

Carbon monoxide occurs only in the engine exhaust. It is resulted as the product of incomplete combustion. Higher CO emission rates are expected at higher engine speeds as the air-fuel mixture becomes richer and the air to fuel ratio decreases more [65]. Moreover,

at higher engine speeds there will be less time available for air-fuel mixing and an increased quantity of injected fuel which further increases CO emission.

It can be clearly noticed in Figure 75 that the CO emission rates have been decreased with the use of GTL fuel. The reason behind that is referred to its high cetane number (=70) which has resulted in better combustion characteristics and a more complete combustion. Also, GTL low viscosity played a major role in enhancing the air-fuel mixing which has resulted in a higher quality of air-fuel mixing and thus less CO emission. The use of GTL has resulted in 14% CO emission reduction. The use of diesel with GTL was found to be effective in decreasing CO emission only at high engine speeds (above 2200 rpm) which indicates that the combustion quality is improved at high engine speeds. The use of this fuel blend has resulted in an average percentage of 12% at high engine speeds.

The addition of waste cooking oil to this fuel blend was found to be very effective in decreasing CO emission at all speed ranges. The reason behind that can be referred to the high oxygen content of waste cooking oil and the high cetane number of GTL in which both properties have resulted in better air-fuel mixing quality. The use of this fuel blend has decreased CO emission rates by an average percentage of 42%. Also, the addition of waste cooking oil to pure diesel in a 50 to 50 volume percentage has recorded a remarkable reduction in CO of about 29%.

Although the corn oil has high oxygen content, however its very high viscosity has affected the mixing quality more and a poor air-fuel mixing was obtained. In fact, the addition of corn oil to diesel-GTL- waste cooking oil fuel blend was found to be slightly effective only at high engine speeds (above 2200 rpm) where there is only a short time available for air-

fuel mixing. The use of this fuel blend has resulted in 10% decrement in CO emission at high engine speeds.

2) Carbon Dioxide

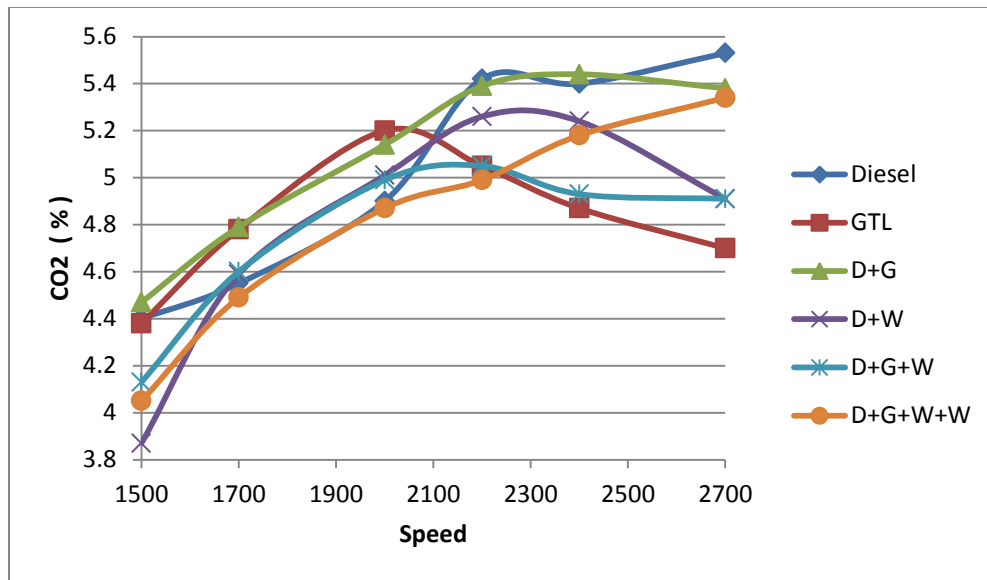


Figure 76. Carbon dioxide emission vs. engine speed for different types of fuel using the 1D manifold

The production of carbon dioxide in the exhaust tailpipe is a function of Hydrogen to Carbon ratio of the fuel and it depends also on the chemical conversion of CO into CO₂. It can be clearly noticed in Figure 76 that CO₂ is increasing as the engine speed increases until it reaches a specific value then starts to decline. The reason behind that is that the equivalence ratio is getting more as the engine speed increases, thus a greater amount of CO is allowed to convert into CO₂. [66]

As noticed in Figure 76, the use of GTL in the pure form has reduced CO₂ emission considerably only at high engine speeds (above 2100 rpm) due to its slightly higher H/C ratio (=2.15) in comparison to diesel fuel (=2.125). The use of GTL has resulted in about 12% CO₂ reduction at high engine speeds. The use of diesel-GTL fuel blend has almost given similar CO₂ emission rates. However, the addition of waste cooking oil to diesel-GTL fuel blend has resulted in about 7% CO₂ reduction at high engine speeds due to the improved oxygen content of the blend caused by the addition of waste cooking oil. The addition of corn oil further to that blend has resulted in about 8% CO₂ reduction at all engine speeds due to the improved oxygen content of the blend caused by the addition of corn oil. Further notice that the use of waste cooking oil with diesel in one fuel blend is also found to be effective in decreasing CO₂ emission, more specifically at high engine speeds (above 2100 rpm). The use of this fuel blend has resulted in 3% CO₂ reduction at high engine speeds.

3) Hydrocarbons

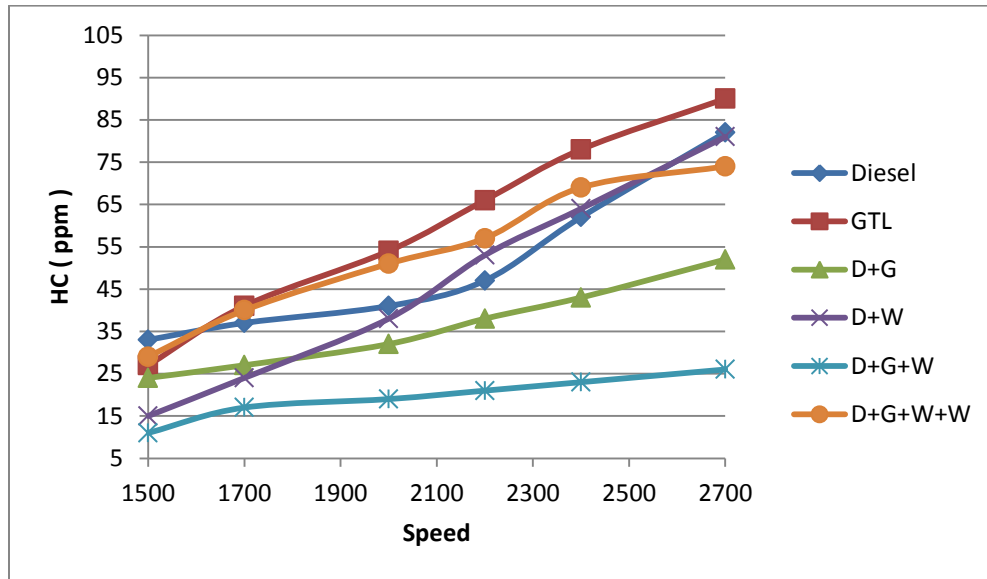


Figure 77. Hydrocarbons emission vs. engine speed for different types of fuel using the 1D manifold

Figure 77 demonstrates the relation between unburned hydrocarbon concentrations with speed increment for six different types of fuels. Unburned HC results mainly because of incomplete combustion of fuel and air. One more source for HC formation is when HC becomes in a direct contact with the walls of the combustion chamber and becomes quenched; more specifically during the cold start period (around 90% of HC is formed).

At the beginning of engine operation (lower engine speeds), it can be clearly noticed that the use of diesel-waste cooking fuel blend is found to be effective due to the high distillation temperature of waste cooking oil (around 318 C) which results in less volatile fuel blend, thus less amount of fuel droplets are quenched and HC emission is reduced. The use of this fuel blend has resulted in 35% reduction in HC emission. However, at higher

engine speeds, the use of diesel-waste cooking oil has almost given similar or slightly higher readings for HC than diesel fuel.

The very low viscosity of GTL fuel in comparison to other used fuels caused the HC emission to be higher because HC results from fuel that over penetrates and wets the cylinder walls during the ignition delay period .As the fuel is less viscous; it is more able to penetrate the cylinder walls and thus results in more HC emission. The use of GTL has resulted in 35% increase in HC emission.

The use of diesel-GTL fuel blend has decreased HC emission by an average percentage of 21%. This may be referred to the improved cetane number of the mixture where less amount of fuel penetrates the cylinder walls due to the shortened ignition delay period.

However, the addition of waste cooking oil to this blend was found to be very effective in reducing HC emission due to the improved distillation temperature of the mixture, thus less amount of fuel is quenched during the engine early operation. The use of this fuel blend has decreased HC emission by an average percentage of 68%.

Nevertheless, the addition of corn oil to that blend has raised HC emission by an average percentage of 10% in comparison to diesel fuel because of its very high viscosity leading to a more incomplete combustion and thus more HC emission. In addition to that, the corn oil has a very low distillation temperature (=160 C) which causes more evaporation of fuel during the cold start period and more fuel is quenched resulting in more HC emission.

4) Nitric Oxide

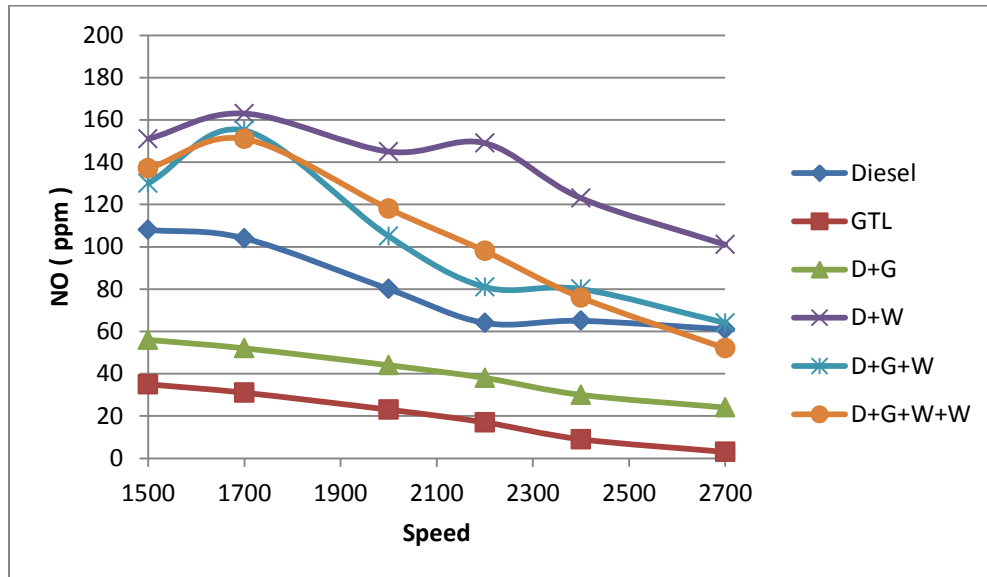


Figure 78. Nitric oxide emission vs. engine speed for different types of fuel using the 1D manifold

Nitric oxide (NO) is formed during the combustion of oil by two mechanisms; high-temperature thermal fixation of molecular oxygen (O₂) and nitrogen (N₂) present in the combustion air and, second, reaction of atmospheric oxygen with nitrogen-containing compounds in the fuel. NO is the major component in the NO_x emission that is why the focus is mainly about it.

It can be noticed from Figure 78 that the values of NO emission at different engine speeds were the lowest when using GTL fuel as it has the highest cetane number in comparison to other used fuels. As a result, the ignition delay period is less and the flame temperature is lower. Consequently, less amount of NO is produced. The use of GTL has reduced NO emission by an average percentage of 52%.

Also, diesel and GTL fuel blend is considered to be a good solution for reducing NO emission in the cylinder when compared with the use of pure diesel due to the enhanced cetane number of the blend. The cetane number was measured for this fuel blend to be equals to 62.5 which is considered to be high with respect to diesel and the other used fuel blends. The use of diesel-GTL fuel blend has reduced NO emission by an average percentage of 40%.

The waste cooking oil which has a very low cetane number (= 32.5) caused the NO emission to be higher when it was blended with diesel alone (around 40% NO increment) or with diesel and GTL together (around 20% NO increment). In the same principle, the addition of corn oil to the aforementioned fuel blend has also resulted in higher NO emission rates due to its low cetane number (around 27% NO increment).

5) Total Particulate Matters

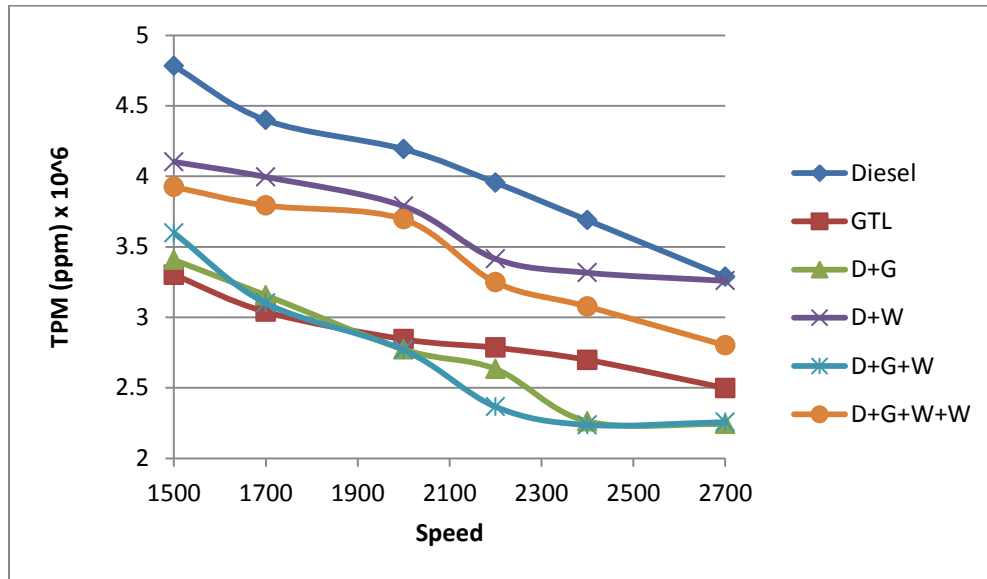


Figure 79. Total particulate emission vs. engine speed for different types of fuel using the 1D manifold

Incomplete combustion is the main reason of the particulate matters emission in diesel engine. Particulate matters are formed mainly in the fuel rich regions during the diffusion burning period. As can be noticed in Figure 79, the total amount of different size particles detected in the engine exhaust decreases with increasing engine speed. The reason behind that is that better turbulence effect is obtained at higher engine speeds in which the extent of complete combustion is improved [67].

Also, it can be observed that using pure GTL fuel or using it in fuel blends is very effective in decreasing PM emission because it has a very low –sulfur and aromatics content in comparison to diesel fuel. The use of pure GTL has reduced PM emission by an average

percentage of 31% and the use of diesel-GTL fuel blend has reduced PM emission by 29% which indicates that the use of GTL is cleaner for the environment than diesel fuel.

The addition of biofuels like the corn oil or using waste cooking oil in fuel blends with the conventional fuels (diesel and GTL) is considered to be a good solution for decreasing the PM emission as those alternative fuel blends also contains less amount of sulfur and aromatics, thus less amount of PM are produced in comparison to diesel fuel. The usage of waste cooking oil with conventional fuels has reduced PM emission by 25% and the addition of corn oil to that fuel blend has resulted in 18% reduction. Also, the use of waste cooking oil with diesel fuel has resulted in 14% reduction in PM emission on average.

6) Smoke number

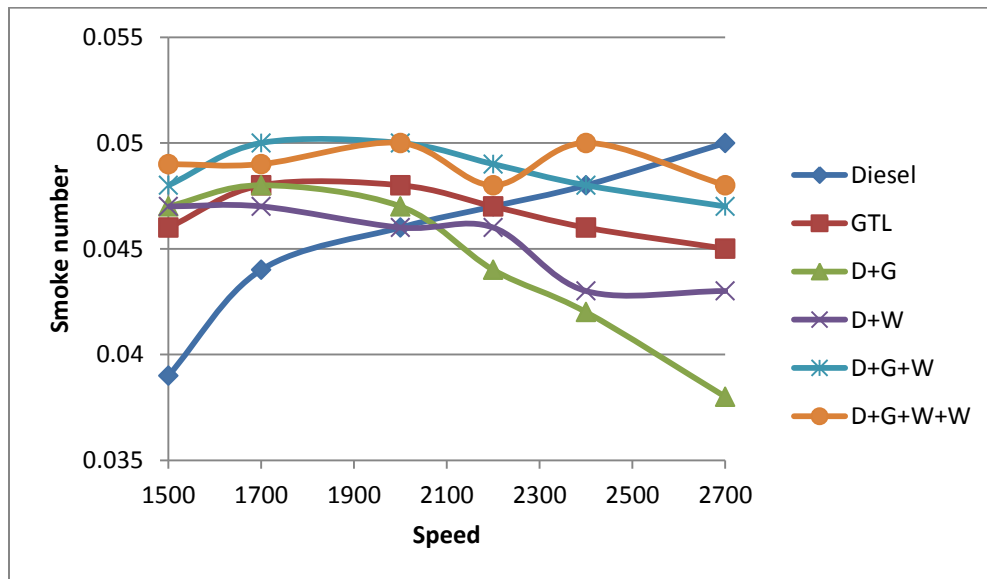


Figure 80. Smoke emission vs. engine speed for different types of fuel using the 1D manifold

The black smoke produced by a fuel is composed of carbon particles released by the thermal cracking of the large hydrocarbon fuel molecules. This process occurs on the rich side of the flame front during the diffusion combustion phase as the PM does. Also, Smoke production is increased at elevated temperatures and this explains why the smoke production in Figure 80 increases with engine speed. Another possible reason is that higher engine speeds lead to a shorter residence time of gases in the combustion chamber.

It can be observed from Figure 80 that the use of waste cooking oil and corn oil in fuel blends with conventional fuels (diesel and GTL) has decreased the smoke production slightly. The reason behind can be referred to their high oxygen content which plays a major role in reducing soot formation and in soot oxidation [68]. The addition of waste cooking oil to diesel-GTL fuel blend has resulted in 23% reduction in smoke opacity and the addition of corn oil further to that mixture has decreased smoke opacity by 26%.

However, the use of diesel fuel with waste cooking oil has resulted in a lower percentage of smoke production which is about 10%.

The use of GTL or diesel-GTL fuel blend is only effective in decreasing smoke production at high engine speeds (above 2200 rpm). The use of pure GTL has resulted in 4% smoke reduction and the use diesel-GTL fuel blend has resulted in 13% smoke reduction at high engine speeds as the chemical structure of the conventional fuels is lack of oxygen content and mainly consists of hydrocarbons. However, the addition of waste cooking oil further to that mixture has decrease smoke concentration by an average percentage of 6% at high engine speeds.

Constant Speed

1) Carbon Monoxide

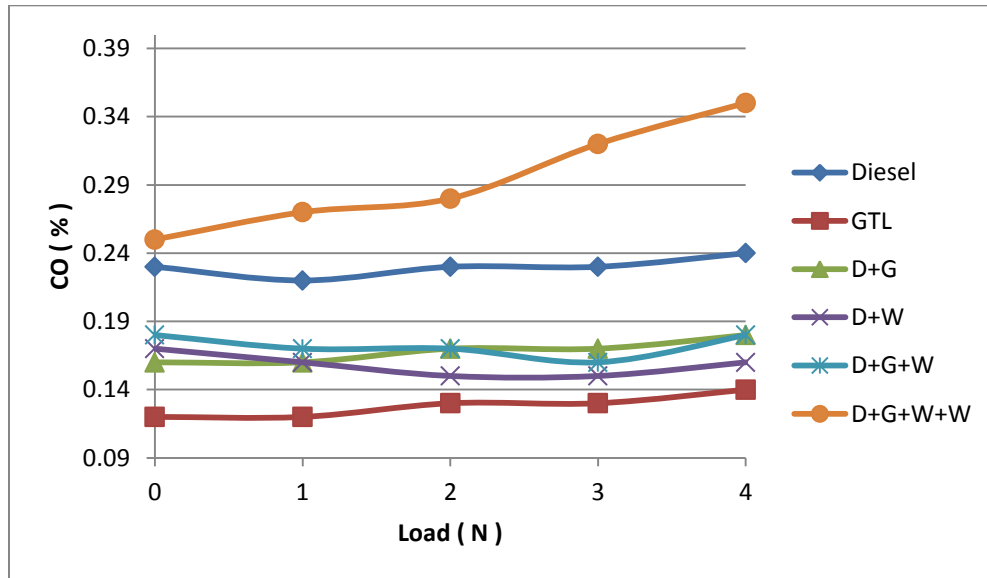


Figure 81. Carbon monoxide emission vs. engine load for different types of fuel using the normal manifold

Carbon monoxide occurs only in the engine exhaust .It is resulted as the product of incomplete combustion. Figure 81 shows the Carbon monoxide readings at different loads for six different types of fuels. As it can be figured out, CO emission increases with load. This is because more fuel is accumulated at higher loads to produce more power due to which higher temperature is achieved in the exhaust.

It can be clearly noticed that the use of GTL fuel which has the highest cetane number (=70) amongst other used fuels has resulted in better combustion characteristics and a more complete combustion. Also, GTL low viscosity played a major role in enhancing the air-

fuel mixing which has resulted in a higher quality of air-fuel mixing and thus less CO emission. The use of GTL fuel instead of diesel fuel can reduce CO emission by about 77%.

In the same concept, the addition of GTL to diesel as expected could enhance the air-fuel mixing quality and has resulted in 30% CO emission reduction. The addition of waste cooking oil to this fuel blend was found to be very effective in decreasing CO emission at all engine loads. The reason behind that can be referred to the high oxygen of waste cooking oil and the high cetane number of GTL in which both fuel properties have resulted in better air-fuel mixing quality. The use of this fuel blend has decreased CO emission rates by an average percentage of 42%. Also, the addition of waste cooking oil to pure diesel in a 50 to 50 volume percentage has recorded a remarkable reduction in CO emission of about 26%. Although the corn oil has high oxygen content, however its very high viscosity has affected the mixing quality more and a poor air-fuel mixing was obtained. The use of this fuel blend has resulted in 20% increment in CO emission.

2) Carbon Dioxide

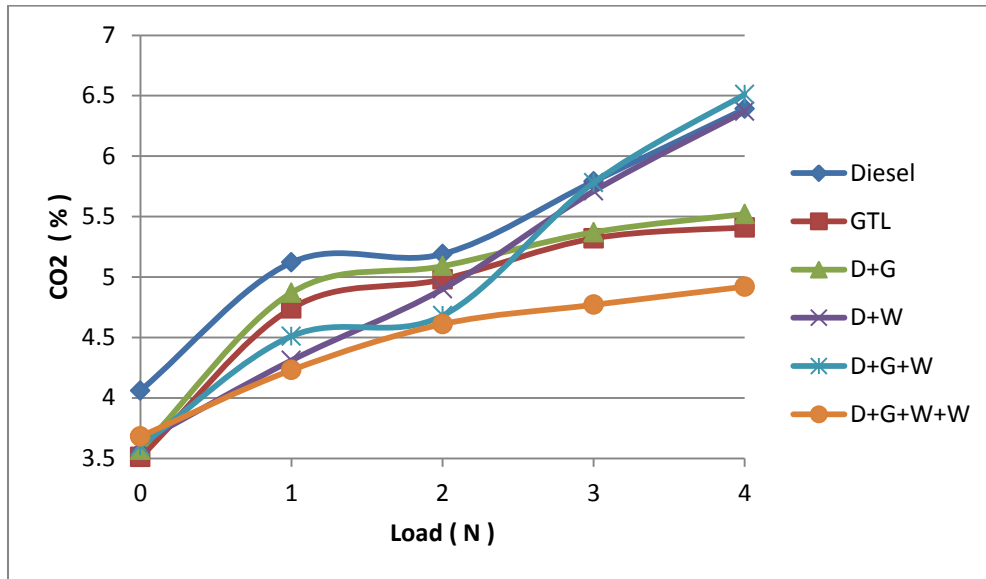


Figure 82. Carbon dioxide emission vs. engine load for different types of fuel using the normal manifold

The production of carbon dioxide in the exhaust tailpipe is a function of Hydrogen to Carbon ratio of the fuel and it depends also on the chemical conversion of CO into CO₂. It can be clearly noticed in Figure 82 that CO₂ is increasing as the engine load increases. The reason behind that is that the mixing charge becomes richer at higher engine loads, thus more fuel is burned and CO₂ emission increases.

As noticed in Figure 82, the use of GTL in the pure form has reduced CO₂ emission slightly due to its higher H/C ratio (=2.15) in comparison to diesel fuel (=2.125). The use of GTL has resulted in 14% CO₂ reduction. As expected, the addition of GTL to diesel in a 50 to 50 volume percentage composition has resulted in CO₂ reduction due to the slightly higher

H/C ratio of that mixture when compared to pure diesel fuel. The use of diesel-GTL fuel blend has resulted in about 12% CO₂ reduction.

However, the addition of waste cooking oil to diesel-GTL fuel blend has resulted in about 13% CO₂ reduction at low and mid-range loads due to the improved oxygen content of the fuel blend caused by the addition of waste cooking oil. The addition of corn oil further to that blend has resulted in about 17% CO₂ reduction at all engine loads due to the improved oxygen content of the blend caused by the addition of corn oil. Further notice that the use of waste cooking oil with diesel in one fuel blend is also found to be effective in decreasing CO₂ emission, more specifically at low engine loads (less than 2N.m). The use of this fuel blend has resulted in 16% CO₂ reduction at low engine loads.

3) Hydrocarbons

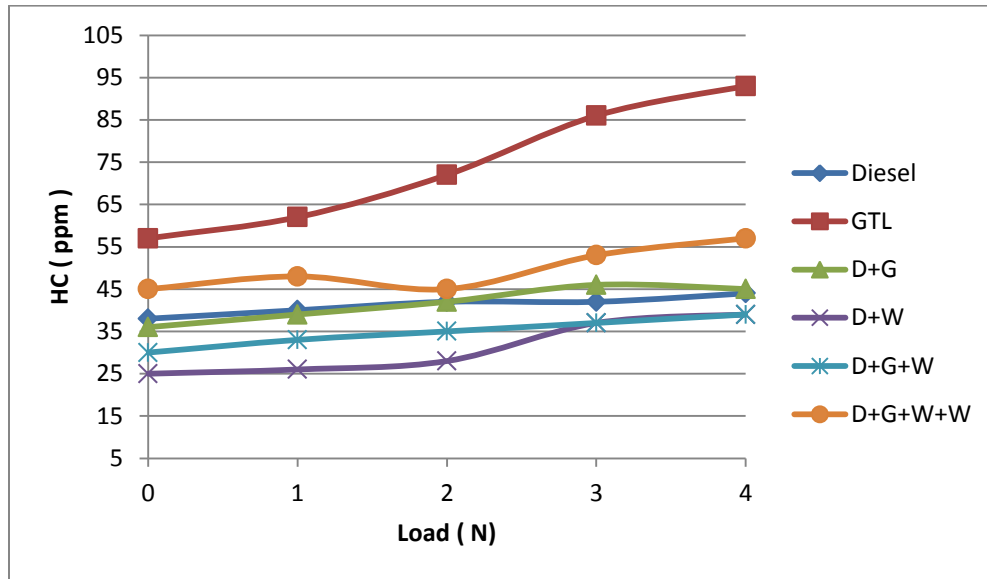


Figure 83. Hydrocarbons emission vs. engine load for different types of fuel using the normal manifold

Figure 83 demonstrates the relation between unburned hydrocarbon concentrations with load increment for six different types of fuel. Unburned HC results mainly because of incomplete combustion of fuel and air. One more source for HC formation is when HC becomes in a direct contact with the walls of the combustion chamber and becomes quenched; more specifically during the cold start period (around 90% of HC is formed).

It can be clearly noticed that the use of diesel-waste cooking fuel blend has resulted in the lowest HC readings. The reason behind that can be the high distillation temperature of waste cooking oil (around 318 C) which results in less volatile fuel blend, thus less amount of fuel droplets are quenched during the cold start period and HC emission is reduced. The use of this fuel blend has resulted in 33% reduction in HC emission.

The very low viscosity of GTL fuel in comparison to other used fuels caused the HC emission to be higher because HC results from fuel that over penetrates and wets the cylinder walls during the ignition delay period .As the fuel is less viscous; it is more able to penetrate the cylinder walls and thus results in more HC emission. The use of GTL has resulted in 35% increase in HC emission. In addition to that, GTL fuel has a distillation temperature that is slightly less than diesel fuel which caused more fuel droplets to be quenched during the cold start period.

The use of diesel-GTL fuel blend has decreased HC emission by only 3%. This may be referred to the improved cetane number of the mixture where less amount of fuel penetrates the cylinder walls due to the shortened ignition delay period. However, the addition of waste cooking oil to this blend was found to be very effective in reducing HC emission due to the improved distillation temperature of the mixture, thus less amount of fuel is quenched during the engine early operation. The use of this fuel blend has decreased HC emission by an average percentage of 18%.

Nevertheless, the addition of corn oil to that blend has raised HC emission by an average percentage of 18% in comparison to diesel fuel because of its very high viscosity leading to a more incomplete combustion and thus more HC emission. In addition to that, the corn oil has a very low distillation temperature (=160 C) which causes more evaporation of fuel during the cold start period and more fuel is quenched resulting in more HC emission.

4) Nitric Oxide

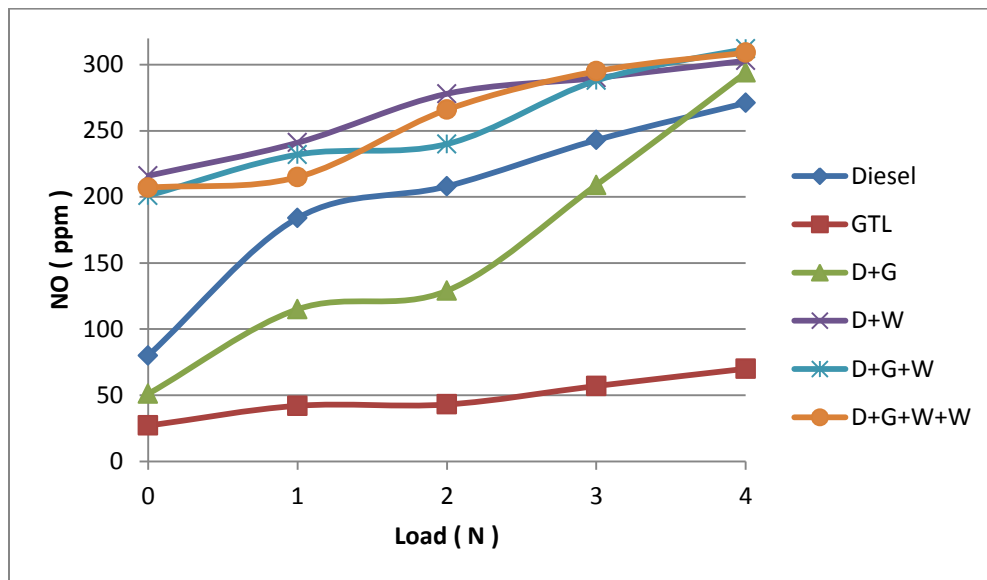


Figure 84. Nitric oxide emission vs. engine load for different types of fuel using the normal manifold

Nitric oxide (NO) is formed during the combustion of oil by two mechanisms; high-temperature thermal fixation of molecular oxygen (O₂) and nitrogen (N₂) present in the combustion air and, second, reaction of atmospheric oxygen with nitrogen-containing compounds in the fuel. NO is the major component in the NO_x emission that is why the focus is mainly about it. As noticed in Figure 84, NO emission becomes higher with more engine load. The reason behind that is that the flame temperature increases as the engine load increases which results in higher NO emission rates.

It can be clearly noticed from Figure 84 that the values of NO emission at different engine loads were the lowest when using GTL fuel due to its highest cetane number. As a result, the ignition delay period is less and the flame temperature is lower. Consequently, less

amount of NO is produced. The use of GTL has reduced NO emission by an average percentage of 66%.

Also, diesel-GTL fuel blend is considered to be a good solution for reducing NO emission due to the enhanced cetane number of the blend. The cetane number was measured for this blend to be equals to 62.5 which is considered to be high with respect to diesel and the other used fuel blends. The use of diesel-GTL fuel blend has reduced NO emission by an average percentage of 36%.

The waste cooking oil which has a very low cetane number (= 32.5) has caused 31% increment in NO emission when it was blended with diesel alone and 26% when it was added to diesel and GTL together. In the same principle, the addition of corn oil to the aforementioned fuel blend was ineffective and has increased NO emission by an average percentage of 17%.

5) Total Particulate Matters

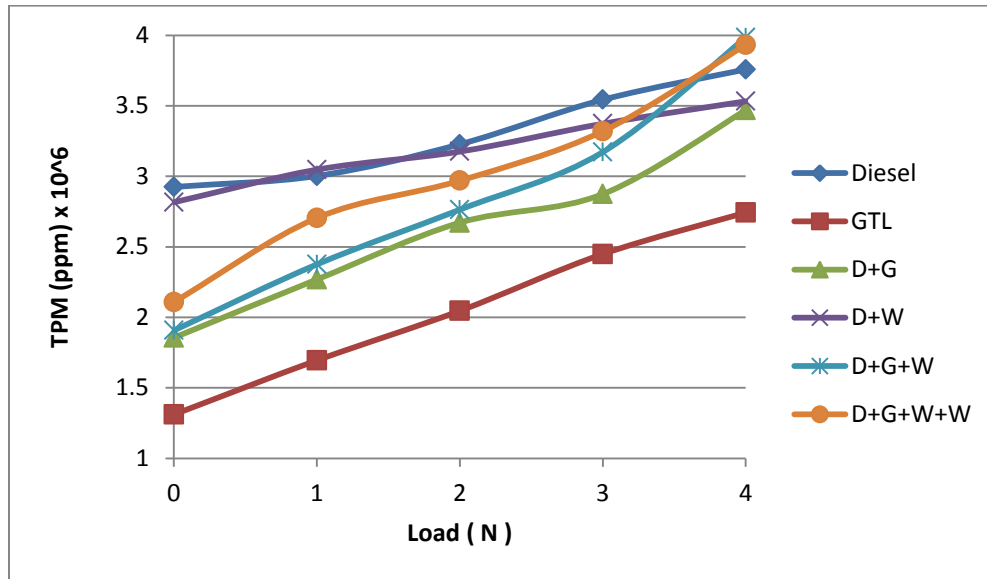


Figure 85. Total particulate emission vs. engine load for different types of fuel using the normal manifold

Incomplete combustion is the main reason of the particulate matters emission in diesel engine. Particulate matters are formed mainly in the fuel rich regions during the diffusion burning period. As can be noticed in Figure 85, the total amount of different size particles detected in the engine exhaust increases with increasing engine load. The reason behind that is that the air-fuel charge becomes richer at higher engine loads which results in more PM.

It can be observed from Figure 85 that using pure GTL fuel or using it in fuel blends is very effective in decreasing PM emission because it has a very low –sulfur and aromatics content in comparison to diesel fuel. The use of pure GTL has reduced PM emission by an

average percentage of 52% and the use of diesel-GTL fuel blend has reduced PM emission by 38% which indicates that the use of GTL is cleaner for the environment than diesel fuel. The addition of a biofuel such as the corn oil or using waste cooking oil in fuel blends with the conventional fuels (diesel and GTL) is considered to be a good solution for decreasing the PM emission as those alternative fuel blends also contains less amount of sulfur and aromatics, thus less amount of PM are produced in comparison to diesel fuel. The usage of waste cooking oil with conventional fuels has reduced PM emission by 10% and the addition of corn oil to that fuel blend has resulted in 8% reduction. Also, the use of waste cooking oil only with diesel fuel has resulted in 5% reduction in PM emission on average.

6) Smoke number

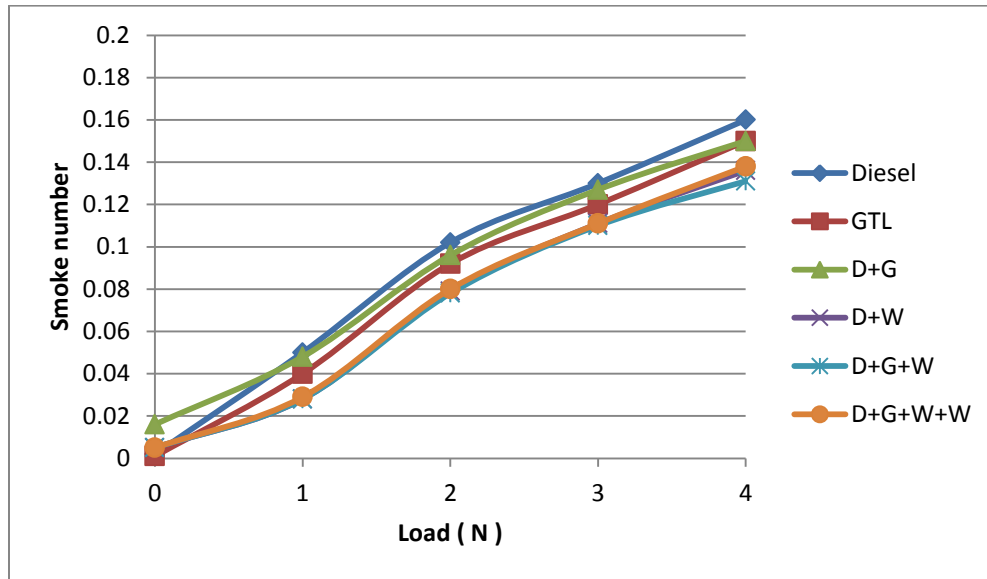


Figure 86. Smoke emission vs. engine load for different types of fuel using the normal manifold

The black smoke produced by a fuel is composed of carbon particles released by the thermal cracking of the large hydrocarbon fuel molecules. This process occurs on the rich side of the flame front during the diffusion combustion phase as the PM does. Also, Smoke production is increased at elevated temperatures and higher loads and this explains why the smoke production in Figure 86 is increasing with engine load.

The use of waste cooking oil and corn oil in fuel blends with conventional fuels (diesel and GTL) has decreased the smoke production slightly. The reason behind that can be referred to their high oxygen content which plays a major role in reducing soot formation and in soot oxidation [68]. The addition of waste cooking oil to diesel-GTL fuel blend has

resulted in 42% reduction in smoke opacity and the addition of corn oil further to that mixture has decreased smoke opacity by 44% in comparison with pure diesel. Also, the usage of waste cooking oil with diesel in one fuel blend has resulted in a lower percentage of smoke production which is around 40%.

The use of GTL or diesel-GTL fuel blend has slightly decreased the smoke production. The use of pure GTL has reduced smoke reduction by about 10% and the use diesel-GTL fuel blend has resulted in 4% smoke reduction as the chemical structure of those conventional fuels is lack of oxygen content and mainly consists of hydrocarbons.

4.3 Summary of the Results

Table 11 and Table 12 summarize the results obtained in section 4.1. Also, a summary for the results obtained in section 4.2 is included in Table 13 and Table 14. These tables show the percentage enhancement or demotion for every combustion, performance or emission criteria with respect to the normal manifold which is taken as a reference for comparison in Table 11 , Table 12. In the same concept, Table 13 and Table 14 present the percentage enhancement or demotion for the tested fuel with respect to diesel fuel.

Table 11

Comparison between new intake manifolds at a constant engine load (Load=1N.m)

Criteria	Type of manifold		
	1D	2D	3D
	% Enhancement of Demotion		
In-cylinder Peak Pressure (bar)	+10%	+10%	-7%
Maximum Pressure Raise Rate (bar/deg)	+7%	+7%	-3%
Exhaust Temperature (C)	+12%	-7%	+6%
BSFC (kg/kw.hr)	+5%	+7%	-8%
Volumetric Efficiency (%)	+10%	-10%	-3%
Carbon Monoxide (%)	+12%	-31%	-38%
Carbon Dioxide (%)	+18%	-10%	-9%
Hydrocarbons (ppm)	+19%	-44%	-68%
Nitric Oxide (ppm)	+32%	+44%	-41%
Particulate Matters (ppm)	+15%	+8%	+8%
Smoke Number	+5%	-2%	-4%

(+): indicates for enhancement, (-): indicates for demotion

Table 12

Comparison between new intake manifolds at a constant engine speed (speed=1700 rpm)

Criteria	Type of manifold		
	1D	2D	3D
	% Enhancement of Demotion		
In-cylinder Peak Pressure (bar)	+5%	+8%	-3%
Maximum Pressure Raise Rate (bar/deg)	+5%	+5%	-1%
Exhaust Temperature (C)	+7%	-11%	+7%
BSFC (kg/kw.hr)	+5%	-21%	-28%
Volumetric Efficiency (%)	+4%	-11%	-8%
Carbon Monoxide (%)	+48%	+21%	-17%
Carbon Dioxide (%)	+19%	+7%	-31%
Hydrocarbons (ppm)	+52%	+61%	-20%
Nitric Oxide (ppm)	+32%	+35%	-10%
Particulate Matters (ppm)	+5%	-8%	-20%
Smoke Number	+3%	-1%	-1%

(+): indicates for enhancement, (-): indicates for demotion

Table 13

Comparison between tested fuels at a constant engine load (Load=1N.m)

Criteria	Fuel				
	GTL	DG	DW	DGW	DGWW
	% Enhancement or Demotion				
In-cylinder Peak Pressure (bar)	+18%	+4%	-2%	-2%	-14%
Maximum Pressure Raise Rate (bar/deg)	+19%	+5%	-10%	-10%	-17%
Exhaust Temperature (C)	+37%	+10%	-7%	+4%	-15%
BSFC (kg/kw.hr)	+20%	+10%	-15%	-4%	-17%
Volumetric Efficiency (%)	-1%	-5%	+2%	+5%	+8%
Carbon Monoxide (%)	+14%	+12%	+29%	+42%	-10%
Carbon Dioxide (%)	+12%	+1%	+3%	+7%	+8%
Hydrocarbons (ppm)	-35%	+21%	+35%	+68%	-10%
Nitric Oxide (ppm)	+52%	+40%	-40%	-20%	-27%
Particulate Matters (ppm)	+31%	+29%	+14%	+25%	+18%
Smoke Number	+4%	+13%	+23%	+6%	+26%

(+): indicates for enhancement, (-): indicates for demotion

Table 14

Comparison between tested fuels at a constant engine speed (speed=1700 rpm)

Criteria	Fuel				
	GTL	DG	DW	DGW	DGWW
	% Enhancement or Demotion				
In-cylinder Peak Pressure (bar)	+11%	+5%	-4%	-4%	-13%
Maximum Pressure Raise Rate (bar/deg)	+18%	+7%	-4%	-4%	-9%
Exhaust Temperature (C)	+26%	+16%	+16%	+16%	-36%
BSFC (kg/kw.hr)	+12%	+7%	-13%	-13%	-15%
Volumetric Efficiency (%)	-1%	-5%	+7%	+7%	+10%
Carbon Monoxide (%)	+77%	+30%	+26%	+26%	-20%
Carbon Dioxide (%)	+14%	+12%	+16%	+16%	+17%
Hydrocarbons (ppm)	-35%	+3%	+33%	+33%	-18%
Nitric Oxide (ppm)	+66%	+36%	-31%	-31%	-17%
Particulate Matters (ppm)	+52%	+38%	+5%	+5%	+8%
Smoke Number	+10%	+4%	+40%	+40%	+44%

(+: indicates for enhancement, (-): indicates for demotion

CHAPTER 5. CONCLUSION & RECOMMENDATIONS

5.1 Conclusions

In conclusion, this project investigates the performance and emission characteristics of diesel engine when varying two parameters. The first parameter was the use of newly designed intake manifolds and the second one was the use of new fuel blends. The new intake manifolds have been 3D designed using Solid Works and manufactured using Fused Deposition Modeling (FDM) and the new fuel blends were carefully prepared at the laboratory and their properties were provided by chemical specialists.

A test rig that combines a single cylinder, four stroke diesel engine with all the measuring devices connected to the engine was used to perform all the experiments. The results of the experiment have considered combustion, performance and emission criteria in the scope of this study.

After analysis, it was found that the use of the 1D new manifold can minimize the maximum pressure raise rate by 7% (10 % at a constant engine speed) and the in-cylinder peak pressure by 10% (5% at a constant engine speed). In addition, the use of 1D manifold has raised the volumetric efficiency and the exhaust temperature readings slightly and has resulted in lower bsfc values. Moreover, it has resulted in significant decrement in all exhaust emission gases in addition to PM and smoke opacity. However, the use of 2D manifold was found to be effective in some criteria such as decreasing the in-cylinder peak pressure, the maximum pressure raise rate and the brake specific fuel consumption. However, its use has resulted in a lower exhaust temperature and volumetric efficiency

readings. Moreover, it has resulted in a higher emission gases except NO and PM which have been found to be less with the use of the 2D manifold. Controversy, the use of the 3D manifold was found to be ineffective in any of the engine combustion, performance or emission criteria except the exhaust temperature which was found to be slightly higher and the PM emission which was found to be less with the use of the 3D manifold.

The use of GTL fuel has significantly improved the engine performance and lower all the emission gases except HC (35% increment). Also, the use of diesel- GTL fuel blend (DG) has almost enhanced the same criteria with slightly less enhancement percentages and the emission of HC was found to be less with the use of this fuel blend. However, the use of the new fuel blends that contain waste cooking oil namely DW and DGW was found to be ineffective in enhancing engine performance (despite the volumetric efficiency) and its combustion characteristics. Nevertheless, it has resulted in significant decrement in all exhaust emission gases (except NO) in addition to PM and smoke concentration. The addition of corn oil to those fuel (DW and DGW) was considered to be a good solution for only increasing the volumetric efficiency and decreasing some of the emission gases such as CO₂, PM and smoke concentration, however its drawbacks were apparent in the remaining criteria.

Finally, it was concluded that the use of the 1D new manifold is more preferable than the standard normal manifold due to its significant improvement of the engine performance and lowering all of its emission gases. However, the use of GTL fuel instead of diesel fuel is considered to be always a safe road for a higher engine performance and a cleaner environment.

5.2 Future Work and Recommendations

The following recommendations are suggested for future work on the new engine intake manifolds and fuel blends:

- It is recommended to perform the experiments on a well-ventilated area to avoid respiratory problems
- The fuel preparation should be done with very much caution as some fuels are highly volatile and extremely hazardous
- It is recommended to perform a calibration test for the measurement devices the most frequently to make sure they are operating in an accurate way.
- The selection of the volume percentage composition of any biofuel in the blend should be done with more study and care prior to the experiment as it plays a major role in characterizing the engine performance and emission.
- For future work, a set of new intake manifolds can be designed by using different inner diameters or outlet angles or both.
- Also, new fuel blends can be prepared by adding other biofuels to diesel with different volume percentage composition and then they can be experimentally tested and compared with diesel fuel

REFERENCES

- [1] “Induction Manifold as an Arrangement of Pipes” 2017. [Online]. Available: www.ebay.com/itm/SKUNK2-ULTRA-BLACK-INTAKE-MANIFOLD-90MM-THROTTLE-BODY--02-05-HONDA-CIVIC-SI-3-5L-/391394379227. [Accessed: 10-Nov-2017].
- [2] W.Pulkrabek, Handbook of Engineering Fundamentals of Internal Combustion Engine, 4th ed. New Jersey: Prentice Hall, 2004.
- [3] R. Costa and J. Sodre, "Compression ratio effects on an ethanol/gasoline fuelled engine performance", Applied Thermal Engineering, vol. 31, pp. 278-283, 2011.
- [4] “Electronic Fuel Injection System” 2014. [Online]. Available: <https://www.slideshare.net/johnalexis3/efis-33404084>. [Accessed: 10-Nov-2017].
- [5] “Direct injection diesel engine” 2009. [Online]. Available: <http://www.dieselhub.com/tech/idi-vs-di.html> [Accessed: 14-Nov-2017].
- [6] “Induction Manifold Design” 2014. [Online]. Available: www.truckcustomizers.com. [Accessed: 10-Nov-2017].
- [7] J.B.Heywood, Internal Combustion Engine Fundamentals, McGraw-Hill, Singapore, 1988.
- [8] T. Rozsas, W.Brandstetter W, “Optimization of the charge process in modern motor vehicle spark ignition engines”. SAE Paper No: 885058, 1988.

- [9] R.Margary, E.Nino,C. Vafidis. The effect of intake duct length on the in-cylinder air motion in a motored diesel engine. SAE Paper No: 900057, 1990.
- [10] P. Ramakrishna, K. Govinda and T. Venkata .Experimental Investigation on Diesel Engines by Swirl Induction with Different Manifolds. International Journal of Current Engineering and Technology SAE Paper 981020. E-ISSN 2277 – 4106,
- [11] A.Chen, A.Veshagh intake Flow prediction of a Transparent DI Diesel Engine. SAE Paper 981020.
- [12] Bugrake, Design of an inlet track of a small IC Engine for swirl enhancement; 1981.
- [13] V.Ganesan, Akira, Flow field development in a direct injection diesel engine with different manifolds; 1990.
- [14] G.Valentino, K.Duane and F.Patrick ,“Intake Valve Flow Measurements Using PIV”, SAE Paper No. 932700.1993.
- [15] J.Lee and P.V.Farrell, “Intake Valve Flow Measurements of an IC Engine Using Particle Image Velocimetry”, SAE Paper No. 930480,1993.
- [16] S.Nadarajah, S.Balabani, M.J.Tindal and M. Yianneskis ,“The effect of swirl on the annular flow past an axisymmetric poppet valve”, Proc I Mech E - Vol 212, Part C,1998.

- [17] P. Ramakrishna, K. Govinda, T. Venkata, Experimental Investigation on Diesel Engines by Swirl Induction with Different Manifolds, International Journal of Current Engineering and Technology. E-ISSN 2277, 2014.
- [18] B.Robert, Diesel-Engine Management: System and Components, fourth ed., Bently Publishers, Cambridge, MA, USA, 2005.
- [19] J.E. Dec, SAE Technical Paper 970873, 1997.
- [20] M.P.B Musculus, P.C. Miles, L.M. Pickett, Prog. Energy Combust. Sci. 39 246–283,2013
- [21] C.A. Idicheria, L.M. Pickett, SAE Technical Paper 2006-01-3434, 2006.
- [22] J.B. Heywood , Internal Combustion Engine Fundamentals , McGraw-Hill, New York, 1988 .
- [23] L.M. Pickett, D.L. Siebers, Non-Sooting, SAE Technical Paper 2004-01-1399, 2004.
- [24] L.M. Pickett , J. Manin , C.L. Genzale , D.L. Siebers , M.P.B. Musculus , C.A. Idicheria , SAE Int. J. Engines 4 764–799,2011.
- [25] D.L. Siebers, B.S. Higgins. SAE Technical Paper 2011-01-0530, 2011.
- [26] U.S. Department of Energy, January 2016.
- [27] W. Ghazali , R. Mamat , H.H. Masjuki , G. Najafi, Renewable Sustainable Energy Rev 51 585–602,2015 .

- [28] M.G. Varanda , G. Pinto , F. Martins , *Fuel Process. Technol.* 92 1087–1094, 2011.
- [29] R. Edwards , H. Hass , J. Larive , L. Lonza , H. Maas , D. Rickeard , *Well-to-Wheels Report Version 4.a, JEC Well-to-Wheels Analysis , European Commis- sion Joint Research CentreInstitute for Energy and Transport, ISBN 978-92-79-33887-8,2014.*
- [30] L.E. Oliveira , M. DaSilva ,in: *International Conference on Renewable Energies and Power Quality (ICREPQ'13) Bilbao, Spain, 2013 .*
- [31] A. Gopinath , S. Puhan , G. Nagarajan , *Int. J. Energy Environ.* 1 295–306 ,2003.
- [32] N. Carareto , C. Kimura , E. Oliveira , M. Costa , A. Meirelles , *Fuel* 96 319–326 ,2012.
- [33] I. Lee , L. Johnson , E. Hammond , *J. Am. Oil Chem. Soc.* 72 ,1155–1160,1995.
- [34] O. Ozener , L. Yuksek , A.T. Ergenc , M. Ozkan , *Fuel* 115 , 875–883 ,2014.
- [35] M. Lapuerta , J. Rodríguez-Fernández , J.R. Agudelo , *Bioresour. Technol.* 99, 731–740,2008.
- [36] United States Environmental Protection Agency (EPA), *Comprehensive Analysis of Biodiesel Impacts on Exhaust Emissions, EPA420-02-001, 2002.*
- [37] S. Kalligeros , F. Zannikos , S. Stournas , et al. , *Biomass Bioenergy* 24 , 141–149 ,2003.
- [38] C.K. Westbrook , W.J. Pitz , H.J. Curran , *J. Phys. Chem. A* 110 6912–6922,2006.

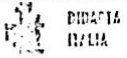
- [39] J. Kim , H. Park , C. Bae , in: 21st International Symposium on Alcohol Fuels (ISAF), Gwangju, Korea, March 10-13, 2015 .
- [40] J. Schmitgal , J. Tebbe , JP-8 and Other Military Fuels (No. TARDEC-22491) , Army Tank Automotive Research Development and Engineering Center, Warren, MI, 2011 .
- [41] G. Fernandes , J. Fuschetto , Z. Filipi , D. Assanis , H. McKee , Proc. Inst. Mech. Eng. Part D: J. Auto- mobile Eng. 221 ,957–970 ,2007.
- [42] A.K. Pandey , M.R. Nandgaonkar , SAE Int. J. Fuels Lubr. 3,238–245,2010.
- [43] Y. Park , C. Bae , C. Mounaïm-Rousselle , F. Foucher , Int. J. Engine Res. 16 ,92–103,2015.
- [44] J. Patel, M. Patel, R. Rathod, , Characteristics of vegetable oils for use as fuel in stationary diesel engines towards specifications for a standard in West Africa, IOSR Journal of Mechanical and Civil Engineering, e-ISSN: 2278-1684, 2015.
- [45] “Corn oil - Wikipedia” . [Online]. Available: https://en.wikipedia.org/wiki/Corn_oil. [Accessed: 10-Nov-2017].
- [46] K.Recai, The effect of raw corn oil and diesel fuel mixture on engine performances and emissions, Energy Education Science and Technology, volume (issues) 28(1): 469-474, 2011.
- [47] U.S. Energy Information Administration, International Energy Outlook ,U.S. Department of Energy, Washington, DC, July 2013 DOE/EIA-0484,2009 .

- [48] International Energy Agency (IEA), Energy Technology Perspective (ETP) ,ISBN 978-92- 64-20800-1,2014.
- [49] International Energy Agency (IEA), Transport, Energy and CO2, ISBN 978-92-64-07316- 6, 2009.
- [50] International Energy Agency (IEA), Energy Technology Perspective (ETP) ,ISBN 978-92- 64-17488-7,2012.
- [51] The Outlook for Energy: A View to 2040, Exxon- mobil.
- [52] A. Kleinov_a, I. Vailing, J. L_abaj, J. Mikulec, J. Cvengro_s, Vegetable oils and animal fats as alternative fuels for diesel engines with dual fuel operation, Fuel Process. Technol. 92 , 1980e1986,2011.
- [53] Z. Franco, Q.D. Nguyen, Flow properties of vegetable oil and diesel fuel blends, Fuel 90 ,838e843,2011.
- [54] D.C. Rakopoulos, C.D. Rakopoulos, E.G. Giakoumis, A.M. Dimaratos, M.A. Founti, Comparative environmental behavior of bus engine operating on blends of diesel fuel with four straight vegetable oils of Greek origin: sunflower, cottonseed, corn and olive, Fuel 90 , 3439e3446,2011.
- [55] A. Kleinov_a, I. Vailing, J. L_abaj, J. Mikulec, J. Cvengro_s, Vegetable oils and animal fats as alternative fuels for diesel engines with dual fuel operation, Fuel Process. Technol. 92 ,1980e1986, 2011.

- [56] J. Blin, C. Brunschwig, A. Chapuis, O. Changotade, S.S. Sidibe, E.S. Noumi, P. Girard, Characteristics of vegetable oils for use as fuel in stationary diesel engines towards specifications for a standard in West Africa, *Renew. Sustain. Energy Rev.* 22,580e597, 2013.
- [57] E. Torres-Jimenez, et al., "Physical and chemical properties of ethanol–diesel fuel blends," *Fuel*, vol. 90, pp. 795-802, 2011
- [58] N .Ashok, A Diesel Fuel Extender That’s The Best Fuel. *New Energy and Fuel*, 2011.
- [59] R.Swami, T. Chandrashekar, N .Banapurmath, ,S. Khandal, Impact of Diesel-butanol Blends on Performance and Emission of Diesel Engine, *Oil & Gas Research*, ISSN 2472-0518 2015.
- [60] M. Bassiouny, "An experimental study on the effect of using gas-to-liquid (GTL) fuel on diesel engine performance and emission", *Alexandria Engineering Journal*, vol. 55, Issue 3, 2016
- [61] “Equation for swirl number in an intake manifold”. [Online]. Available: <https://www.google.com/patents/EP2476954A1?cl=en> [Accessed: 11-Nov-2017].
- [62] M.Ceviz, M.Akın, Design of a new SI engine intake manifold with variable length plenum, *Energy Conversion and Management*; 51: 2239–2244, 2010.
- [63] M.Ceviz, Intake plenum volume and its influence on the engine performance, cyclic variability and emissions. *Energy Convers Manage*; 48:961–6, 2007.

- [64] J.Martins, Design of an inlet track of a small IC Engine for swirl enhancement, International Congress of Mechanical Engineering, 20, Gramado, Brazil, 2009.
- [65] S.Nur, Design and development of alternative fuel in direct injection hydrogen fueled engine to engine performance, Universiti Teknologi MARA, Malaysia, 2015.
- [66] A.Saad, Comparative analysis of performance and emission of a homogenous combustion compressed natural-gas direct injection engine, International Journal of Mechanical & Mechatronics Engineering, Malaysia, 2016.
- [67] A.Ali, Reduction a particulate matter of diesel emission by the use of several oxygenated diesel blends fuels, Tafila Technical University, Jordan, 2014.
- [68] A.John, B.Pedro,V.Adrian, Evaluation of nitrogen oxide emissions and smoke opacity in a HSDI diesel engine fuelled with palm oil biodiesel, Universidad de Antioquia, Colombia, 2010.
- [69] “Aerocet 531 operation manual”. [Online]. Available:
<https://www.newstarenvironmental.com/particle-counters/met-one-aerocet-531s-mass-particle-counter-dust-monitor.html>. [Accessed: 20-Nov-2017].

APPENDIX A: AIR MASS FLOW RATE CHART



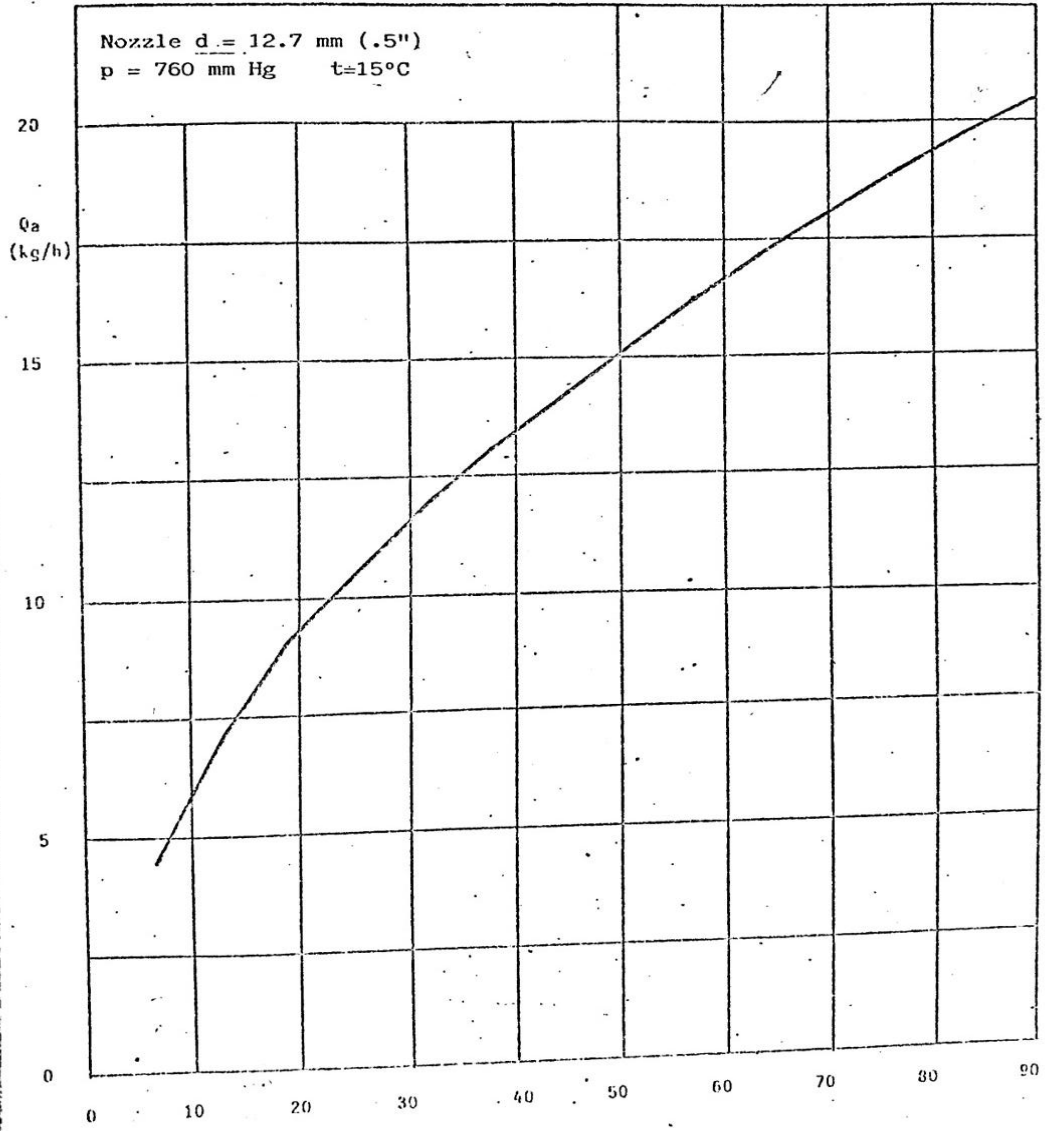
ENTE
C S.p.A.

T85D - INTERNAL COMBUSTION ENGINES TEST BED

page 82

Le proprietà intellettuali di questo disegno è tutelata a termini di legge. È vietato riprodurre o rendere note o terzi e Dito connessioni senza nostra autorizzazione scritta.

E-14081:20
 mod. 34



Δp (mm H₂O)

APPENDIX B: MEASURING DEVICES

1) The Oscilloscope

Description of the Device

The selected oscilloscope for this project is the GW-Instek (Model GDS-3152) Digital Storage Oscilloscope with a sampling rate of 150 MHz as shown in Figure 87.



Figure 87. GW-Instek oscilloscope

The oscilloscope has a function to save the screenshots of the plots, as well as the 48 sampling data in the form of Excel file, which can be used for further analysis. The oscilloscope can also be linked directly to the computer for ease of data retrieval.

This is a digital storage oscilloscope that has a function to save the screenshots of the plots, as well as the 48 sampling data in the form of Excel file, which can be used for further analysis. The oscilloscope can also be linked directly to the computer for ease of data interval. It can be used for many applications such as product design, debugging, repair and serving, and electrical engineering education. Moreover, it has a vertical sensitivity to 2mV per division for capturing low-level signals. It comes with different connection with the computer such as USB, RS232, and LAN interfaces and by using these connections the waveforms and readings can be viewed and data can be collected. Figure 88 shows how the voltage-time wave looks on the oscilloscope screen.

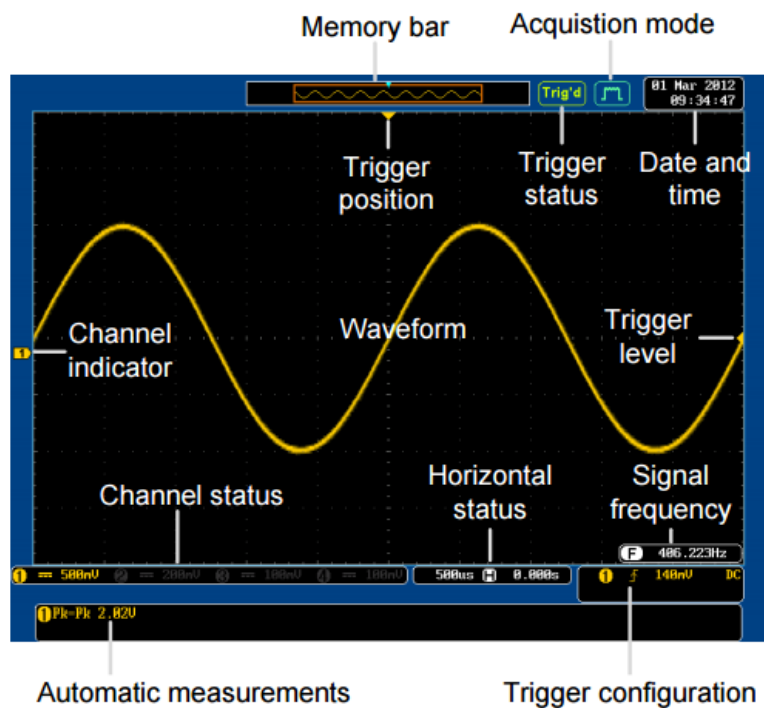


Figure 88. How the waveform is displayed on the oscilloscope's screen

Features and Specifications

The oscilloscope combines a lot of features and its interface has many specifications and they are as the following:

Performance

- 1) High sampling rate: up to 5GSa/s real-time(4GSa/s GDS-350X), 100GSa/ s equivalent time
- 2) Deep memory: 25K points record length
- 3) Minimum 2ns peak detection

Features

- 1) 2 and 4 channel models
- 2) Bandwidth up to 500 MHz
- 3) 5GSa/ s (200ps resolution) real time sampling rate (4GSa/ s, 250ps resolution for GDS-350X)
- 4) 100GSa/ s equivalent sample rate
- 5) VPO waveform processing
- 6) Large 8'' 800 x 600 high-resolution TFT LCD
- 7) Unique split window function
- 8) Flexible application modules
- 9) Three standard input impedances (50Ω/75Ω /1MΩ)
- 10) Optional power measurement functions are available for fast analysis of power quality tests
- 11) 2 and 4 channel models available up to 500MHz

- 12) On-screen Help
- 13) 64 MB internal flash memory
- 14) Free Wave remote control software (free download)

Interface

- 1) USB host port: front and rear panel, for storage devices
- 2) USB slave port(Optional GPIB to USB), RS-232C port: for remote control
- 3) Calibration output
- 4) Go-No-Go output
- 5) Trigger output
- 6) Ethernet port

The Pressure Transducer

The pressure transducer that is selected for that project is the Charge Output Pressure Sensor (Model 116B03) from PCB Piezotronics, having a range of 7 bars and an operating temperature up to 343C. The output pressure response from this sensor is 10 pC/psi, which is then converted by an In-Line Charge Converter (Model 422E35) also from PCB Piezotronics.

The inline charge amplifier converts the signal with a gain of 0.99 mV/pC. Therefore, the final calibration factor used for converting the transducer signal to pressure is 6.238 mV/psi. Figure 89 shows both the pressure sensor and the charge converter.

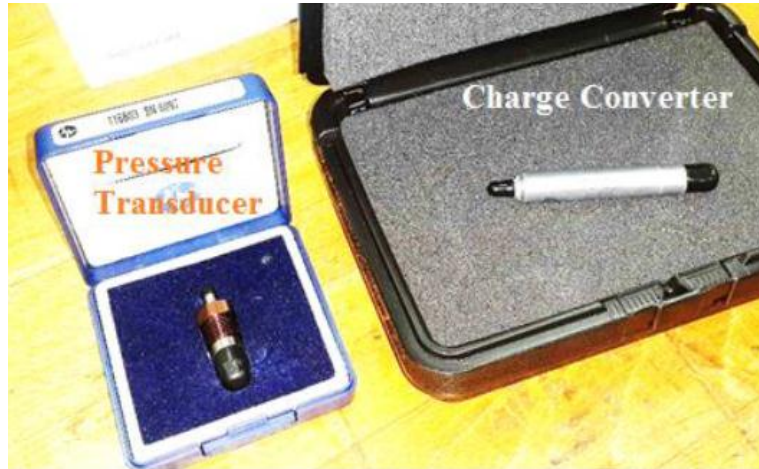


Figure 89. PCB pressure transducer and PCB charge converter

The in-cylinder pressure and crank angle position are obtained by AVL QH 33D water cooled piezoelectric pressure transducer and PALAZZOLI digital shaft encoder; the output of the pressure transducer is amplified by an AVL charge amplifier and then the output signals displayed on Instek GDS-3152 Digital Storage Oscilloscope with 150 MHz sampling rate. The accuracy of this pressure transducer is ± 13 mV/bar. Figure 90 demonstrates the shape of the used pressure transducer.



Figure 90. The pressure transducer used in this experiment

Calibration Process

Even though the pressure sensors are provided with its calibration chart, it was decided to do the re-calibration to test the proper working of the ordered pressure transducers. The calibration was done using a separate sensors calibration testing facility available at Qatar University.

The pressure transducer was calibrated by connecting it with the testing machine and a measured amount of pressure was applied as shown in Figure 91. The output signal was recorded and compared with the calibration charts provided with the sensors. Both the pressure sensors were found to be working perfectly and giving correct readings. The pressure transducer was provided with a testing and calibration certificates which showed the relationship between the output voltage and pressure. These certificates are attached to this report in Appendix C.

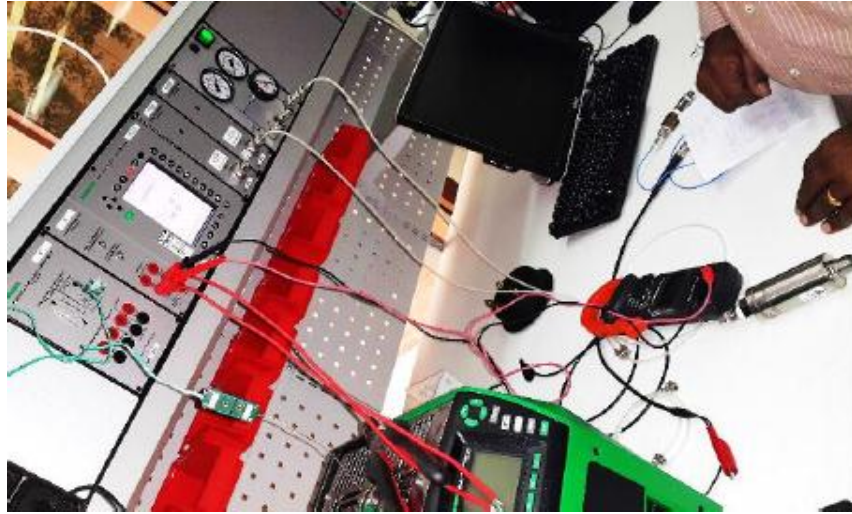


Figure 91. Pressure sensor calibration

2) Gas analyzer

Description of the Device

The HM5000 gas analyzer permits the operator to estimate four or five gas components in the combustible exhaust. It is equipped for determining the volume concentration of O₂ (oxygen), HC (as N-hexane), CO (carbon monoxide), CO₂ (carbon dioxide), and optionally NO_x (nitric oxide).

Based on the knowledge of gas concentrations the analyzer will compute the Lambda (λ), Air to Fuel Ratio (AFR) and Grams per Mile (GPM). It will likewise give a read-out to optional speedometer that peruses up to 30,000 RPM. With this much data in one place, one can concludes and tunes any fuel-related issues while having the greater part of the significant data showed in a super-bright graphical LCD screen. Notwithstanding the above elements, the analyzer can quantify exhaust gasses from two tail pipes and average the

readings, or set up exhaust dilution value to caution if there are air-leakage in the exhaust system.

Having a weight that is less than 2 pounds, the analyzer is assigned for use in different outdoor or indoor situations. The analyzer is designed to be connected to standard electrical plug voltages on a range between 100 VAC to 260 VAC, with a frequency oscillating between 50 to 60 Hz. It likewise has a Nickel hydride metal battery that can be recharged; hence it can be packed and brought to outdoor environment for a drive test to perceive how the vehicle acts in genuine driving circumstances. Utilizing the inner record includes, the information can be put away amid the test drive and the data can be downloaded on a Personal Computer or Laptop when returning back. Figure 92 demonstrates the HM5000 gas analyzer labeled with its features buttons.



Figure 92. HM5000 handled gas analyzer

Features and Specifications

The Gas analyzer unit combines a lot of features in addition to various types of specifications and they are as the following:

Features

- 1) It permits the operator to estimate four or five gas components in the combustible exhaust.
- 2) It has an optional NO_x port.
- 3) It can compute the Lambda (λ), Air to Fuel Ratio (AFR) and Grams per Mile (GPM)
- 4) It provides a read-out for the optional Tachometer that reads up to 30,000 RPM
- 5) It has a super-bright graphic LCD screen.
- 6) Portable, Compact and light weight construction.
- 7) It gives the operator an option to select between 4-stroke and 2-stroke engine configurations.
- 8) Provides the operator with a warning in the Measure Mode that the probe in the exhaust pipe may not be positioned properly or there is an air-leakage in the exhaust system.
- 9) The analyzer as shipped has been calibrated at the factory and is designed to maintain calibration accuracy for extended periods of operation.
- 10) It has the ability to correct itself for temperature and atmospheric pressure variations that continually change through the course of the day.

Principle Gases / Ranges Measured

Table 15 summarizes the type of exhaust gases that the analyzer can read and their ranges

Table 15

Principle gases and their allowed ranges

Principle Gases	Ranges Measured
CO , CO_2 , HC	Non-Dispersive Infrared (NDIR)
NO_x and O_2	Electrochemical Cell
Carbon Monoxide	0 - 10.00 %
Hydrocarbons	0 to 10000 ppm
Carbon Dioxide	0 - 20 %
Oxygen	0 - 25 %
Nitric Oxide	0 to 5000 ppm

Physical and Technical Specifications

Table 16 summarizes the physical and technical specifications of the gas analyzer

Table 16

Physical and technical specifications of the gas analyzer

Parameter	Value
Weight	Less than two pounds
Size	7.5" x 3.5" x 2"
Display	Backlit LCD, Graphic, 128 x 64
Internal Power	Rechargeable Lithium-Ion Battery Pak
External Power	10-16VDC, less than 1A
Operating Temperature	35 to 110 F

Probe Specifications

Table 17 summarizes the technical specifications of the gas analyzer's probe

Table 17

The technical specifications of the gas analyzer's probe

Parameter	Value
Material	Pistol grip with Stainless Steel
Hose Length	10 Centimeters
Requirements	90/97 Bar
Insertion Length	Up to 27 Centimeters

Principle of Operation

Once the gas analyzer is turned on, there will be a POWER INDICATOR to indicate that the power is available. After that, the operator should zero the analyzer so it starts taking readings in reference to the gas concentrations that has been calibrated to during the calibration process. This process takes around 30 seconds to be accomplished automatically.

The user then should allow for the gases to be sensed by the gas analyzer's probe and that could be done by pressing PUMP which allow for the exhaust gases to reach the analyzer. Waiting a few seconds for stable readings, the user then can display the reading on the LCD screen on various ways .Moreover; the readings can be paused for an instance or can

be recorded and replayed to show their variation along a short period of time. Furthermore, data can be printed out using the printer serial port at the bottom of the gas analyzer. Along with the gas concentration readings, the user is able to detect the speed of the engine in RPM using the tachometer installed on the gas analyzer and he can also do a fuel consumption test by operating the Grams per Mile option (GPM).

Calibration Process

The analyzer as transported has been calibrated at the processing plant and is intended to keep up alignment precision for extended times of operation. Because of the complicated system of wiring utilized in the analyzer, repeated calibration is definitely not recommended. Be that as it may, it is prescribed to make a gas calibration about at regular intervals of time to make sure that the analyzer is with the system. A few countries have controls governing the time intervals between the processes of calibration. It is essential to agree to the representing controls for the being area.

When checking the gas analyzer calibration or when performing calibration, a bottle of calibration gas is needed. A gas cylinder that has a high gas pressure should be used with a regulator to regulate it to the required pressure for testing and calibrating the gas analyzer, also this regulator is used to monitor the pressure of calibration process.

These calibration gas cylinders have a gas concentration that is known of HC, CO, CO₂ and NO_x. However these gases as it was mentioned in Table 7 have a range of concentrations. When checking or calibrating the Gas Analyzer it is recommended to use the following concentrations:

- 1) HC (propane) 1200 ppm
- 2) CO_2 12.0%
- 3) CO 4.0%
- 4) Balance N_2

These mentioned values are the default ones for the gas analyzer so when initiating the analyzer they are automatically displayed. Hence, using these concentrations eliminates the need to enter different or new values amid the calibration process.

However, if a new calibration process is needed later on to enter different values, it must accommodate with the following ranges:

- 1) HC (propane) 140 ppm to 3400 ppm
- 2) CO_2 5% to 15%
- 3) CO 0.9% to 8.5%

A calibration certificate that shows the values for the gas concentrations when the calibration process has been done to the HM5000 gas analyzer is attached on Appendix C.

3) Particulate Meter

Description of the Device

The AEROCET 531 is a small portable unit used as a particle counter and a mass concentration detector operated with battery. It can be used in two modes, one of them for counting the number of particles and the other is for detecting the mass concentration of the particles. In the particle counting mode, it displays on the LCD screen the number of particles detected on the exhaust for the particle sizes $>0.5\mu\text{m}$ and $>5.0\mu\text{m}$ after one minute of operation. In the mass concentration mode, it provides the mass concentration of the particle per cubic meter for the sampled air. It can test particle sizes as fractions of PM1, PM2.5, PM7, PM10 and TSP.

The AEROCET 531 estimations can contrast positively and costly reference strategies. The AEROCET 531 utilizations the put away molecule counting information from eight distinctive molecule measure ranges and an exclusive calculation to determine the mass concentration for the airborne measured sample.

The sensor in the AEROCET 531 joins a long life laser diode, a productive light gathering curved (elliptical) mirror and exceptional optics to give a high focus restrain. The AEROCET 531 contains a 6V Ni-MH Self-contained battery pack, a vacuum pump, an isokinetic probe, an electronic microprocessor, a PC interface and a LCD show across the board little bundle. Figure 93 shows the AEROCET 531 particulate meter used in this project.



Figure 93. Met one instruments Aerocet-531 mass particle counter dust monitor

Features and Specifications

The particulate meter combines a lot of features in addition to various types of specifications such as the technical, electrical and the general specifications and they are as the following

Features

- 1) It has two modes of operation: particle count and mass concentration mode.
- 2) It can display the measurement within only one minute of operation with a high accuracy.

- 3) Portable and easy handled and light weight construction.
- 4) It has an LCD screen.
- 5) It has a durable ,long life 6V Ni-MH Self-contained battery pack
- 6) It has a PC interface in which it can display the results on the computer.
- 7) Very simple and can be easily used.
- 8) It can measure the ambient temperature and the relative humidity in addition to particles detection.

Specifications

The performance characteristics as well as the physical, environmental and the electrical characteristics of the particulate meter are listed in table below. In addition, the main and the additional accessories are listed.

Table 18

AEROCERT 531 main features with the accompanying accessories [69]

Performance Characteristics

Particle Mode

Size Of Particles	2 channels of 0.5 and 5.0 μm
Concentration	From 0 to 3,000,000 particles per cubic foot
Sample Time	One minute.
Accuracy	+/-10 percent

Sensitivity 0.5 μm
Flow Rate 0.1 Cubic foot per minute

Mass Mode

Concentration of

Particle Mass TSP, PM10, PM7, PM2.5, and PM1

Concentration Range From 0 - 1 mg/m³

Sample Time Two minutes.

Interface

Keyboard 7-key membrane

Display 16-character x 4 line LCD

Physical Characteristics

Size Height 6.25 inches (15.9 centimeters) Width = 4" (10.2 centimeters)

Thickness 2.1 inches (5.4 centimeters)

Weight 1.94 pounds 31 ounces (0.88 kilograms)

Environmental

Operating 0 to +50 degrees C

Temperature

Storage Temperature -20 to +60 degrees C

Electrical Characteristics

Light Source Laser diode, 5 mW, 780 nm

Power 6 Volt battery pack Ni-MH (self-contained) supplies 8 hrs. typical intermittent operation, 5 hours max. continuous use.

AC AC to DC module, 100 to 240 Volts AC to 9 Volts DC @ 350

Charger/Adapter millAmps typical

Communications RS-232

Certification Meets and/or exceeds ISO, CE, JIS, and ASTM, international certifications

Mass Certification Needs proper usage of the correct K factors specific to the measured material

Accessories

Included

- Isokinetic Sample Probe
- Zero Particulate Filter
- Operation Manual

Communication Software

Custom Serial Cable

AC - DC Converter Module w/IEC AC Power Cord

Screwdriver

Carrying Case

Optional Accessories

Flow Meter

Portable Printer

RH & Temperature Probe

Principle of Operation

The aerosol sample is brought from the exhaust to the unit by a cable and later on the individual particle are counted with scattered laser light and the equivalent mass concentration is calculated using a proprietary algorithm.

Calibration of the Device

The particulate meter was calibrated at the factory using NIST polystyrene spheres. When there is a difference in the reading detected between a typical aerosol and a measured aerosol, a compensation by a “K-factor” must be applied to track that error. Using a software called AEROComm , this “K-factor” can be compensated. The calibration certificate for this device is attached on Appendix C.

4) The Smoke Meter

Description of the Device

Eco smoke 100 is intended for measurement of smoke level of Diesel engines. Diesel exhaust smoke is allowed to flow through a smoke chamber tube of 215mm length and a diameter of 32mm at a regulated pressure of 75mm water column. The light beam from a green LED source of 550nm to 750nm wavelength are focused and allowed to travel through the smoke path which is reflected by a plain mirror from the other end to a detector fixed near the light source. Detector senses the intensity of light which depends upon the opacity of the smoke in the tube, since the smoke tends to obstruct the light more when it is more opaque. Finally smoke level is displayed in term of % opacity as well as in terms of light absorption coefficient ('K'). The smoke number can be converted to smoke opacity and vice versa through the table in Appendix E.

Apart from the smoke level, the instrument also displays the "Engine RPM" and the "Engine RPM Meter (Standard for Eco Smoke 100 APP model only) and standard accessories.

The Smoke chamber unit is the housing for smoke chamber, electronic & mechanical assemblies. Connectors are provided at the rear side of Chamber unit for interfacing various measurement modules that includes a serial port also for interfacing equipment with personal computer (optional feature). AC input socket and power switch are provided in the rear side. The smoke inlet is available at the side of the cabinet to pass the smoke

sample for measurement. Figures 94, 95 show how does the smoke chamber unit looks form the front and rear side respectively.



Figure 94. Front side of the smoke meter



Figure 95. Rear side of the smoke meter

Features and Specifications

The smoke meter unit combines a lot of features in addition to various types of specifications such as the technical, electrical and the general specifications and they are as the following:

Features

- 1) Measurement of emission level in Diesel engine by partial flow method using optics based on folded geometry.
- 2) Suitable for free acceleration test.
- 3) Operates on Universal AC input.
- 4) Operates on DC power.
- 5) Measures smoke opacity in ‘%’ Opacity and ‘K’ value.
- 6) Portable, Compact and light weight construction.
- 7) Error messages before & during test.
- 8) Hand held, Menu driven LCD Remote Control Unit with prompts for easy operation.
- 9) Automatic Zero and Span calibration.
- 10) Display and printout of engine RPM and Oil temperature.

Technical Specifications

Table 19 summarizes the technical specifications of the smoke meter

Table 19

Smoke meter's technical specifications

Measurement Parameters	Range	Resolution	Condition
Opacity	0 - 99.9 %	0.10 %	-
K-value	0 - 9.99 m^{-1}	0.01 m^{-1}	-
Linearity	$\pm 0.1 m^{-1}$	-	Std. test condition
Repeatability	$\pm 0.1 m^{-1}$	-	Std. test condition
Zero & Span drift	$\pm 0.1 m^{-1}$	-	-
Response time-Physical	< 0.4 sec.	-	-
Response time-Electrical	< 1 milli. sec.	-	-
Warm up time	< 7 min.	-	25 C & above
Smoke measuring cell length	215 mm	-	-
	400 - 9990		
RPM	RPM	10 RPM	-
Engine Oil temperature	0-150 C	1 C	-

Electrical Specifications

Table 20 summarizes the technical specifications of the smoke meter

Table 20

Smoke meter's electrical specifications

Specification	Range
AC Power supply	100 - 265V AC single phase, 50 / 60 Hz
DC Power supply	12V DC 2V Battery (NA in Eco Smoke 100 APP)
Power consumption	300 Watts for AC and 100 Watts for DC

General Specifications

Table 21 summarizes the technical specifications of the smoke meter

Table 21

Smoke meter's general specifications

Specification	Range
Operating temperature	100 - 265V AC single phase, 50 / 60 Hz
Machine Dimension- Unpacked	12V DC 2V Battery (NA in Eco Smoke 100 APP)
Machine Dimension - Packed	300 Watts for AC and 100 Watts for DC
Machine Weight - Unpacked	10.5 Kg

Principle of Operation

Eco Smoke 100 depends on the principle of absorbing light, which is a characteristic parameter of the level of smoke exist in an exhaust smoke test sample coming from a Diesel engine. A Green (Source) driven by a pulsating steady current source, emanates a light beam having the peak spectral intensity between 550-570nm wave length. The Detector is designed to have a spectral response from between 350nm to 1100nm with a peak spectral response that has a value of 850nm. IN order to eliminate noise signal, the LED is chopped. That chopped signal is detected by the Detector, with reasonable conditioning circuitry.

The light beam passes from one end of the smoke chamber, gets reflected by a mirror to cover an optical path of 430mm and reaches the Photo diode (Detector) which continuously senses the intensity of light incident on it, and converts it into an electrical signal. This signal is further processed by signal conditioning circuit; the output signal is given to a Microcontroller finally to have digitized readout. A UART communication is utilized for serial communication of data transfer to the host Display unit. The final output is given as % Opacity (N) and light absorption co-efficient (K) in 1/m on the host Display unit where they can be then converted from one to another (Appendix D).

The smoke meter is of partial flow type and the smoke is sampled in a Smoke chamber tube, which has an effective optical path length of 430mm (215mm x 2). This chamber is provided with Heaters to maintain the its temperature of 75C to ensure that condensation does not take place in Chamber and also eliminates the Zero drift. To allow free flow of Smoke through the Smoke chamber tube, Fans are used to create a venture effect at the

ends of the tube, while the smoke is sampled from the center of the tube. These fans are also used as air curtains to keep the lenses clean.

A pressure regulator at the Smoke inlet regulates the Smoke pressure in the Chamber automatically to within 75mm of water column.

5) Speed Tachometer

Description of the device

Using the testo 465 tachometer, a non-contact rpm measurement can be performed using only one hand. This gives the tachometer a great suitability for use in the lab, for instance, a measurement of a rotating part such as a shaft or fan. Figure 96 demonstrates the speed tachometer that is used to detect the engine crank shaft speed



Figure 96. Testo 465 speed tachometer used to detect engine speed

The tachometer also gives the operator these following options:

- 1) Save the minimum / maximum and the mean value in addition to the last measured value.
- 2) The scope of the device is shipped with a protective Soft Case.
- 3) The scope also includes a reflective marker as well as a transport case.
- 4) The operator has a free measurement distance that reaches up to 600 mm.

Features and Specifications

Technical Specifications

Table 22 summarizes the technical specifications of the speed Tachometer

Table 22

Technical specifications of the speed tachometer

Parameter	Value
Measuring range	1 to 99999rpm
Accuracy	± 0.02 % of mv
Resolution	0.1 rpm (100 to 999.9 rpm) , 1 rpm (10000 to 99999 rpm)

General Specifications

Table 23 gives general specifications of the used speed Tachometer

Table 23

General specifications of the used speed tachometer

Parameter	Type
Display type	LCD
Display size	one-line

Principle of Operation

The process of measuring the engine rpm is simple and can be initiated by sticking a reflective marker on the engine crankshaft, after that the red beam of the tachometer is directed towards this reflective rotating paper and the measurement appears on the screen.

The operator has a freedom to set far from the engine crankshaft up to a distance of 600 mm when using the tachometer.

APPENDIX C: CALIBRATION CERTIFICATES

OMEGADYNE INC.
An Affiliate of Omega Engineering, Inc.

PRESSURE TRANSDUCER
FINAL CALIBRATION

0.00 - 10.00 BAR
Excitation 28.000 Vdc

Job: Serial: 204832
 Model: PXM01MDO-010BARGST Tested By: CHRIS
 Date: 9/20/2013 Temperature Range: +16 to +71 C
 Calibrated: 0.00 - 10.00 BAR Specfile: PXM01-5T

Pressure BAR	Unit Data Vdc
0.00	0.005
5.00	2.504
10.00	5.003
5.00	2.506
0.00	0.006

Balance 0.005 Vdc
 Sensitivity 4.998 Vdc
 80% Shunt 4.011 Vdc Change at 0.00 BAR (-INPUT to -OUTPUT)

ELECTRICAL LEAKAGE: PASS
 PRESSURE CONNECTION/FITTING: G 1/4 Female
 ELECTRICAL WIRING/CONNECTOR: Green = + Signal
 White = - Signal
 Black = - Excitation
 Red = + Excitation
 Blue & Orange = Shunt

This Calibration was performed using Instruments and Standards that are traceable to the United States National Institute of Standards Technology.

S/N	Description	Range	Reference	Cal Cert
0076/90-03	AUTO 1000 PSI DRUCK	0 - 10.00 BAR	C-2501	C-2501
US37034809	AT34970 DMM	Unit Under Test	C-3002	C-3002

Q.A. Representative : *Chris Ding* Date: 9/20/2013

This transducer is tested to & meets published specifications. After final calibration our products are stored in a controlled stock room & considered in bonded storage. Depending on environment & severity of use factory calibration is recommended every one to three years after initial service installation date.

Omegadyne Inc., 149 Stelzer Court, Sunbury, OH 43074 (740) 965-9340
 http://www.omegadyne.com email: info@omegadyne.com (800) USA-DYNE

Figure 97. Calibration certificate for the pressure transducer

Infrared Industries, Inc.

Certificate of Compliance

Customer: Handasa Spot

Calibration Date: 6/21/2016
Calibrated By: W.Hinz

Model: HM5000
 Hexane 0-10,000ppm
 CO 0-10%
 CO2 0-20%
 NOX 0-5000ppm

PO:
IRI SO#: 28941

Serial No. 4688

Gas	Conc. (ppm)	Display	% Error (Absolute)
HC	1638	1634.00	0.04%
HC	482	474.00	0.08%
HC	102	97.00	0.05%

Gas	Conc. (%)	Display	% Error (Absolute)
CO	9.01	9.00	0.10%
CO	2.41	2.37	0.40%
CO	0.50	0.48	0.20%

Gas	Conc. (%)	Display	% Error (Absolute)
CO2	16.06	16.05	0.05%
CO2	3.60	3.75	0.75%
CO2	6.00	6.10	0.50%

Gas	Conc. (ppm)	Display	% Error (Absolute)
NOX	3096	3083	0.26%
NOX	906	882	0.48%
NOX	298	274	0.48%

Span High Cal
 Leak Test Pass
 Bag Test Pass

Tach	
Pulse	RPM
5.00	600.0
10.00	1200.0
25.00	2990.0

INFRARED INDUSTRIES, INC. CERTIFIES THE ABOVE LISTED INSTRUMENT HAS BEEN MANUFACTURED, INSPECTED, AND CALIBRATED TO MEET OR EXCEED ALL NEW PRODUCT SPECIFICATIONS. MEASUREMENTS ARE DERIVED FROM ACCEPTED VALUES OF NATURAL PHYSICAL CONSTANTS (SPECTRA / GAS ABSORPTION) AND NBS TRACEABLE CALIBRATED TEST EQUIPMENT WHERE APPLICABLE.

Figure 98. Calibration certificate for the gas analyzer



1600 Washington Blvd
 Grants Pass, OR 97526
 (541) 471-7111
 (541) 471-7116 (Fax)
 Service@metone.com

Met One
 Instruments

Calibration Certificate

The calibration results on this report certify that this instrument complies with the product specifications at the time of calibration. Calibration was performed according to accepted industry methods using equipment, procedures, and standards that are traceable to NIST and ASTM and JIS.

Recommended calibration interval is 12 months from the first day of use.

Instrument Model# Aerocet-531S Instrument Serial# W13544
 Date of Calibration 3/3/2017 Sensor # 16132
Darleen Best ^{AT} Calibration Technician [Signature] Quality Check

Temperature 23 °C Relative Humidity 30 %

Test Procedure: AEROCET-531S-6100

PSL Size (µm)	Test Results	Test Spec.	Lot# NIST	Expiration
0.3	Pass	± 10%	45164	11/30/2018
0.5	Pass	± 10%	43335	06/30/2017
1.0	Pass	± 10%	169240	5/31/2019
2.5	Pass	± 10%	43195	4/30/2017
5.0	Pass	± 10%	44148	02/28/2018
10.0	Pass	± 10%	43497	07/31/2017

Standards	Model	SN	Cal Due
Particle Counter	GT-526	M1760	3/28/2017
FLOWMETER	DC-L	537	4/18/2017
DMM	189 Multimeter	94060816	8/16/2017
RH/Temp Sensor	083E-1-35	R17149	7/25/2017

This calibration certificate shall not be reproduced except in full, without the written approval of Met One Instruments Inc.

Figure 99. Calibration certificate for the particulate meter

APPENDIX D: GTL MSDS

Shell GTL Distillate

Version 1.5

Effective Date 03.11.2011

Safety Data Sheet

heated above the flash point. Do not puncture, cut or weld uncleaned drums. Do not pollute the soil, water or environment with the waste container. Comply with any local recovery or waste disposal regulations.

Local Legislation : Disposal should be in accordance with applicable regional, national, and local laws and regulations. Local regulations may be more stringent than regional or national requirements and must be complied with.

14. TRANSPORT INFORMATION

Land (as per ADR classification): Regulated

Class : 3
Packing group : III
Hazard identification no. : 30
UN No. : 1202
Danger label (primary risk) : 3
Proper shipping name : GAS OIL

IMDG

This material is not classified as dangerous under IMDG regulations.

IATA (Country variations may apply)

This material is either not classified as dangerous under IATA regulations or needs to follow country specific requirements.

Additional Information : MARPOL Annex 1 rules apply for bulk shipments by sea. This product is being carried under the scope of MARPOL Annex I.

Special Precautions: Refer to Chapter 7, Handling & Storage, for special precautions which a user needs to be aware of or needs to comply with in connection with transport.

15. REGULATORY INFORMATION

The regulatory information is not intended to be comprehensive. Other regulations may apply to this material.

Classification triggering components : Contains distillates (Fischer Tropsch), C8 - C26 branched and linear alkanes.

16. OTHER INFORMATION

Additional Information : This document contains important information to ensure the safe storage, handling and use of this product. The information in this document should be brought to the attention of the

Print Date 03.11.2011

10/11

000000021991
MSDS_IMO

Safety Data Sheet

- Developmental Toxicity**
Specific target organ toxicity - single exposure : High concentrations may cause central nervous system depression resulting in headaches, dizziness and nausea; continued inhalation may result in unconsciousness and/or death.
- Specific target organ toxicity - repeated exposure** : Not expected to be a hazard.

12. ECOLOGICAL INFORMATION

- Basis for Assessment** : Information given is based on product data, a knowledge of the components and the ecotoxicology of similar products.
- Acute Toxicity** : Expected to be practically non toxic: LL/EL/IL50 > 100 mg/l (to aquatic organisms) (LL/EL50 expressed as the nominal amount of product required to prepare aqueous test extract).
- Mobility** : Floats on water. Partly evaporates from water or soil surfaces, but a significant proportion will remain after one day. Large volumes may penetrate soil and could contaminate groundwater.
- Persistence/degradability** : Not Persistent per IMO criteria. International Oil Pollution Compensation (IOPC) Fund definition: "A non-persistent oil is oil, which, at the time of shipment, consists of hydrocarbon fractions, (a) at least 50% of which, by volume, distills at a temperature of 340°C (645°F) and (b) at least 95% of which, by volume, distills at a temperature of 370°C (700°F) when tested by the ASTM Method D-86/78 or any subsequent revision thereof." Readily biodegradable.
- Bioaccumulative Potential** : Contains constituents with the potential to bioaccumulate.
- Other Adverse Effects** : Films formed on water may affect oxygen transfer and damage organisms.

13. DISPOSAL CONSIDERATIONS

- Material Disposal** : Recover or recycle if possible. It is the responsibility of the waste generator to determine the toxicity and physical properties of the material generated to determine the proper waste classification and disposal methods in compliance with applicable regulations. Do not dispose into the environment, in drains or in water courses. Do not dispose of tank water bottoms by allowing them to drain into the ground. This will result in soil and groundwater contamination. Waste arising from a spillage or tank cleaning should be disposed of in accordance with prevailing regulations, preferably to a recognised collector or contractor. The competence of the collector or contractor should be established beforehand.
- Container Disposal** : Send to drum recoverer or metal reclaimer. Drain container thoroughly. After draining, vent in a safe place away from sparks and fire. Residues may cause an explosion hazard if

Safety Data Sheet

Kinematic viscosity	: 1.8 - 4.5 mm ² /s at 40 °C / 104 °F
Vapour density (air=1)	: > 5
Evaporation rate (nBuAc=1)	: Data not available
Flammability	: Combustible liquid.

10. STABILITY AND REACTIVITY

Chemical Stability	: Stable under normal conditions of use.
Possibility of Hazardous Reactions	: Data not available
Conditions to Avoid	: Avoid heat, sparks, open flames and other ignition sources.
Incompatible Materials	: Strong oxidising agents.
Hazardous Decomposition Products	: Hazardous decomposition products are not expected to form during normal storage. Thermal decomposition is highly dependent on conditions. A complex mixture of airborne solids, liquids and gases, including carbon monoxide, carbon dioxide and other organic compounds will be evolved when this material undergoes combustion or thermal or oxidative degradation.

11. TOXICOLOGICAL INFORMATION

Basis for Assessment	: Information given is based on product data, a knowledge of the components and the toxicology of similar products.
Likely Routes of Exposure	: Inhalation is the primary route of exposure although absorption may occur through skin contact or following accidental ingestion.
Acute Oral Toxicity	: Low toxicity: LD50 > 5000 mg/kg , Rat
Acute Dermal Toxicity	: Expected to be of low toxicity: LD50 >2000 mg/kg , Rabbit
Acute Inhalation Toxicity	: Expected to be of low toxicity: LC50 >5 mg/l , 4 h, Rat
Skin Corrosion/Irritation	: Expected to be slightly irritating. Prolonged/repeated contact may cause defatting of the skin which can lead to dermatitis.
Serious Eye Damage/Irritation	: Expected to be slightly irritating.
Respiratory Irritation	: Expected to be slightly irritating.
Respiratory or Skin Sensitisation	: Not expected to be a sensitiser.
Aspiration Hazard	: Aspiration into the lungs when swallowed or vomited may cause chemical pneumonitis which can be fatal.
Germ Cell Mutagenicity	: Not mutagenic.
Carcinogenicity	: Not a carcinogen.
Reproductive and	: Not a developmental toxicant. Does not impair fertility.

Safety Data Sheet

	appropriate positive pressure breathing apparatus. Where air-filtering respirators are suitable, select an appropriate combination of mask and filter. All respiratory protection equipment and use must be in accordance with local regulations.
Hand Protection	: Personal hygiene is a key element of effective hand care. Gloves must only be worn on clean hands. After using gloves, hands should be washed and dried thoroughly. Application of a non-perfumed moisturizer is recommended. Suitability and durability of a glove is dependent on usage, e.g. frequency and duration of contact, chemical resistance of glove material, glove thickness, dexterity. Always seek advice from glove suppliers. Contaminated gloves should be replaced. Select gloves tested to a relevant standard (e.g. Europe EN374, US F739). When prolonged or frequent repeated contact occurs, Nitrile gloves may be suitable. (Breakthrough time of > 240 minutes.) For incidental contact/splash protection Neoprene, PVC gloves may be suitable.
Eye Protection	: Chemical splash goggles (chemical monogoggles).
Protective Clothing	: Chemical resistant gloves/gauntlets, boots, and apron (where risk of splashing).
Thermal Hazards	: Not applicable.
Monitoring Methods	: Monitoring of the concentration of substances in the breathing zone of workers or in the general workplace may be required to confirm compliance with an OEL and adequacy of exposure controls. For some substances biological monitoring may also be appropriate.
Environmental Exposure Controls	: Local guidelines on emission limits for volatile substances must be observed for the discharge of exhaust air containing vapour.

9. PHYSICAL AND CHEMICAL PROPERTIES

Appearance	: Colourless. Liquid.
Odour	: Data not available
Odour threshold	: Data not available
pH	: Data not available
Initial Boiling Point and Boiling Range	: 160 - 360 °C / 320 - 680 °F
	:
Flash point	: > 75 °C / 167 °F
Upper / lower Flammability or Explosion limits	: 0.5 - 5.0 %(V)
Auto-ignition temperature	: > 215 °C / 419 °F
Vapour pressure	: < 0.1 kPa at 25 °C / 77 °F
Relative Density	: 0.77
Density	: Typical 0.77 g/cm ³ at 15 °C / 59 °F
Water solubility	: Negligible.
Solubility in other solvents	: Data not available
n-octanol/water partition coefficient (log Pow)	: > 6.5
Dynamic viscosity	: Data not available

Safety Data Sheet

- amine-adduct cured epoxy paint. For seals and gaskets use: graphite, PTFE, Viton A, Viton B.
- Unsuitable Materials** : Some synthetic materials may be unsuitable for containers or container linings depending on the material specification and intended use. Examples of materials to avoid are: natural rubber (NR), nitrile rubber (NBR), ethylene propylene rubber (EPDM), polymethyl methacrylate (PMMA), polystyrene, polyvinyl chloride (PVC), polyisobutylene. However, some may be suitable for glove materials.
- Container Advice** : Containers, even those that have been emptied, can contain explosive vapours. Do not cut, drill, grind, weld or perform similar operations on or near containers.
- Other Advice** : Ensure that all local regulations regarding handling and storage facilities are followed.

8. EXPOSURE CONTROLS/PERSONAL PROTECTION

If the American Conference of Governmental Industrial Hygienists (ACGIH) value is provided on this document, it is provided for information only.

Occupational Exposure Limits

None established.

- Additional Information** : In the absence of a national exposure limit, the American Conference of Governmental Industrial Hygienists (ACGIH) recommends the following values for Diesel Fuel: TWA - 100 mg/m³ Critical effects based on Skin and Irritation.

Biological Exposure Index (BEI) - See reference for full details

No biological limit allocated.

- Appropriate Engineering Controls** : The level of protection and types of controls necessary will vary depending upon potential exposure conditions. Select controls based on a risk assessment of local circumstances. Appropriate measures include: Use sealed systems as far as possible. Adequate ventilation to control airborne concentrations below the exposure guidelines/limits. Local exhaust ventilation is recommended. Eye washes and showers for emergency use.
- Individual Protection Measures** : Personal protective equipment (PPE) should meet recommended national standards. Check with PPE suppliers.
- Respiratory Protection** : If engineering controls do not maintain airborne concentrations to a level which is adequate to protect worker health, select respiratory protection equipment suitable for the specific conditions of use and meeting relevant legislation. Check with respiratory protective equipment suppliers. Where air-filtering respirators are unsuitable (e.g. airborne concentrations are high, risk of oxygen deficiency, confined space) use

Safety Data Sheet

Chapter 8 of this Material Safety Data Sheet. Use the information in this data sheet as input to a risk assessment of local circumstances to help determine appropriate controls for safe handling, storage and disposal of this material. Air-dry contaminated clothing in a well-ventilated area before laundering. Properly dispose of any contaminated rags or cleaning materials in order to prevent fires. Prevent spillages. Use local exhaust ventilation if there is risk of inhalation of vapours, mists or aerosols. Never siphon by mouth. Contaminated leather articles including shoes cannot be decontaminated and should be destroyed to prevent reuse. For comprehensive advice on handling, product transfer, storage and tank cleaning refer to the product supplier. Maintenance and Fuelling Activities - Avoid inhalation of vapours and contact with skin.

Precautions for Safe Handling

: Avoid inhaling vapour and/or mists. Avoid prolonged or repeated contact with skin. When using do not eat or drink. Extinguish any naked flames. Do not smoke. Remove ignition sources. Avoid sparks. Earth all equipment. Electrostatic charges may be generated during pumping. Electrostatic discharge may cause fire. Not expected to be a health hazard when used under normal conditions. The vapour is heavier than air, spreads along the ground and distant ignition is possible.

Conditions for Safe Storage

: Drum and small container storage: Drums should be stacked to a maximum of 3 high. Use properly labelled and closeable containers. Tank storage: Tanks must be specifically designed for use with this product. Bulk storage tanks should be diked (bunded). Locate tanks away from heat and other sources of ignition. Must be stored in a diked (bunded) well-ventilated area, away from sunlight, ignition sources and other sources of heat. The vapour is heavier than air. Beware of accumulation in pits and confined spaces. Keep in a bunded area with a sealed (low permeability) floor, to provide containment against spillage. Prevent ingress of water.

Product Transfer

: Avoid splash filling. Wait 2 minutes after tank filling (for tanks such as those on road tanker vehicles) before opening hatches or manholes. Wait 30 minutes after tank filling (for large storage tanks) before opening hatches or manholes. Keep containers closed when not in use. Do not use compressed air for filling, discharging or handling. Contamination resulting from product transfer may give rise to light hydrocarbon vapour in the headspace of tanks that have previously contained gasoline. This vapour may explode if there is a source of ignition. Partly filled containers present a greater hazard than those that are full, therefore handling, transfer and sampling activities need special care.

Recommended Materials

: For containers, or container linings use mild steel, stainless steel. Aluminium may also be used for applications where it does not present an unnecessary fire hazard. Examples of suitable materials are: high density polyethylene (HDPE) and Viton (FKM), which have been specifically tested for compatibility with this product. For container linings, use

Safety Data Sheet

- Protective Equipment & Precautions for Fire Fighters** : Wear full protective clothing and self-contained breathing apparatus.
- Additional Advice** : Keep adjacent containers cool by spraying with water.

6. ACCIDENTAL RELEASE MEASURES

Avoid contact with spilled or released material. For guidance on selection of personal protective equipment see Chapter 8 of this Material Safety Data Sheet. See Chapter 13 for information on disposal. Observe the relevant local and international regulations. Evacuate the area of all non-essential personnel. Ventilate contaminated area thoroughly.

- Personal Precautions, Protective Equipment and Emergency Procedures** : Do not breathe fumes, vapour. Do not operate electrical equipment.
- Environmental Precautions** : Shut off leaks, if possible without personal risks. Remove all possible sources of ignition in the surrounding area. Use appropriate containment (of product and fire fighting water) to avoid environmental contamination. Prevent from spreading or entering drains, ditches or rivers by using sand, earth, or other appropriate barriers. Attempt to disperse the vapour or to direct its flow to a safe location for example by using fog sprays. Take precautionary measures against static discharge. Ensure electrical continuity by bonding and grounding (earthing) all equipment.
- Methods and Material for Containment and Clean Up** : For small liquid spills (< 1 drum), transfer by mechanical means to a labelled, sealable container for product recovery or safe disposal. Allow residues to evaporate or soak up with an appropriate absorbent material and dispose of safely. Remove contaminated soil and dispose of safely. For large liquid spills (> 1 drum), transfer by mechanical means such as vacuum truck to a salvage tank for recovery or safe disposal. Do not flush away residues with water. Retain as contaminated waste. Allow residues to evaporate or soak up with an appropriate absorbent material and dispose of safely. Remove contaminated soil and dispose of safely. Shovel into a suitable clearly marked container for disposal or reclamation in accordance with local regulations.
- Additional Advice** : Notify authorities if any exposure to the general public or the environment occurs or is likely to occur. Local authorities should be advised if significant spillages cannot be contained. Maritime spillages should be dealt with using a Shipboard Oil Pollution Emergency Plan (SOPEP), as required by MARPOL Annex 1 Regulation 26.

7. HANDLING AND STORAGE

- General Precautions** : Avoid breathing vapours or contact with material. Only use in well ventilated areas. Wash thoroughly after handling. For guidance on selection of personal protective equipment see

Safety Data Sheet

Chemical Identity	CAS	Conc.
Distillates (Fischer-Tropsch) C8-26 - Branched and Linear	848301-67-7	100.00 %

4. FIRST AID MEASURES

Inhalation	: Remove to fresh air. If rapid recovery does not occur, transport to nearest medical facility for additional treatment.
Skin Contact	: Remove contaminated clothing. Immediately flush skin with large amounts of water for at least 15 minutes, and follow by washing with soap and water if available. If redness, swelling, pain and/or blisters occur, transport to the nearest medical facility for additional treatment.
Eye Contact	: Flush eye with copious quantities of water. If persistent irritation occurs, obtain medical attention.
Ingestion	: If swallowed, do not induce vomiting; transport to nearest medical facility for additional treatment. If vomiting occurs spontaneously, keep head below hips to prevent aspiration. If vomiting occurs spontaneously, keep head below hips to prevent aspiration. If any of the following delayed signs and symptoms appear within the next 6 hours, transport to the nearest medical facility: fever greater than 101 °F (38.3°C), shortness of breath, chest congestion or continued coughing or wheezing.
Most Important Symptoms/Effects, Acute & Delayed	: If material enters lungs, signs and symptoms may include coughing, choking, wheezing, difficulty in breathing, chest congestion, shortness of breath, and/or fever. The onset of respiratory symptoms may be delayed for several hours after exposure. Defatting dermatitis signs and symptoms may include a burning sensation and/or a dried/cracked appearance.
Immediate medical attention, special treatment	: Treat symptomatically.

5. FIRE FIGHTING MEASURES

Clear fire area of all non-emergency personnel.

Specific hazards arising from Chemicals	: Hazardous combustion products may include: A complex mixture of airborne solid and liquid particulates and gases (smoke). Carbon monoxide. Unidentified organic and inorganic compounds. Carbon monoxide may be evolved if incomplete combustion occurs. Will float and can be reignited on surface water. Flammable vapours may be present even at temperatures below the flash point.
Suitable Extinguishing Media	: Foam, water spray or fog. Dry chemical powder, carbon dioxide, sand or earth may be used for small fires only.
Unsuitable Extinguishing Media	: Do not use water in a jet.

Safety Data Sheet

Storage	: P403+P235: Store in a well-ventilated place. Keep cool. P405: Store locked up.
Disposal	: P501: Dispose of contents and container to appropriate waste site or reclaimer in accordance with local and national regulations.
Other Hazards which do not result in classification	: Slightly irritating to respiratory system. Breathing of high vapour concentrations may cause central nervous system (CNS) depression resulting in dizziness, light-headedness, headache and nausea. May cause slight irritation to skin. Repeated exposure may cause skin dryness or cracking. May ignite on surfaces at temperatures above auto-ignition temperature. Vapour in the headspace of tanks and containers may ignite and explode at temperatures exceeding auto-ignition temperature, where vapour concentrations are within the flammability range. Electrostatic charges may be generated during pumping. Electrostatic discharge may cause fire.
Additional Information	: This product is intended for use in closed systems only.

3. COMPOSITION/INFORMATION ON INGREDIENTS

Preparation Description	: A complex combination of hydrocarbons obtained from a feedstock derived from the catalytic hydrogenation of carbon monoxide (the Fischer - Tropsch Process), optionally followed by one or more of the following processes: hydrotreatment, hydroisomerisation, hydrocracking. It consists predominantly of branched and linear aliphatic hydrocarbons having carbon numbers in the range of C8 to C26 and boiling in the range of approximately 120C to 380C (248F to 716F).
Synonyms	: Shell GTL Fuel Shell GTL Gasoil Heavy Normal Paraffin Shell GTL Sarasol 100 Hydrocarbon Solvent 108 Liquid Paraffin 100 Mixed Paraffin 360 Paraffin Oil Paraffin Oil W Paraffin Oil (Liquid Paraffin) C12 - C22 Shell GTL Saraline 200 Shell GTL Saraline 98V Shell GTL Saraline 185V Liquid Paraffin 85 Shell GTL Sarasol 85 ISO N-Paraffins Shell GTL Light Fuel Liquid Paraffins - H Paraffin Oil - H


Hazardous Components

Safety Data Sheet

1. IDENTIFICATION OF THE SUBSTANCE/PREPARATION AND COMPANY/UNDERTAKING

Material Name	: Shell GTL Distillate
Other Names / Synonyms	: Shell GTL Fuel
Recommended Use / Restrictions of Use	: Alkanes, middle distillate-range, hydrotreated. / Automotive gasoil blending component.
Supplier	: Shell Trading Rotterdam B.V. Weena 70 3012 CM Rotterdam Netherlands
Contact Telephone	: +31 10 441 5000
Emergency Telephone Number	: +1 703-527-3887
MARPOL Annex I Category	: Gas Oils (including ship's bunkers)
Description on Bill of Lading (B/L)/Bunker delivery note/Shipping document	: Distillates (Annex 1, Appendix 1 Name)
Other Information	: See Section 14 for transportation information related to the Bill of Lading, other shipping documents.

2. HAZARDS IDENTIFICATION

GHS Classification	: FLAMMABLE LIQUIDS, Category 4 ASPIRATION HAZARD, Category 1
GHS Label Elements Symbol(s)	: 
Hazard Statement	: PHYSICAL HAZARDS: H227: Combustible liquid.
Signal Words	: Danger HEALTH HAZARDS: H304: May be fatal if swallowed and enters airways.
GHS Precautionary Statements	
Prevention	: P210: Keep away from heat/sparks/open flames/hot surfaces. - No smoking. P280: Wear protective gloves/protective clothing/eye protection/face protection.
Response	: P301+P310: IF SWALLOWED: Immediately call a POISON CENTER or doctor/physician. P331: Do NOT induce vomiting.

Safety Data Sheet

	person in your organisation responsible for advising on safety matters.
MSDS Version Number	: 1.5
MSDS Effective Date	: 03.11.2011
MSDS Revisions	: A vertical bar () in the left margin indicates an amendment from the previous version.
Uses and Restrictions	: This product must not be used in applications other than those recommended in Section 1, without first seeking the advice of the supplier.
MSDS Distribution	: The information in this document should be made available to all who may handle the product.
Disclaimer	: This information is based on our current knowledge and is intended to describe the product for the purposes of health, safety and environmental requirements only. It should not therefore be construed as guaranteeing any specific property of the product.

APPENDIX E: ABSORPTION "K" TO OPACITY "%"

<i>k</i>	%	<i>k</i>	%	<i>k</i>	%
0	0	3.9	81.31	7.8	96.51
0.1	4.21	4	82.09	7.9	96.65
0.2	8.24	4.1	82.85	8	96.79
0.3	12.1	4.2	83.57	8.1	96.93
0.4	15.8	4.3	84.26	8.2	97.06
0.5	19.35	4.4	84.92	8.3	97.18
0.6	22.74	4.5	85.56	8.4	97.3
0.7	25.99	4.6	86.17	8.5	97.41
0.8	29.11	4.7	86.75	8.6	97.52
0.9	32.09	4.8	87.31	8.7	97.63
1	34.95	4.9	87.84	8.8	97.73
1.1	37.69	5	88.35	8.9	97.82
1.2	40.31	5.1	88.84	9	97.91
1.3	42.82	5.2	89.31	9.1	98
1.4	45.23	5.3	89.76	9.2	98.09
1.5	47.53	5.4	90.19	9.3	98.17
1.6	49.74	5.5	90.61	9.4	98.24
1.7	51.86	5.6	91	9.5	98.32
1.8	53.88	5.7	91.38	9.6	98.39
1.9	55.82	5.8	91.74	9.7	98.46
2	57.68	5.9	92.09	9.8	98.52
2.1	59.46	6	92.42	9.9	98.58
2.2	61.17	6.1	92.74	9.91	98.59
2.3	62.81	6.2	93.05	9.92	98.59
2.4	64.37	6.3	93.34	9.93	98.6
2.5	65.87	6.4	93.62	9.94	98.61
2.6	67.31	6.5	93.89	9.95	98.61
2.7	68.68	6.6	94.15	9.96	98.62
2.8	70	6.7	94.39	9.97	98.62
2.9	71.26	6.8	94.63	9.98	98.63
3	72.47	6.9	94.85	9.99	98.64
3.1	73.63	7	95.07		
3.2	74.74	7.1	95.28		
3.3	75.8	7.2	95.48		
3.4	76.82	7.3	95.67		
3.5	77.8	7.4	95.85		
3.6	78.73	7.5	96.02		
3.7	78.83	7.6	96.19		
3.8	80.49	7.7	96.35		

APPENDIX F: SAFETY RULES

There were many precautions and instructions that had to be followed during the experimental work in the laboratory and also when using the measuring devices and they were as the following:

- 1) Keep the fuel in metallic containers, accurately locked and clearly labeled
- 2) Put the containers in a ventilated place, so to limit the danger of explosion of the combustible gases
- 3) Never effectuate the filling of the container with the engine warm, to avoid the danger of a fire in the case the fuel drops should come in contact with the engine itself
- 4) Execute the aforementioned operation, and every other operation in which the fuel is concerned , only after having moved away every free flame or non-free flame
- 5) Predispose a fire extinguisher of adequate capacity, and which is suitable for the type of combustible substance being used.
- 6) Always make sure that the engine is equipped with either an opposite pipework to convey to the outside the exhaust gases, or a ventilation system for the suction of those gases.
- 7) Do not use the gas analyzer for testing exhaust emission when the engine is smoking excessively or when it is in obvious need of repair.
- 8) Keep the probe tip openings clean and free of debris for the gas analyzer and for the smoke meter

- 9) Do not place the probe tip in liquids or allow liquids to be drawn into the analyzer or the smoke meter's sampling
- 10) Do not place the probe in an exhaust pipe until the vehicle is at normal operating temperature. This allows the exhaust system time to burn off any residual moisture.
- 11) Perform a leak test periodically, especially after probe changes and filter services, to ensure accurate analysis.
- 12) To comply with anti-tampering laws, always follow the manufacturer's specifications when working on emission control devices.
- 13) Always comply with the governing emission control standards and regulations in your locality when testing exhaust emission levels.
- 14) Insert the test probe fully into the tailpipe when testing exhaust emissions to prevent diluted readings.
- 15) Check the manufacturer's specifications and procedures before testing a vehicle.