# EXPERIMENTAL STUDIES ON THE UTILIZATION OF ACETYLENE AND OXYGENATED BLENDS IN A DUAL FUEL ENGINE

**A Thesis submitted to the Delhi Technological University, Delhi in fulfillment of the requirements for the award of the degree of**

# DOCTOR OF PHILOSOPHY

in

**Mechanical Engineering**

by

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**May 2021**

# DECLARATION

I hereby declare that the thesis entitled **"EXPERIMENTAL STUDIES ON THE UTILIZATION OF ACETYLENE AND OXYGENATED BLENDS IN A DUAL FUEL ENGINE"** is an original work carried out by me under the supervision of Dr. Naveen Kumar, Professor, Department of Mechanical Engineering, Delhi Technological University, Delhi. This thesis has been prepared in conformity with the rules and regulations of the Delhi Technological University, Delhi. The research work reported, and results presented in the thesis have not been submitted either in part or full to any other university or institute for the award of any other degree or diploma.

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Place: New Delhi

# CERTIFICATE

This is to certify that the work embodied in the thesis entitled **"EXPERIMENTAL STUDIES ON THE UTILIZATION OF ACETYLENE AND OXYGENATED BLENDS IN A DUAL FUEL ENGINE"** by **Roshan Raman, (Roll No.-2K16/PhD/ME/59)** in partial fulfillment of requirements for the award of Degree of **DOCTOR OF PHILOSOPHY in Mechanical Engineering**, is an authentic record of student's own work carried by him under my supervision.

This is also certified that this work has not been submitted to any other Institute or University for the award of any other diploma or degree.

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***Dedicated to…***

# ***My Parents***
# ***&***
# ***All Family Members***

***"Your lives of faith, family, and farming sowed the seeds of my future"***

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New Delhi **(Roshan Raman)**

# ABSTRACT


The energy requirement has increased multifold globally with the steep rise in industrialization and socio-economic growth. The various sectors that require energy are industry, transport, agriculture, domestic, and many others. However, the world's transportation depends heavily on petroleum fuels. Diesel engines are highly demanded because of their higher thermal efficiency and versatility. However, these engines emit harmful exhaust emissions such as carbon monoxide (CO), hydrocarbons (HC), oxides of nitrogen (NOx), particulate matter, and smoke while burning liquid diesel. Many engine scientists have recommended to utilize alternative fuels to tackle twin catastrophes i.e., environmental degradation and fossil fuel crisis.

Hence, alternative fuels are getting more prominence as concerns over escalated exhaust emissions as well as replacement of petroleum fuels. Utilizing gaseous fuels is one of the most promising options to mitigate exhaust pollutants as well as to improve conventional engine performance without major modifications. Since they have higher auto-ignition temperature, higher calorific values, and clean-burning affinity, hence are suitable for compression ignition engines under dual fuel mode. In this regard acetylene has attractive combustion properties like hydrogen and the utilization of acetylene is not exhaustively explored in literature. Hence acetylene is utilized as a primary fuel in dual fuel engine to investigate the combustion, performance, and emission characteristics in the present study. In the first stage of the experiment, the optimized flow rate of acetylene is found by examining the engine characteristics. The flow rate of acetylene is varied from 2 LPM to 8 LPM in the step count of 2 LPM. It is noticed that when the acetylene is inducted at 4 LPM, brake thermal

efficiency (BTE) is comparable at 80% load as well as HC, CO emissions are reduced in comparison to pure diesel. However, the NOx level is increased in comparison to neat diesel.

Furthermore, the experimental results are fed as input and targets in the ANN network to predict the accuracy of experimental results. Predicted results show that R values closer to unity and mean square error value approximately 0.001. In the second phase of the experimentation, oxygenated fuels, i.e., diesel/DEE and diesel/n-butanol blends, are utilized in the dual fuel engine at optimized induction rate (4 LPM) of acetylene gas. Again, the performance, combustion, and emission parameters are evaluated in a modified engine. It is observed that BDEE10 shows the highest brake thermal efficiency (BTE), i.e., 1.67% higher, whereas BBU10 shows 1.21% higher BTE compared to baseline diesel. Simultaneously, NOx emission is also reduced to a substantial level, i.e., 22% and 27% lower than neat diesel while utilizing BDEE10 and BBU10 blend under dual fuel mode of combustion. Moreover, HC and CO emissions are reduced at a significant rate while utilizing BDEE10 and BBU10 blend in dual fuel mode. Overall, it can be concluded that the performance of the acetylene fuelled engine can be improved by utilizing oxygenated blends up to 10%.

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# NOMENCLATURE

| | |
|---|---|
| @ | At the rate |
| A | Acetylene |
| ADF2 | Acetylene @ 2 LPM |
| ADF4 | Acetylene @ 4 LPM |
| ADF6 | Acetylene @ 6 LPM |
| ADF8 | Acetylene @ 8 LPM |
| A/F | Air to Fuel |
| ASTM | American Society for Testing and Materials |
| ATDC | After Top Dead Center |
| AVL-437 | AVL-437 Smoke Meter |
| BBU05 | 5% n-Butanol + 95 % Diesel |
| BBU10 | 10% n-Butanol + 90 % Diesel |
| BBU15 | 15% n-Butanol + 85 % Diesel |
| BBU20 | 20% n-Butanol + 80 % Diesel |
| BDEE05 | 5% DEE + 95 % Diesel |
| BDEE10 | 10% DEE + 90 % Diesel |
| BDEE15 | 15% DEE + 85 % Diesel |
| BDEE20 | 20% n-Butanol + 80 % Diesel |
| BDC | Bottom Dead Centre |
| BMEP | Brake Mean Effective Pressure |
| BSEC | Brake Specific Energy Consumption |
| BSFC | Brake Specific Fuel Consumption |
| BTE | Brake Thermal Efficiency |
| BTDC | Before Top Dead Center |
| C | Carbon |
| $^{o}$CA | Degree Crank Angle |
| $CaCO_3$ | Calcium carbonate |
| CC | Cubic centimeter |
| CI | Compression-Ignition |
| CO | Carbon Monoxide |
| CO2 | Carbon Dioxide |
| cSt | Centi Stoke |
| CV | Calorific Value |

| | |
|---|---|
| D100 | Neat Diesel or Baseline |
| DI | Direct Injection |
| DEE | Di-ethyl ether |
| DFE | Dual Fuel Engine |
| DFC | Dual Fuel Combustion |
| EGR | Exhaust Gas Circulation |
| g | gram |
| K | Degree Kelvin |
| kg | Kilogram |
| F/A | Fuel to Air |
| HC | Hydrocarbon |
| HP | Horse Power |
| IC | Internal Combustion |
| ID | Ignition Delay |
| IDI | Indirect Injection |
| IR | InfraRed |
| IT | Injection Timing |
| KVA | Kilo Volt Ampere |
| kW | Kilo Watt |
| kWh | Kilo Watt Hour |
| LHV | Latent heat of vapourization |
| LPM | Liter per minute |
| $\dot{m}$ | Mass flow rate of fuel |
| Min. | Minute |
| Mt | Million Tonnes |
| Mtoe | Million Tonnes of Oil Equivalent |
| n | normal |
| Nos. | Numbers |
| NOx | Oxides of Nitrogen |
| PC | Personal Computer |
| PM | Particulate Matter |
| PP | Pour Point |
| ppm | Parts per million |
| P-θ | Pressure – Crank Angle |
| rpm | Revolutions Per Minute |
| R | Gas Constant |
| T | Temperature |
| TDC | Top dead centre |
| SCR | Selective Catalyst Regenerative |

| | |
|---|---|
| SFC | Specific Fuel Consumption |
| SI | Spark Ignition |
| SOC | Start of Combustion |
| SOI | Start of Injection |
| SOx | Oxides of Sulphur |
| TDC | Top Dead Center |
| UBHC | Unburnt Hydrocarbon |
| V | Instantaneous Cylinder Volume, $m^3$ |

CHAPTER 1

# INTRODUCTION

## Overview

*This chapter sets the background of the research. It begins with the motivation of the present study and introduces the energy scenario of India in detail. Furthermore, the environmental issues are also discussed concerning engine exhaust emissions followed by alternative options to mitigate these harmful pollutants in near future with the aid of alternative fuels and advanced technology. Also, it highlights the working of a dual fuel engine. The structure of the present thesis is outlined at the end of this chapter.*

## 1.1 Motivation for the present work

The growth of any nation is reliant on the sufficient availability of energy resources and advanced technologies. At present, conventional fossil fuels are the main source of energy. Hence, to meet increasing energy demands, researchers always look for better alternatives to fossil fuels. The industrial sector is the major consumer of energy, followed by transportation and agriculture. Diesel engines have a higher compression ratio and superior thermal efficiency, so they have become part and parcel of modern life. Harmful pollutants such as oxides of nitrogen ($NO_x$), particulate matter (PM), sulphur oxide ($SO_x$), carbon dioxide ($CO_2$), smoke, and many others are products of diesel engines. These toxic gases are very injurious to human health and may cause severe lung infections and breathing issues. They are also the prime reasons for environmental degradation, global warming, depletion of the ozone layer, and natural calamities.

Cleaner combustion technology is the need of the hour. In this regard, dual fuel combustion mode can be considered a promising technique to alleviate these harmful pollutants without compromising engine performance. The present work aims to explore dual fuel combustion technique using alternative fuels for reducing exhaust emissions with the improved thermal efficiency of the engine.

**1.2 Energy Scenario**

Multitudinous fuel and energy consumption seem to be an inevitable aspect of rapidly developing countries like India. The industrial revolution and rapid modernization have increased the consumption pattern in various spheres like transport, agriculture, domestic uses, etc. Transportation fuels are chiefly petroleum products that keep the world running at a fast pace. India, a diesel-driven economy, has the highest consumption volume of diesel fuel around 88.2 billion liters in 2020 whereas petrol with a consumption volume of about 37.2 billion liters for the same year. 34% of India's overall energy intake is fulfilled by crude oil, out of which 83% of crude oil is imported from foreign countries, in foreign currencies creating immense economic instability in the country (MOPNG, Annual Report 2019).

Hence the majority of the Indian economy gets affected by the oil price fluctuations in the global market. Essential supply, demand conditions, production costs, political turmoil, and even interest rates altogether influence the price of crude oil in international markets (World Energy Outlook-IEA, 2019). The perturbation of crude oil price per barrel is shown in Figure 1.1 from December 2019 to July 2020. The industrial sector is the highest energy-intensive industry, accounting for around 56% of overall energy consumption. For the period 2011-12 to 2017-18, energy usage per capita was recorded at 2.54% (Energy Statistics, 2019). India's growth rate of energy consumption is estimated to be higher by over 3 percent than all other countries (BP Energy Outlook, 2019).

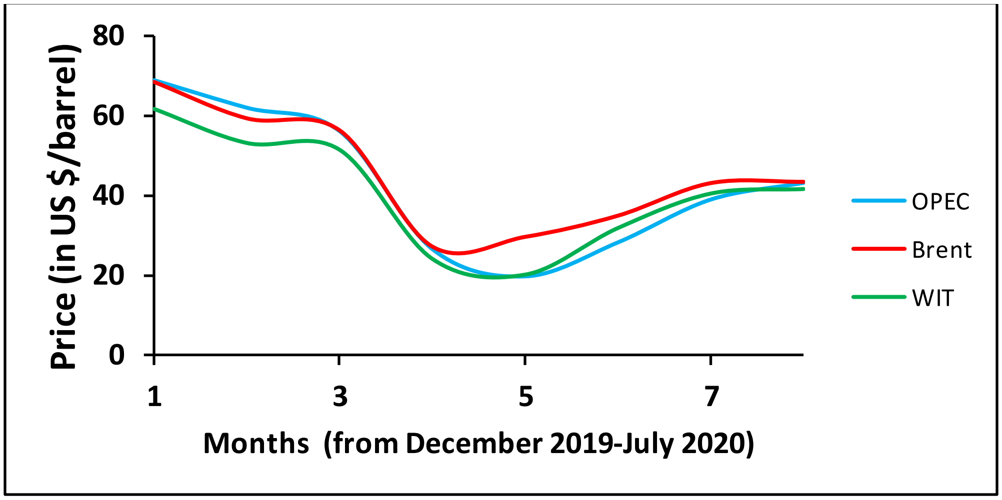


**Figure 1.1:** Daily price of Indian basket crude oil from Dec. 2019 to July 2020(Diesel Net, 2020).

The tremendous demographic expansion and technological advancement over the last decade have improved the living standards in developing countries like India. The mindset of the Indian community has changed, and now no more personal vehicle is treated as luxury goods rather than essential commodities. Moreover, India experienced a 355% rise in car production and a 617% increase in trucks relative to 2010-11 in 2017-18 (World Energy Outlook-IEA, 2019). Most vehicles in India are compression ignition engine because of their versatile application. However, such engines are also contributors to harmful emissions, and there is an urgent need to search for alternatives to petroleum-derived diesel to reduce these harmful emissions in the environment.

### 1.3 Environmental Issues

Over a few decades, researchers have concluded that environmental degradation at an alarming rate is a major concern for the entire world. India emits 2251 million tons of carbon dioxide ($CO_2$) emissions in the year 2018-19, while the world emits 24,785 million tons of $CO_2$ (World Energy Outlook-IEA, 2019). Moreover, the percentage of $CO_2$ emissions alone from petroleum fuels accounts for 28% in the year 2018 (Diesel Net: Engine & Emission Technology, 2019). Figure 1.7 shows the $CO_2$ gas emissions in the years 1990, 2005, and 2018, respectively. It has been noticed that around 45% of $CO_2$ emission is contributed by crude oil in the year 2018. Thus, an increase in the $CO_2$ concentration at a higher rate leads to global warming due to the greenhouse gas (GHG) effect.

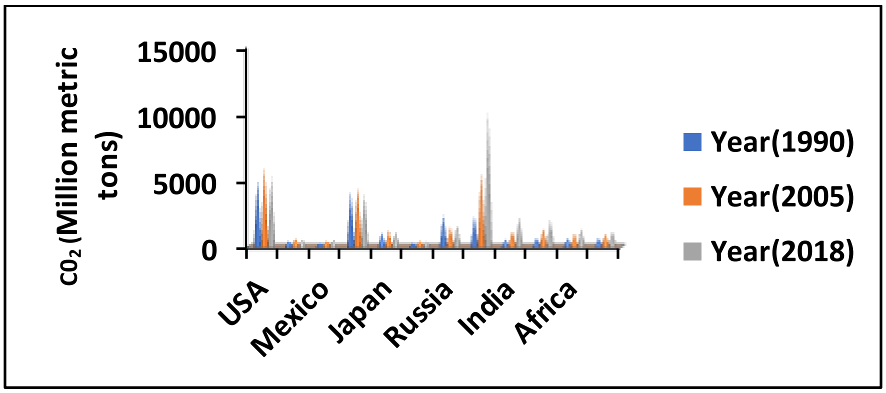


**Figure 1.2:** $CO_2$ emissions country wise (World Energy Outlook-IEA, 2019)

Primarily greenhouse gases are carbon dioxide, methane, and water vapor at a higher level of concentration. In contrast, ozone, nitrous oxide, and fluorinated gases are also responsible for the greenhouse effect to some extent (World Energy Outlook-IEA, 2019). GHG disturb radiation balance by absorbing infrared radiation from the Earth's surface and dissipating it back to the Earth. Eventually, abrupt climate pattern changes lead to ecological imbalance, rise in sea level due to ocean tides, and sometimes calamities (Emissions Gap Report, UN Environment Programme, 2019).

To restrict the vehicular pollution, the government of India along with the ministry of road transport and highways (MORTH) moved a step closer to lowering automobile emissions, by deciding to implement Bharat Stage (BS) VI emission standard from April, 2020. For diesel vehicles, the BSVI emission norms aim to reduce major three pollutants viz. hydrocarbons (HC), particulate matter (PM), and oxides of nitrogen (NOx) by 43%, 68%, and 82% respectively. Due to this extreme legislation, the available engines must be upgraded with suitable hardware modifications in a short interval of time which may create chaos.

Moreover, advanced technology such as lean NOx traps (LNT), selective catalytic reduction (SCR), and diesel particulate filters (DPFs) which are to be fitted for upgrading diesel engines complying with BSVI norms is too expensive. (Diesel Net: Engine & Emission Technology, 2019). Hence looking ahead to the future demands many engine researchers have attempted to search the alternative fuels and advanced technology to mitigate twin crises i.e., depletion of fossil fuels and environmental degradation. It is expected that by utilizing alternative fuels emissions levels can be significantly lowered down and could solve the purpose of the clean and green mission of India.

### 1.4 Alternative Fuels and Advanced Technology

The over-dependency on fossil fuels and environmental issues has stimulated engine scientists and industrial experts to take initiative and to find a better substitute for diesel fuel. Consequently, the fuel should not solely match the performance criteria like diesel rather it must also be readily available, economically viable, and meet the current standard emissions norms.

Thus, sustainable, clean-burning fuels that can be naturally replenished by non-conventional ways are designated as alternative fuels (Agarwal., 2007). The use of alternative fuels is one of the approaches to curb harmful exhaust emissions from diesel engines. Over a few decades, much research has been attempted to reduce petroleum fuel consumption by using alternative fuels in diesel engines. India, being an agricultural country, non-edible vegetable oils could be used as a preferred alternative fuel because of abundant availability and large productivity (Kumar and Agarwal, 2008). Nevertheless, vegetable oil causes improper combustion due to higher viscosity and inferior atomization of fuel.

Conversion of biodiesels from raw vegetable oil is one way to reduce viscosity and improve combustion characteristics. However, the production cost at the initial stage may be too high, and even the performance of the engine might deteriorate because of the lower energy content of biodiesel. Moreover, engine scientists have also observed a higher level of NOx emissions with biodiesel fuelled engines (Dhar et al., 2017; Asharful et al., 2015). Subsequently, numerous engine experts have also explored primary alcohols like methyl alcohol and ethyl alcohol blended with diesel in the conventional engine but due to low boiling point and higher vapour pressure vapour lock is the serious issues with these fuels even though cheaply available.

The relatively high latent heats of methyl and ethyl alcohol cause problems in mixing these alcohols with air and transporting them through the intake manifold of the engine. On the other hand, few engine experts have utilized higher order alcohols like n-butanol and observed emission benefits with n-butanol-diesel engine. Normal butanol has better miscibility with diesel fuel than methanol or ethanol. Moreover, n-butanol has a higher cetane number, higher viscosity, and higher heating value in comparison to primary alcohols. Similarly, few researchers have also explored the potential of Di-ethyl ether as a transportation fuel because of its attractive physicochemical properties such as higher calorific value and higher cetane index in the diesel engine.

Many alternative liquid fuels have been tried and tested at the laboratory level but still difficulty gaining acceptance at the commercial level because they are not available abundantly and generally refrained to be used in the neat form due to inferior results. Now a day's gaseous fuels are getting more popular in comparison to liquid fuels because of clean-burning property. However, the self-ignition temperature of gaseous fuels is very high that makes it difficult to be used directly in CI engines. Hence, it can be utilized with the dual fuel approach (Karim, 2015a). Various gaseous fuels such as biogas, LPG (liquified petroleum gas), LNG (liquified natural gas), CNG (compressed natural gas), hydrogen, and acetylene gas have been used in a modified diesel engine (Banapurmath et al., 2014; Das, 2002; Ashok and Kumar, 2015; Karim, 1983; Wei and Geng, 2016).

Most of the work in the literature is focused on CNG, hydrogen, and LPG engines whereas very little quantum of work are concentrated on acetylene fuelled engine. CNG engines have major challenges of inferior performance and refilling issues whereas hydrogen considered as the cleanest fuel has its drawback regarding higher NOx emissions and poor volumetric efficiency and there is the fear of premature ignition and flashback as major challenges associated with running an engine on hydrogen. Acetylene possesses similar properties to hydrogen and could be used as an alternate fuel in IC engines. It is a colourless gas and smells like garlic (Schobert, 2014; Sharma et al., 2012). The substantial properties such as wide flammability range and higher flame speed render acetylene a viable alternative fuel (Raman and Kumar, 2019a.). The wall wetting problems of diesel engines can be eliminated using acetylene gas as it can easily mix with air (Raman and

Kumar, 2020). However, storage facilities and filling stations are still significant challenges of gaseous fuels (Reed, 2006; Banapurmath et al., 2014).

### 1.5 Dual Fuel Engine

Rudolf Diesel is the one who discovered first an engine working on a dual-fuel principle (R. Diesel, 1901). A dual fuel engine (DFE) or dual fuel combustion (DFC) system is a promising technique to utilize gaseous and liquid fuels in diesel engines. In a DFC system, gaseous fuel is directly injected into the combustion chamber or mixed with fresh air. Then it is inducted into the engine cylinder. However, because of its very high self-ignition temperature, gaseous fuel does not auto-ignite itself. Thus, petroleum diesel-like fuel is injected closer to the compression stroke to start the ignition process. However, liquid pilot fuel is burnt first due to auto-ignition. Thus, flame generated by the ignition of pilot fuel is used as a prime source of ignition of the combustible gaseous-air fuel mixture (Raman and Kumar, 2019). In this way, flame propagates like that of a spark-ignition engine. Consequently, the dual fuel combustion engine combines both CI and SI engine features complicatedly. Dual fuel has fuel adaptability, i.e., can operate on gaseous fuel depending upon gas or liquid fuel availability when gaseous fuel is deficient. The working of dual fuel engines shown in Figure 1.3.

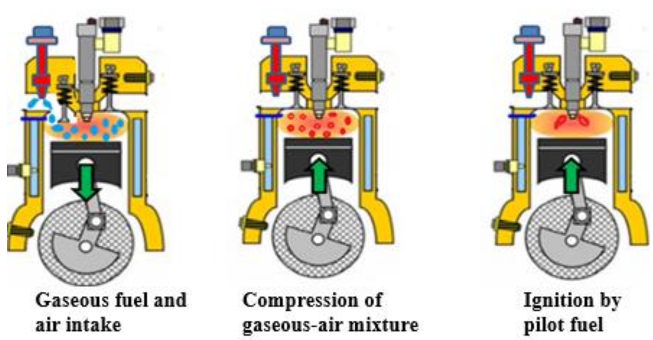


**Figure 1.3:** Working of dual fuel engine (Raman and Kumar, 2020)

## 1.6 Organization of Thesis

The Ph.D. thesis contains five chapters. The chapters are arranged as follows:

**CHAPTER 1: (INTRODUCTION)** – This chapter describes a general and brief outline of the background and subject of the current research. It starts with motivation followed by environmental issues and introduces the dual fuel concept in a diesel engine as the possible option to utilize the alternative liquid and gaseous fuels.

**CHAPTER 2: (LITERATURE REVIEW) -** Chapter 2 includes a detailed literature survey of dual fuel engines (working, advantages, and limitations) with various combinations of gaseous fuels and pilot fuels. The diverse performance, combustion, and emission parameters of dual fuel engines are discussed in detail. The physico-chemical properties of various gaseous fuel and pilot (liquid fuels) are reviewed, and multiple parameters are highlighted to scrutinize the performance characteristics of a dual fuel engine. The utilization of oxygenated fuel has evolved as a promising solution in a modified diesel engine for sustainable development. Furthermore, the importance of artificial intelligence in analyzing the internal combustion engine has also been discussed.

**CHAPTER 3: (EXPERIMENTAL SETUP AND TEST PROCEDURES)** – This chapter explains the details of dual fuel engine test rig, including working principles of various equipment and accessories employed for the engine testing. This chapter also includes the description and specification of measuring instruments for measuring physico-chemical properties of selected fuels/blends and compared with ASTM standard limits. The experimental procedures along-with experimental data calculation uncertainties are also described in detail. Each measurement is recorded thrice, and the average is taken for further analysis. It also explains the methodology for optimizing dual fuel engines' performance parameters using an artificial neural network (ANN) technique.

**CHAPTER 4: (RESULTS AND DISCUSSION)** –This chapter outlines all the findings derived from the experimental studies. It provides a thorough coverage, including results from the performance, combustion, and emission characteristics for different fuel combinations (acetylene and oxygenated blends) in a dual fuel engine. Followed by a detailed discussion and analysis, the main findings of the present study are compared with the existing results reported in the literature. In addition to this predictive analysis, results have been analyzed using the ANN technique in MATLAB 2018.

**CHAPTER 5: (CONCLUSION)** –It summarizes the significant findings and the key outcomes of the present research and propose some recommendations for the future development in dual fuel engine.

**CHAPTER 2**

# LITERATURE REVIEW

## Overview

*The dual fuel combustion technique is one of the most reliable methods to reduce vehicular emissions with a clean combustion mechanism. This chapter contains a literature analysis of different available gaseous fuels for the dual fuel engine operation. The chapter discusses in detail the effects of engine design and operating parameters on the efficiency, combustion, and emission characteristics of dual fuel diesel engines working on several gaseous as well as pilot fuels. The various works on simulation using the artificial neural network has also been highlighted for dual fuel engine. Finally, the predetermined objectives have been derived from the literature research gap.*

## 2.1 Compression Ignition Engine

The development of the internal combustion (IC) engine was initiated during the last decades of the 19th century. After that IC engines have been utilized as prime movers to propel and progress human civilization. Because of its high power-to-weight ratio, the IC engines can be found in several applications, such as transportation of people and goods via land and waterways, power generation, and industrial applications (Pulkrabek, 2004). In India, compression ignition engines have been widely accepted due to their excellent performance and economy. These engines are commonly preferable in the agricultural sector to irrigate the farms and mechanized farm machinery.

Moreover, they are also used for transportation and power generation purpose. Furthermore, these engines are operated on petroleum diesel which has an adverse effect on the environment due to incomplete combustion. Many researchers have raised the availability issues of petroleum fuels, as stated vast of the resources have been exhausted and it is expected within a few years scarcity of fossil fuels will be felt. India ranks 3rd in the world for petroleum oil consumption, accounting for about 4.6% of the world's total consumption of 97,103,871 barrels per day. Hence, the security of fuel availability as an energy carrier should be maintained for the long term.

## 2.2 Combustion in CI Engine

In a compression ignition (CI) or diesel engine, a highly pressurized liquid fuel is introduced in the combustion chamber late in the compression stroke near the top dead centre. Then, it mixes with the high pressure and high-temperature air charge which auto-ignites. The diesel combustion process is a complicated turbulent, unsteady, highly heterogeneous, and three-dimensional event. The combustion process starts with the start of the injection (SOI) event, which causes the fuel to absorb energy from the surroundings, and consequently, a sharp decrease in the energy release is observed. This phenomenon is known as the evaporation phase, where the fuel droplets change to the vapor phase. The fuel continues to mix with the charge i.e., air and trapped exhaust gases, and certain pre-reactions, resulting in cool flames, occur before the start of combustion (SOC). Due to the injected fuel characteristics, the main combustion event starts shortly after SOI in those regions where temperature and fuel-air equivalence ratio allow ignition.

The time between SOI and SOC is typically referred to as ignition delay. The primary stage of combustion begins with the premixed combustion phase. It is characterized by a rapid rise in heat release rate, and it is significantly controlled by chemical kinetics and, consequently, ignition delay. This event increases in pressure and temperature in the combustion chamber, which triggers the ignition in other regions. It is essential to highlight that this phase and the fuel injection event can co-occur (Heywood, 2018).

The next stage is named spray driven combustion phase. A diffusion flame is established while the fuel-injected is being atomized, vaporized, mixed, and ignited. The mixing rate limits the heat release rate, and this stage finalizes at the end of the injection (EOI). The last step is generally referred to as late mixing-controlled combustion, and the remaining unburned fuel ignites while the heat release rate gradually decreases. This heat release model provides global information about the combustion process, but it does not explain the diesel jet's fundamental functions, determining the combustion characteristics. In recent years, diagnostics on optical engines have enabled a real understanding of the diesel combustion process (John et al., 2007).

### 2.3 Diesel Engine- Emissions

A combusting diesel jet is made of different fuel-lean and rich zones that burn simultaneously and where different types of soot or nitrogen oxide are formed. The process starts at the injector of the nozzle, where liquid diesel fuel penetrates the combustion chamber and progressively mixes and vaporizes within the ambient charge. Hence, a rich vapor fuel-air mixture inside the jet (soot formation area) is surrounded by a diffusion flame burning close to stoichiometric conditions in the periphery of the jet (nitrogen oxides formation area). That confirms the typical trade-off between soot and nitrogen oxide emissions related to diesel combustion engines. This map links in-cylinder conditions with regions where pollutant formation occurs. As observed in figure 2.1, the diesel combustion process typically crosses both soot and nitrogen oxide formation areas during the cycle.

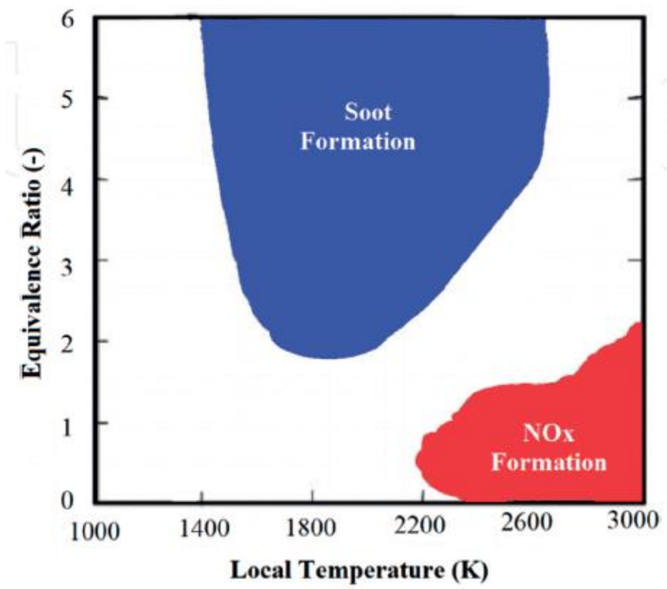


**Figure 2.1:** Emissions relative to local temperature with the change in the local equivalence ratio (Kassa and Hall, 2018)

Other than soot and NOx the burning of petroleum diesel also forms unburnt hydrocarbons (UHC), carbon monoxide (CO), and sulphur oxides (SOx) which are very poisonous gases (Dhar et al., 2015). Hydrocarbons are produced from the incomplete combustion of hydrocarbon fuel. The concentration level is related to misfire and poor charge mixture due to insufficient mixing between fuel and air. There are several sources for the formation of the HCs emissions such as oil film layers, wall flame quenching, deposits and crevices in the combustion chamber, liquid fuel, and exhaust-valve leakage (Alkidas, 1999). However, the most influential factor for HCs source is fully warmed conditions is combustion chamber crevices (about 38%) whereas under cold start or cold conditions the main HCs sources are wall flame quenching, lubricating oil films, and fuel preparation (Alkidas, 1999).

The formation process of CO is started by the reaction between oxygen and gaseous or liquid hydrocarbon fuel. The CO produced than can react either with oxygen or hydroxyl (OH) radicals to produce carbon dioxide. However, the oxidation rate of CO is slower than those of hydrocarbon species because CO still can be produced although there is enough oxygen (Bagal et al., 2009). Sulphur oxides (SOx) are formed during the combustion process in the engine because of the presence of sulphur content in the fuel. Carbon dioxide is the main product of carbon-contained fuel combustion. It is one of the primary emissions of greenhouse gases with nitrous oxide, methane, perfluorocarbons, sulphur hexafluoride, and hydrofluorocarbons (Kaewmai et al., 2012). Indeed, $CO_2$ is already present in the atmosphere but in very low concentration. Unfortunately, the amount has increased over the years at an alarming rate. Currently, the transportation sector accounts for roughly 28% of the total world $CO_2$ emissions from fuel combustion (World Energy Outlook-IEA, 2019).

In addition to this global effect, the use of diesel engines also implies local threats due to the adverse effects of NOx, UHC, CO, and particulate matter (PM) on human health. As a result, these topics motivated the implementation of the first emission legislation during the first decades of the 20th century. Since then, stringent emission legislation has driven the automotive industry and original equipment manufacturer (OEM) sector to look for more efficient and cleaner combustion techniques while maintaining engine performance. However, the entire automotive sector is currently facing new challenges in implementing BSVI emission guidelines in existing engines.

Simultaneously, the Government of India is banning old diesel vehicles in metro cities in initial phases and planning to phase them out from tier two and tier three cities may be in the next five years. This scenario has increased the interest in low-carbon fuels, the utilization of renewable fuels, technologies that offer higher fuel conversion efficiently, and alternative combustion modes. Nevertheless, diesel engines are still expected to be part of future technology solutions (World Energy Outlook-IEA, 2019). Consequently, more efforts on efficiency maximization and pollutant emission minimization will be required. One of the proposed strategies followed in the literature to mitigate emissions without compromising performance is modifying the existing engines to operate on dual fuel mode of combustion.

## 2.4 Dual Fuel Combustion Engine

The concept of dual fuel combustion was studied in way back 1901 by Rudolf Diesel. In this engine, the combustion of the primary gaseous fuel is initiated by secondary liquid fuel also called pilot fuel. However, the last decade of the 19$^{th}$ century shows significant growth in the emerging field of a dual fuel engine. The availability of gaseous fuels, as well as price fluctuations of liquid fuels, stimulated engine experts to carry their work in a modified diesel engine. Wartsilla is a commercially available dual fuel engine dedicated to the marine industry.

In a dual fuel engine, gaseous fuel is combusted through a pilot-ignited mode in which a small amount of diesel fuel is used to ignite a homogeneous mixture (García et al., 2017). Compared to ignition methods in spark ignition (SI) engines, ignition energy is increased in several orders of magnitude. However, the low mass of the pilot fuel multiple ignition centres are created, which improves ignitability near the lean range. Consequently, this modified combustion concept works as a hybrid model between SI and CI combustion processes, where multiple mechanisms overlap and influence each other. The conventional pilot-ignited dual fuel combustion (DFC) process can be analyzed in 4 different stages, viz.,

- Stage 1: Firstly, gaseous fuel is introduced into the combustion chamber. Typically, a mixture of gas and air is admitted during the intake stroke, which results in a homogeneous mixture by the end of the compression stroke.
- Stage 2: The second one consists of the period between the start of the pilot injection and combustion, named ignition delay. Here, the pilot fuel mixes with the surrounding fuel-air mixture until self-ignition takes place. The duration of this stage depends on the distribution of diesel fuel and the thermodynamic conditions (temperature, pressure, and oxygen level) in the engine cylinder. (Lata and Misra, 2011). In most cases, the pilot injection ends before the onset of combustion. Pilot injection timing, pressure, and fuel mass heavily affect this stage due to their effects on evaporation and mixing (Lakshmanan, T., and G. Nagarajan., 2011a.; Behera et al., 2014)
- Stage 3: The third stage is mainly linked to the combustion of the pilot fuel, and it is usually characterized by a high premixed peak event and ignition of the entrained gas into the pilot injection and the gas mixture in the vicinity of the pilot spray (Nagarajan and Nagalingam, 2008 )
- Stage 4: Finally, the combustion of gaseous fuel occurs in the remaining gas-air mixture. This stage is severely influenced by the interaction between diesel and gaseous during the ignition delay period. Ignition of the bulk charge can occur due to the gaseous fuel-air mixture's excessively at a high reaction rate (Karim, 1983). The previous categorization is adapted from the combustion model described by (Karim et al., 1999).

The various techniques to introduce gaseous fuel in the combustion chamber can be categorized depending on the location of an injector. Some of the strategies that have been followed in the literature are as follows:

- Gas Fumigation: Gas is supplied upstream of the engine intake manifold for mixing with the incoming air, maintaining a constant air-fuel ratio. Fuel flow is independent of the different valves of the cylinder, and the mixture tends to be highly homogeneous. Carburetors and mixers are examples of this type of injection.
- Port Fuel Injection: A fuel injector is mounted on the intake port, as close as possible to the intake valves, and supplies fuel to the combustion chamber during the gas exchange process.
- Direct Injection (DI): In this case, gaseous fuel is directly injected into the combustion chamber. Both low pressure and high-pressure direct injection systems exist, although both are still under development. In the low-pressure system, gas is directly injected into the combustion chamber no later than approximately $120^{o}$ CA BTDC, while the high-pressure system relies on gas injection closer to TDC. This method generates a higher degree of fuel stratification in the combustion chamber than fumigation or PFI systems.

The literature review suggests that most gaseous fuels are injected through indirect injection systems such as fumigation or PFI due to their simplicity. However, this technique has a slight drawback in terms of higher unburnt hydrocarbon levels until and unless combustion parameters are optimized (Karim et al., 2010). In dual fuel operation, the lower air-fuel ratio due to air replacement results in high HCs emission compared with normal diesel operation. The high heat capacity and lack of oxygen in the combustion chamber leads to a temperature reduction in in-cylinder as a result of misfiring and poor mixing, and hence resulting in a slower combustion rate allowing the unburned mixture to leave the combustion chamber (Sahoo et al., 2009).

Direct injection systems address the issues of an indirect injection system by increasing the gaseous fuel stratification levels in the combustion chamber. Moreover, the knocking tendency is also reduced in such engines. Furthermore, these engines inherently get benefited from increased volumetric efficiency in comparison to indirect gas injection engines (Selim, 2004). However, this technology is still in progress because of increased complexity in hardware systems as well as higher cost (Karim, Klat, and Moore, 1966). Many engine scientists have stated that by modifying injection techniques significant reduction in $NO_x$ and $CO_2$ emissions can be obtained in dual fuel operation in comparison to neat diesel operation. Gaseous fuels are the lifeline of a dual fuel engine. Hence various gaseous fuels like CNG, hydrogen, LPG, biogas, syngas, producer gas, acetylene, etc. have been explored by several engine experts.

### 2.5 Alternative Gaseous Fuels

Gaseous fuels are very eco-friendly in nature and have excellent mixing properties. Moreover, they do not have problems like corrosion and ash formation issue as liquid fuels do. Hence it is utilized by many engine experts in their study. The various gaseous fuels, which have been explored in the literature are presented below. Their impact on the performance, combustion, and emissions characteristics are explained in detail.

#### 2.5.1 Compressed Natural Gas (CNG)

Ryu (2013), Yang et al., (2013), and Mustafi et al., (2013) evaluated the performance of the natural gas-fuelled diesel engine. Many engine experts have used electronic injectors to inject both CNG as well as pilot fuels in dual fuel operation mode (Wang et al., 2013; Nwafor, 2000). Wang et al., (2015) and Cheenkachorn et al., (2013) studied the impact of a varying flow rate of natural gas (NG) on the emissions characteristics of the heavy-duty engine with turbocharger. Furthermore, Subramanian and Kalsi (2017) investigated the performance of CNG engine supplemented with hydrogen on RCCI mode.

Cheenkachorn et al., (2013) and Raman and Ram (2013) observed lower thermal efficiency and lower volumetric efficiency in the case of a CNG dual fuel engine, relative to baseline diesel. Moreover, similar trends were observed by Nwafor (2000) and Mustafi et al., (2013). Additionally, Mustafi et al., (2013) noticed that the thermal efficiency of the dual fuel engine deteriorated by 5% while increasing the quantity of CNG (80% substitution of diesel). Furthermore, Ahmat and Yahya (2019) also observed a similar declining trend of thermal efficiency in their experimentation. They explained the reason for decreased thermal efficiency to be the lower burning velocity of natural gas.

Vijayabalan et al., (2009) found that a dual fuel engine suffers low brake thermal efficiency (BTE) compared to a regular diesel engine, especially at low loading conditions because of lean fuel-air mixture formation at low load. In contrast, Subramanian and Kalsi (2017) found higher thermal efficiency at 64% of CNG assisted with hydrogen at full load which shows better utilization of gaseous fuels at top loading conditions. Similarly, Kumar and Kumar (2016) and Wang et al., (2015) observed that (BTE) was inclined to 32.1% and 35% at peak load respectively while analyzing their engine under dual fuel mode. As the load increases the mixture gets richer which increases the combustion temperature of the fuel-air mixture consequently leading to the reduction in the ignition delay period of CNG (Nithyanadan et al., 2016).

Due to higher combustion temperature, an improved CNG utilization occurs inside the engine cylinder which further increases the thermal efficiency at higher loads (Banapurmath et al., 2018). Furthermore, Kumar and Kumar (2016) reported lower brake specific fuel consumption (BSFC) i.e., 0.203 kg/kW-h while investigating Jatropha oil methyl ester (JOME) as pilot fuel with CNG in a dual fuel mode. The reason stated for decreased BSFC was improved ignition characteristics of biodiesel pilot fuel.

Tarabet et al., (2014) utilized eucalyptus biodiesel as a pilot fuel whereas, Korakianitis et al., (2011) exploited three different pilot fuels, i.e., dimethyl ether (DME), emulsified biodiesel, and Rapeseed methyl ester (RME) in a modified diesel engine operated with CNG. Subramanian and Kalsi (2017) and Paul et al., (2015) employed Pongamia pinnata biodiesel as an ignition source for constant speed (1500 rpm) engine at a maximum limit of 5 kW output power. On the other hand, Cheenkachorn et al., (2013) analyzed the performance of dual fuel engine by varying their speed from 1100 rpm to 1900 rpm and found optimum performance at 1300 rpm with 77.90% of diesel replaced by natural gas. Tarabet et al., (2014) fixed the maximum amount of pilot fuel equivalent to 10 % of total energy.

Banapurmath et al., (2018), Paul et al., (2015), Tarabet et al., (2014), and Mustafi et al., (2013) have observed reduced levels of $CO_2$, CO, and HC at higher loads while injecting biodiesel/diesel blends as a pilot source whereas Cheenkachorn et al., (2013) and Korakianitis et al., (2011) found the rise in HC and CO emission levels in CNG fuelled engine compared to neat diesel. However, Wang et al., (2015), Paul et al., (2015), Mustafi et al., (2013), and Korakianitis et al., (2011) found that soot and NOx emissions were lessened considerably during dual fuel combustion mode relative to standard diesel fuel. This makes a dual fuel engine more advantageous than a conventional engine. The reason behind the lowering of NOx emission was lean premixed combustion which reduces the combustion temperature of engine.

However, some penalty of NOx emissions was observed by Ahmat and Yahya (2019), Subramanian and Kalsi (2017), and Tarabet et al., (2014) with biodiesel -CNG engine. The higher rate of NOx emissions is attributed to the lower energy content of pilot fuel as well as the higher amount of oxygen present in the biodiesel. To restrict NOx emissions few engine researchers have explored the exhaust gas recirculation (EGR) technique (Mustafi et al., 2013). Bertoa et al., (2020) measured the unregulated pollutants using CNG vehicles. Wang et al., (2015) and Nwafor (2000) evaluated the combustion characteristics of heavy-duty CI engine fuelled with natural gas by optimizing the injection angle. The optimized injection angle was 42.5$^{o}$ crank angle before top dead centre (BTDC) for Wang et al., (2015) whereas Nwafor (2000) detected that ignition angle was advanced by 1.5$^{o}$. However, Banapurmath et al., (2018) observed the optimum performance at injection timing of 27°BTDC. Hence it is recommended not to change injection parameters frequently for the smooth running of an engine.

### 2.5.2 Hydrogen

Korakianitis, Namasivayam, and Crookes (2011) injected RME and diesel fuel as a pilot source in a modified CI engine running on hydrogen. The results showed that thermal efficiency was comparable while inducting hydrogen and RME/diesel pilot fuels compared to baseline diesel. Furthermore, Geo, Nagarajan, and Nagalingam (2008) studied the impact of neat rubber seed oil (RSO) and its biodiesel (RSOME) on the engine parameters of hydrogen dual fuel engine. They observed higher BTE than neat diesel at higher loads with hydrogen RSOME and RSO pilot fuels because of the superior ignition quality of hydrogen fuel. Moreover, Köse and Ciniviz (2013) also noticed an inclining pattern of BTE while increasing the percentage of hydrogen in the modified engine because of higher specific energy density. The reasons for improved performance might be the efficient mixing of gas-air fuel mixture, promoting better combustion (Köse and Ciniviz, 2013).

Subramanian and Thangavel (2020) analyzed the performance and emission characteristics of the hydrogen-fuelled engine by varying the flow rates of hydrogen from 6 LPM, 12 LPM, 18 LPM, 24 LPM, 30 LPM, and 36 LPM along with the air in the intake. It was noticed that BTE declined for all operating conditions except at 6 LPM of hydrogen gas. This may be because of the higher flow rate, a larger heat transfer from the burning gas to the combustion chamber walls takes place which reduces the thermal efficiency (Tsujimura and Suzuki, 2017). Rahman et al., (2017) fixed the flow rate of hydrogen at 4 LPM for optimizing the performance of dual fuel engine whereas Gnanamoorthi and Vimalananth (2020) found an optimal hydrogen flow rate at 30 LPM for CRDI engine. Furthermore, an improvement of 30.65 % in BTE was achieved with a reduction in BSEC of 23,48 % was also noticed during their analysis.

Gnanamoorthi and Vimalananth (2020), Subramanian and Thangavel (2020), and Korakianitis, Namasivayam, and Crookes (2011) stated that NOx emissions were inclined during a hydrogen-fuelled engine with diesel/biodiesel pilot fuels, whereas CO, smoke, and HC emissions remained similar that of the regular diesel engine. It was also observed that the delay period increased in the case of dual fuel engine, thus might be due to higher flame propagation speeds of gaseous fuels, eventually increasing the higher-combustion temperature causing higher NOx emissions (Korakianitis et al., 2011). Few engine experts have also stated the higher level of NOx emissions might be due to instantaneous combustion caused by hydrogen gas at elevated load (Saravanan et al., 2008; White et al.,2006). However, Tutak et al., (2020), Ouchikh et al., (2019), and Li et al., (2017) also observed escalated NOx levels while injecting hydrogen and natural gas in dual fuel engine.

Hydrogen addition in a CNG engine increases peak pressure and the temperature of the combustion chamber resulting in higher NOx emissions (Tutak et al.,2020). Moreover, Verma et al., (2018) noticed that hydrogen intake has less impact on low-load combustion parameters, however, at high-load combustion, NOx trends drastically change with higher heat release rates and elevated combustion pressures. However, Kumar and Ibrahim (2020) Khatri and Kishore (2020), and Saravanan et al., (2008) found a substantial reduction in NOx emissions by utilizing EGR. However, a slightly increased level of smoke emissions was also found in these engines (Dhar et al., 2019). Moreover, Lu et al., (2016) utilized 50 % EGR whereas Rahman et al., (2017) employed 10% EGR and observed that the level of NOx emissions was reduced at a substantial rate using port injection through a common rail. Saravanan et al., 2008 found the optimized injection angle was $5^o$ CA BTDC and $40^o$ ATDC for DEE pilot fuel, whereas Khatri and Kishore (2020) observed the optimized injection timing at 21 °CA BTDC. Gnanamoorthi and Vimalananth (2020) injected hydrogen into the engine cylinder at a crank angle of $23^o$ BTDC for improving the performance of the dual fuel engine.

Kumar et al., (2018) examined the impact of ethanol and water injection techniques on the hydrogen- Madhuca longifolia oil dual fuel engine. The maximum permissible limit of hydrogen proportion was 20%. Ethanol and water, along with hydrogen, may be injected to boost efficiency. Simultaneously DEE was utilized as an ignition source to improve the combustion characteristics of a dual fuel engine. The maximum share of hydrogen energy was limited to 19%, in the case of Tutak et al., (2020) analysis whereas Dhar et al., (2019) and Ouchikh et al., (2019) observed that the maximum share of hydrogen energy about 30%. Moreover, at 10%, hydrogen enrichment cumulative heat release was maximum (Ouchikh et al., 2019) whereas Verma et al. (2018) noticed that the maximum addition of hydrogen was limited to 20% for maximization of performance. Dimitriou et al., (2018) utilized up to 98% hydrogen and found that the engine could still operate smoothly.

### 2.5.3 Liquified Petroleum Gas (LPG)

Saleh (2008) conducted experimentation to investigate the impact of inducting LPG with various constituents on the dual fuel engine's performance. It was observed with varied LPG composition, the exhaust pollutants were reduced significantly (Saleh, 2008; Dinçer et al., 2020). The performance at intermediate loads was improved by employing the EGR technique (Saleh, 2008; Ashok, Denis, and Kumar, 2015). The authors also concluded by utilizing LPG, the thermal efficiency improved significantly at higher loading conditions while the performance of the modified engine gets deteriorated at lower loads (Tsolakis et al., 2014). The reason might be a poor quality of mixture formation at lower loads causing an improper combustion process. (Ashok, Denis, and Kumar, 2015).

To improve the performance of dual fuel engine Tsolakis et al., (2014) utilized diesel/ rapeseed methyl ester (RME) as pilot fuel. Moreover, Lata, Misra, and Medhekar (2012) and Tsolakis et al., (2014) observed that the performance improved substantially while utilizing LPG/RME under dual combustion mode. Furthermore, CO, HC, and soot emissions were alleviated considerably with a slight penalty of NOx emissions relative to baseline diesel (Tsolakis et al., 2014). The emission results were contradicting with Lata, Misra, and Medhekar (2012). Lata, Misra, and Medhekar (2012) observed escalated CO and HC levels, while NOx and smoke levels were alleviated considerably relative to petroleum diesel. To reduce the NOx levels, few researchers have also utilized n-butanol/diesel blend as a pilot source with excess propane supplied in the intake manifold (Saleh, 2008). The reduction of NOx levels might be due to their shorter combustion duration and higher-energy content compared with primary alcohol-diesel blends.

However, Wu, Horng-Wen, and Hung (2014) used the EGR method to restrict NOx emissions in a modified engine fuelled with diesel/biodiesel and LPG. It was found that NOx and smoke emissions were reduced substantially at an intermediate loading condition compared to standard diesel (Wu, Horng-Wen, and Hung, 2014). Hence it is recommended to use a dual fuel engine for reducing harmful pollutants. Many engine experts have used electronic injectors to optimize the performance of dual fuel engines (Lata, Misra, and Medhekar, 2012). Maximum substitution of diesel was observed 60% in the case of LPG engine. Furthermore, Ashok, Denis, and Kumar (2015) suggested that parameters like the quantity of pilot fuel, injection parameters, intake charge conditions, and quantity of gaseous fuel are very important to optimize the performance of a dual fuel engine.

### 2.5.4 Producer Gas

Murugan et al., (2013) introduced the producer gas in the inlet manifold of the engine at a flow rate of 4 LPM, 6 LPM, and 8 LPM, respectively, while diesel was injected into the engine as the main fuel. Moreover, Ramadhas, Jayaraj, and Muraleedharan (2008) investigated the performance of coir-pith-derived producer gas. The outcomes were comparable BSEC and slightly higher emissions levels during dual fuel operation than standard diesel. Raviranjan et al., (2016) observed that at lower loads, dual fuel operation shows lower efficiency with producer gas may be due to lower combustion rate cause by the low calorific value of the producer gas. Murugan et al., (2013) noticed that a reduction in the consumption of diesel was observed when operated on dual fuel mode despite a reduction in brake thermal efficiency. Ramadhas, Jayaraj, and Muraleedharan, (2008) also found similar trends in their study.

Banapurmath, Tewari, and Hosmath (2014) studied the influence of diesel, Honge oil, and its methyl ester (HOME) on engine running on producer gas. Comparable BTE was found for diesel, Honge oil, and its methyl ester (HOME) in producer gas engine. Moreover, NOx and smoke levels were lessened drastically, while HC and CO emissions were escalated at higher loads than baseline diesel (Banapurmath, Tewari, and Hosmath, 2014; Ramadhas, Jayaraj, and Muraleedharan, 2006). In another study carried by Murugan et al., (2013) it was observed that a similar declining pattern of emission analysis by utilizing producer gas. Hence, to reduce the pollutant emissions like NOx and PM of diesel engines, the addition of small gaseous fuel amounts or dual-mode operation has been proved as potential techniques.

However, at high load dual fuel engine may suffer from $CO_2$ emission as well as higher consumption of pilot fuel may take place to burn producer gas completely (Raviranjan et al., 2016). Raviranjan et al., (2016) and Ramadhas, Jayaraj, and Muraleedharan (2006) observed that diesel substitution was 56% at 20% opening of producer gas, whereas diesel fuel substitution reaches 78% at 100% opening of producer gas. Furthermore, at 80% load, the engine produced optimum efficiency.

### 2.5.5 Biogas

Barik and Murugan (2016a.) studied the impact of DEE/KME injection on the dual combustion engine's performance operated with biogas. Bora and Saha (2015) compared biogas fuelled CI engine performance with three different pilot fuels, i.e., Pongamia oil methyl ester, rice bran oil methyl ester, and palm oil methyl ester. The thermal efficiency was found to be lower in comparison to neat diesel at a higher load. This is due to the lower calorific value of biogas in comparison to diesel. However, few researchers have utilized DEE to enhance the performance of biogas dual fuel engine (Barik and Murugan, 2016a.). Moreover, NOx and smoke levels were reduced significantly with biodiesel pilot fuel in biogas dual fuel engine due to lower HRR and lower cumulative energy (Barik and Murugan, 2016a; Bora and Saha, 2015; Jagadish and Gumtapure 2020).

However, the levels of CO and HC emissions were higher with biodiesel pilot fuels and it should be reduced by using after-treatment technologies (Verma et al., 2019; Barik and Murugan, 2016a). Very few engine experts have varied the pilot fuel injection timing (26°, 29°, and 32°) as well as compression ratio (18, 17.5, 17, and 16) to enhance the performance of biogas fuelled engine (Bora and Saha, 2015). At 29° BTDC and compression ratio of 18 superior performance was observed for biogas/diesel engines with an 82% diesel substitution rate. Moreover, CO and HC levels were also lowered down by advancing injection angle by $6^{o}$, (Owczuk et al., 2020) however this result contradicted Barik and Murugan (2016a.) results. Furthermore, Jagadish and Gumtapure (2020) employed different mixtures of biogas at $27.5^{o}$ BTDC for various loads. In general, it can be concluded that injection timing variations have a significant effect on the performance and pollutant formation in dual fuel engines.

### 2.5.6 Syngas

Azimov et al., (2011) studied the performance parameters of dual fuel engine by utilizing syngas. It was observed that with higher hydrogen content, thermal efficiency improved substantially. Moreover, lower HC and CO levels were observed with a slightly higher rate of NOx emissions (Azimov et al., 2011). However, Sahoo, Niranjan, and Saha (2012) observed reduced thermal efficiency in syngas dual fuel engine may be due to low calorific value of syngas. Similar results have been reported by Wagemakers and Leermakers (2012). However, the level of HC and CO escalated, whereas NOx levels were alleviated at optimum performance than diesel (Sahoo, Niranjan, and Saha, 2012; Paykani et al., 2020). Syngas could make the combustion of diesel-air mixture faster, reducing the duration time of higher temperature and decreasing the NO$x$ emission. Furthermore, Guo, Neill, and Liko (2016) evaluated the combustion parameters of syngas fuelled diesel engines at intermediate loads.

The results obtained were similar to those of Sahoo, Sahoo, and Saha (2009). Moreover, the engine may suffer from poor combustion and misfire at partial loads (Singh, Singh, and Mohapatra, 2019). Ali et al., (2019) examined the combustion dynamics of a dual fuel engine with syngas-diesel under very lean fuel-air mixing conditions. The analysis was carried out on 3D CFD modeling combined with chemical kinetics for evaluating hydrogen content in the syngas. The findings of the CFD analysis were in fair accordance with the results of the experiment.

### 2.5.7 Acetylene as a Novel Gaseous Fuel

Conventionally acetylene is produced from calcium carbide (biproduct of calcium carbonate) by reacting calcium carbide with water (Srivastava et al., 2017). However, nowadays acetylene is also produced from natural gas which is renewable in nature. It is a colourless gas and smells like garlic (Schobert, 2014; Sharma et al., 2012). Acetylene possesses similar ignition properties like hydrogen and it is less hazardous than hydrogen. The significant combustion properties such as wide flammability range and higher flame speed make acetylene a viable alternative fuel (Raman and Kumar, 2019a.). Additionally, the adiabatic flame temperature of acetylene is higher than hydrogen and the minimum ignition energy of acetylene–air mixture is least in comparison to other flammable gases available.

However, it is difficult to burn acetylene because of higher auto ignition temperature. Hence, many engine scientists have utilized acetylene in diesel engine under dual fuel mode of combustion. Moreover, the wall wetting problems of diesel engines was eliminated using acetylene gas because of its better mixing property with air (Raman and Kumar, 2020). The acetylene gas can be easily inducted in diesel engines even at higher compression ratio (Srivastava et al., 2017). Lakshmanan and Nagarajan (2010) injected the acetylene at the rate of 110 g/s, 180 g/s, and 240 g/s, with the help of electronic injectors. However, in the separate study Lakshmanan and Nagarajan (2011a.) inducted acetylene at the flow rate of 0.20 kg/h, 0.26 kg/h, and 0.39 kg/h using a dual fuel combustion approach. Furthermore, Lakshmanan and Nagarajan (2011b.) studied the performance features of acetylene by modifying injection techniques.

During their experimentation, acetylene was inducted into the intake port at the rate of 110 g/h, 180 g/h, and 240 g/h. Moreover, Behera et al., (2014) successfully operated diesel and biodiesel blends with acetylene in the modified diesel engine at diverse injection rates of acetylene gas (132 g/h, 198 g/h, 264 g/h, and 330 g/h). However, Srivastava et al., (2018) investigated the performance of dual fuel engine by varying the gas flow rate of 60 lph, 120 lph, 180 lph, and 240 lph at compression ratio (CR) of 18, 19, 20, 21, and 22. Moreover, Choudhary, Nayyar, and Dasgupta (2018) analyzed the effect of inducting acetylene at a steady flow rate of 7 LPM at different compression ratio (CR) of 18:1, 18.5:1, 19:1, and 19.5:1. Subsequently, Koli and Rao (2020) carried out an experimental investigation with a decreased compression ratio (CR) from 18 to 13 and varying the flow rate of acetylene from 2 LPM to 5 LPM in a variable compression ratio

(VCR) engine. However, Sonachalam and Manieniyan (2020) investigated the performance by injecting acetylene gas and mahua oil methyl ester on the RCCI engine at a steady flow rate of 4 LPM. According to the observation by Lakshmanan and Nagarajan (2011a.), when acetylene was inducted into intake manifold smoke, HC and CO levels were lessened drastically with a slight reduction in BTE relative to neat diesel operation. Moreover, Lakshmanan and Nagarajan (2011b.) and Lakshmanan and Nagarajan (2010) also found similar results in their study except for a rise in smoke and a reduction in NOx emissions.

NOx emission levels were increased extensively because of a rise in peak cylinder pressure and higher flame speed of gaseous fuel, finally causing extreme combustion rate (Lakshmanan and Nagarajan, 2011a.; Srivastava et al., 2018). However, Sudheesh and Mallikarjuna (2012) used DEE as an additive to reduce NOx and smoke levels at a higher load under HCCI mode. Additionally, they found that BTE of the acetylene/DEE HCCI engine was improved than regular diesel (Sudheesh and Mallikarjuna, 2012; Sudheesh and Mallikarjuna, 2010). However, HC and CO emissions were escalated at peak load conditions (Sudheesh and Mallikarjuna, 2012) which are the major challenges of HCCI engines. Hence, Mahla et al., (2012) blended DEE with diesel up to 20 % under dual fuel mode of operation for the improvement in the combustion characteristics and emission reductions.

It was found that the addition of DEE to pilot diesel fuel generally results in the reduction in density, viscosity, flash point, fire point, cloud point, pour point and heating value while it also improves cetane number of the blended fuel. The improved physicochemical properties of DEE/diesel blends are the primary reason of the effective combustion of dual fuel engine. Additionally, Behera et al., (2014) used biodiesel blends in modified diesel engine to reduce exhaust pollutants. Furthermore, it was observed that peak cylinder pressure and BTE of dual fuel increased. However, NOx increased sharply with biodiesel/diesel blends in dual fuel engine. Similar trends were observed by Choudhary, Nayyar, and Dasgupta, (2018) by increasing compression ratio to 19.5.

Furthermore, Behera et al., (2014) analyzed the combustion parameters by optimizing injection timing. It was noticed that the ignition delay was reduced by $3^{o}$ CA at a higher load, and the maximum cylinder pressure was also reduced by 25% when acetylene was inducted at full load (Behera et al., 2014). However, injection timing of $5^{o}$ after top dead centre (ATDC) and injection duration of $90^{o}$ CA (9.9 ms) gives an optimized performance (Lakshmanan and Nagarajan, 2011b.). Moreover, an optimized injection angle of 10° ATDC and a 90° crank angle duration were noticed by Lakshmanan and Nagarajan, (2010).

Specific techniques like timed manifold injection and port injection were adapted to lessen NOx levels and pre-ignition problems during acetylene induction (Lakshmanan and Nagarajan, 2011a.; Lakshmanan and Nagarajan, 2010). Consequently, Wulff et al., (2001) suggested utilizing alcohol as a pilot fuel will improve the combustion characteristics of pilot fuel. Hence, Brusca et al., (2014) conducted an experimental study on alcohol and acetylene with multiple injectors. They observed comparable BTE and relatively lower pollutants during dual fuel mode of combustion (Brusca et al., 2014). Table 2.1 shows the physico-chemical properties of gaseous fuels.

**Table 2.1:** Physico-chemical Characteristics of gaseous fuels (Raman and Kumar, 2019)

| **Properties** | **Diesel** | **Hydrogen** | **CNG** | **LPG** | **Producer Gas** | **Syn Gas** | **Bio-gas** | **Acetylene** |
|---|---|---|---|---|---|---|---|---|
| Formula | $C_{12}H_{26}$ | $H_2$ | $CH_4$ | $C_3H_8$ | $CO + H_2$ | 50%CO +50% $H_2$ | $CH_4+CO_2+ H_2+N_2$ | $C_2H_2$ |
| Auto ignition temperature (K) | 530 | 858 | 730 | 743 | NA | NA | 700 | 578 |
| Flammability limits (vol% in air) | 0.7-5 | 4- 75 | 5.3-15 | 2.2-9.5 | NA | NA | 7.5-14 | NA |
| Stoichiometric Air-Fuel Ratio | 14.5 | 34.3 | 17.3 | 15.6 | 1.12 | 4.5-8 | 6 | 13.2 |
| Density (kg/m$^3$) | 850 | 0.09 | 0.08 | 2.24 | 1.287 | 0.67 | 1.1 | 1.095 |
| Net heating value (MJ/kg) | 42.5 | 120 | 50 | 46.3 | 4.8 | 14.8 | 0.5 | 48.2 |
| Flame velocity(cm/s) | 30 | 265-325 | 34 | 38.25 | 50 | 18 | 25 | 150 |
| Adiabatic-flame Temperature(K) | 2200 | 2400 | 2214 | 1980 | 1873 | NA | NA | 2500 |

## 2.6 Alternative Pilot Fuels

In a dual-fuel combustion process, the ignition of the primary gaseous fuel depends on liquid fuel injection. It plays a very crucial role in igniting a gaseous fuel-air mixture. Hence the liquid fuel is also called pilot fuel. Many engine experts have utilized diesel as a pilot fuel in dual fuel engine. However, few engine experts have also explored biodiesel/diesel blends as an ignition source. Moreover, it has been observed that the performance of such engines deteriorated because biodiesel fuel has less energy per unit volume than neat diesel. Sometimes deposits and clogging are also noticed in the biodiesel-based engine due to the low quality of biodiesel. Moreover, availability issues and higher production costs with biodiesel have been fairly reported by various scientists.

Furthermore, biodiesel blends have a higher tendency of NOx formations in conventional diesel engine due to their higher oxygen content. Oxygenated fuels such as alcohols and ethers have the potential to provide reliable sources, and environmentally friendly fuel to the world's increasing future energy demands. Such fuels have a promised future since are renewable and produced from several biomass sources, which also can be produced locally. Hence, many engine scientists have explored DEE as an additive with diesel/biodiesel blend by varying the blend ratio from 5 to 20 % to improve the performance and emission characteristics. (Jothi et al. .2008, Patil and Thipse, 2015). They have reported that the blends of DEE-diesel/biodiesel significantly improve physicochemical characteristics such as density, viscosity, acid value, and pour point (Górski et al., 2014). Different researchers have reported that mixing DEE into pure diesel improved air/fuel mixture quality improves and gradually increases thermal efficiency(Subramanian et al., 2008).

Furthermore, NOx, PM, smoke, and soot emissions were drastically reduced by utilizing DEE in CI engine (Murugan et al., 2016; Patil and Thipse,2015; Lee and Kim, 2017). However, several studies have revealed that over 15% DEE/diesel blend showed a little bit of instability and variability in engine speed and power output. Simultaneously, a few researchers have also pointed out that the engine will release a higher level of smoke at full load with a higher quantity of DEE. (Karabektas et al., 2014). Severe knocking, hard-starting, and poor engine performance were also observed by many researchers (Rakopoulos et al., 2013) when the DEE proportion was increased more than 20 % or above because of the lower viscosity of DEE and may cause leakages in the fuel supply system.

Furthermore, Rakopoulos et al., (2016) have blended DEE up to 24% with neat diesel and found that decreased emissions rate of smoke, CO, and NOx meanwhile HC emissions increased drastically. Bailey et al., (1997) have reported the cost analysis of DEE production. Few scientists have also recommended by retarding the injection timing and using EGR could reduce harmful emissions like $NO_x$ substantially with a marginal increased rate of UBHC, CO, and particulate matter (Mallikarjun, 2015). Numerous experiments focused on conventional alcohols, such as methanol and ethanol blends in CI engines (Zhenet al., 2015; Verhelstet al., 2019; Shahiret al., 2005). However, a limited study on the use of higher-order alcohols, such as butanol, has been investigated (No et al., 2016).

Many engine experts have found n-butanol to be more beneficial than lower-order alcohols such as methanol and ethanol because of its higher calorific content, quite similar to straight diesel, excellent miscibility with diesel oil, lower viscosity, and flashpoint (No et al., 2016). Moreover, it favors fuel atomization, low latent heat of vaporization heat (LHV), improved blending stability, less vulnerability to oxidation, less volatile, higher cetane index, and a lower auto-ignition temperature than lower carbon alcohol (Rajesh et al., 2016). Many engine experts have cited butanol as a revolutionary biofuel produced by utilizing food wastage and biomass, eukaryotic micro-organisms, and other cellulose-containing plants(Cheng et al., 2011; Kumari et al., 2018).

However, the use of butanol as the primary fuel in CI engines is problematic due to its lower calorific value and low octane number (No et al.,2016 Rajeshet al., 2016). Recently very effective technologies have been implemented to produce n-butanol at higher rates (Gu et al., 2011; Malaviyaet al., 2012). Rakopoulos et al., (2010) noticed that the delay period was prolonged while increasing the n-butanol percentage. Moreover, peak pressure rises with a marginal inclination in the BTE were also observed (Rakopouloset al., 2010).

## 2.7 Optimizing Injection Parameters

Many engine experts have suggested that by modifying injection variables performance of the dual fuel engine can be controlled to a large extent. The injection parameters are studied in brief.

### 2.7.1 Injection Timing

This parameter controls combustion phasing. Most of the researcher has found that by advancing pilot injection, a reduction of HC emissions and higher NO levels under lean operation is observed, consequently, a trade-off between these two groups of species exist (Kuleshov, Kozlov and Mahkamov, 2010; Lakshmanan and Nagarajan, 2009). This is due to an improvement in combustion efficiency with increased mixing of pilot fuel vapor with the premixed mixture. However, this trend does not apply to mixtures closer to stoichiometric conditions. As explained earlier, the unburned hydrocarbons trapped in the cylinder crevices are released during the expansion stroke's early stages.

Since combustion efficiency is not a limiting factor for lean mixtures, the relative effect of crevices on HC emissions increases and retarded combustion phasing. Eventually producing lower

HC emissions due to the oxidation of fuel released crevices (Basha et al., 2016). Moreover, advancing pilot injection timing further causes longer ignition delays, which deteriorates combustion efficiency (Behera et al., 2014). Hence it is recommended to optimize the injection angle for the escalated performance of dual fuel engine.

### 2.7.2 Injection Pressure

The variation of pilot injection pressure directly affects the dispersion and location of the diesel fuel vapours in the combustion chamber. For the same amount of fuel, higher injection pressure results in shorter injection duration, which further increases the mixing time of the pilot fuel with the premixed charge. Consequently, the equivalence ratio at which pilot fuel auto ignites is reached earlier and ignition is advanced. This effect is generally offset by higher temperature and pressure around TDC in general shortens the ignition delay. Many engine experts have concluded that increased injection pressure by modifying the injection rate of the fuel pump and nozzle area, improves smoke and fuel consumption, particularly at low and medium speeds. To extend these effects of high-pressure injection, a more optimized combustion system and minimized injection equipment drive torque must be required (Choudhary et al., 2018; Lakshmanan and Nagarajan. 2010a).

### 2.7.3 Injection Duration

The injection duration governs an immense impact on pilot size. Given a fixed pilot injection timing and injection pressure, longer injection duration, and consequently higher pilot fuel mass injected results in earlier combustion phasing and higher peak cylinder pressures (Soni et al., 2008). In general, the proportion of diesel fuel should be reduced compared to gaseous fuel due to economic considerations. However, there are some limiting aspects. Firstly, reduced pilot mass means lower ignition energy, which in cold starts or excessive lean operation results in high HC emissions and reduced combustion stability (Kumar and Kumar, 2016). Besides, load changes are usually advantageous due to the higher temperature and pressure present in the combustion chamber for improved combustion efficiency.

Further, it indicates a decrease in pilot fuel consumption as load increases. However, overheating of the pilot injector tip may occur (Korakianitis et al., 2014). Lastly, the accuracy and repeatability of the diesel injector for small pilots (below 5% of total fuel energy) may be reduced since the required short injection duration can result in ballistic behavior of the injector (Mathur, Das, and Patro, 1993). For these reasons, some dual fuel engines have a dedicated pilot injection system, independent from the primary diesel injection system.

### 2.8 Strategies to Reduce Emissions

Exhaust gas recirculation is the most well-known and established technique for reducing NOx due to its simplicity and low-cost implementation. This effective NOx emission control method is commonly used in many modern high speed direct injection (HSDI) diesel engines. The application of EGR is simple and straightforward especially in a naturally aspirated diesel engine. The exhaust gas flows to the intake manifold through the throttling valve as exhaust pressure is higher compared with the intake (Kumar and Kumar, 2016). The reduction in NOX is achieved mainly because EGR displaces oxygen concentration in the combustion chamber hence reducing overall flame temperature. The presence of EGR will alter the combustion process and heat released in three major ways. First, the $O_2$ concentration is reduced as a consequence of the exhaust gas presence in the combustion chamber (dilution effect). This causes the injected fuel to spread further to find oxygen for stoichiometric combustion.

Therefore, the flammable region is extended. Apart from the stoichiometric mixture the region also includes a portion that contains $CO_2$, $H_2O$, and $N_2$ from the exhaust gas. The reduced oxygen concentration also lowers the oxygen partial pressure affecting the kinetics of NOX formation reactions (Zhenet al., 2015). The second effect of EGR is the thermal consequence. $CO_2$ and $H_2O$ as the inherent components of EGR have higher specific heat capacity compared to those of substances from ambient air such as $O_2$ and $N_2$. Consequently, the overall heat capacity on the cylinder charge is reduced and hence lowering the flame temperature and propensity in NOx production (Karim et al., 2010).

The modification of in-cylinder temperature and composition of the inlet charge (the addition of $CO_2$ and $H_2O$) due to the application of EGR also affects the start of ignition. It is generally observed that the ignition delay increases. However, the main problem caused by the EGR application is the significantly increased particulate matter (PM) formation. The reduced in-cylinder temperature and pronounced diffusion combustion are the main reasons for decreased oxidation ability of the combustion product. Other adverse effects of the utilization of EGR include an increase in the specific fuel consumption and deterioration of engine durability, lubricant oil quality, while the engine may suffer high cycle-to-cycle variations and power loss (Kumar and Ibrahim, 2020)

## 2.9 Prediction of Engine Parameters Using Artificial Neural Network

Response surface methodology (RSM) has been used by many engine experts for optimizing the production of biofuels as well as to modify the performance results. However, RSM has certain limitations i.e., the magnitude of the approximation errors is difficult to analyze and it is invalid for regions other than the studied ranges of factors. Hence, an artificial neural network (ANN) has been implemented by various researchers to predict the results. Furthermore, the ANN model can predict engine performance and exhaust emissions quite well with correlation coefficients with very low root mean square errors. Root mean squared error (RMSE) is the square root of the mean of the square of all of the errors.

The use of RMSE is very common, and it is considered an excellent general-purpose error metric for numerical predictions. ANN-based engine modelling offers the potential for a multidimensional, adaptive, learning control system that does not require knowledge of the governing equations for engine performance or the combustion kinetics of emissions formation that a conventional map-based engine model requires. Most of the engine experts have considered engine speeds, mass fuel injection rates, and intake air temperatures as input variables. However, few of the researchers have also selected engine load, engine speed, and fuel blending ratio as input parameters. Moreover, brake specific fuel consumption (BSFC), exhaust gas temperature (EGT), brake thermal efficiency (BTE), nitrogen oxides ($NO_x$), hydrocarbons (HC), carbon monoxides (CO), and smoke were selected as the output parameters.

In order to find the relations between input and output parameters, many engine scientists have utilized the ANN technique using 75% of the experimental results for training. The outputs from the ANN for some set of control parameters were compared with experimental results. The low value of RMSE was observed by many experts which shows good coordination between actual and predicted results. Consequently, few researchers have used correlation coefficient (R), the fraction of variance ($R^2$), and mean absolute percentage error ($\xi$) for comparing the performance of the network. Moreover, Levenberg-Marquardt (LM) algorithm with 11 neurons produced the best results among the examined combinations of learning criteria to predict engine performance and exhaust emissions. After completing the training process, the inspection and testing phase is carried out to analyze predicted results. Moreover, the validation is done by feeding a new packet of input data into the network to justify results.

Ho et al., (2010), Wan Mansor et al., (2019) and Hariharan et al., (2020) predicted exhaust emissions of hydrogen dual fuel modified engine. However, Karthic and Kumar (2019) predicted the performance as well as emission characteristics of a Mahua oil-hydrogen dual fuel engine. Furthermore, Paul et al., (2018) and Yusuf et al., (2012) studied the CNG and LPG fuelled customized diesel engine respectively to investigate the performance-emissions features employing ANN architecture. Furthermore, Singh, Jain, and Mahla (2020) predict the emissions and performance parameters of biogas-biodiesel fuelled dual fuel engine using hybrid-based Artificial Neural Network-Harris Hawks and whale optimization algorithm.

The various parameters that were accessed by Syed et al., (2015) were torque, BSFC, HC, and CO emission whereas Yusuf et al., (2010) evaluated BTE, BP, torque, NOx, $CO_2$, EGT, and $O_2$. Furthermore, engine load, engine speed, and the mass flow rate were considered input parameters during their analysis. However, Paul et al., (2018) selected throttle position, mass air flow rate, manifold air pressure, fuel pulse width, ignition angle, engine temperature, air temperature, and air-fuel ratio as input variables. Furthermore, many engine experts adopted training algorithms such as trainlm, traingdx, trainrp, trainscg whereas logsig and tansig as transfer function were used to analyze the hidden layer. Moreover, Syed et al., (2015) varied neurons between 1 to 25, whereas Yusuf et al., (2010) varied neurons from 2 to 20 to form the network of the hidden layer.

Yusuf. al., (2012) distributed the data sets obtained from the network in 75:15:15 ratio for training, validation, and testing purpose whereas Ghobadian et al. (2010) utilized 80% of the data for training purposes, whereas 20% of the data was used to test and validate. However, Yusuf. al., (2010) utilized 70:30 ratio of experimental datasets for training and validating purposes. Wan Mansor et al., (2019) predicted emission results of a dual fuel engine with an accuracy of 93.47%. However, Uslu and Mustafa (2018) forecasted engine emissions and power output with an accuracy of 98.78%. Moroever, Singh, Jain, and Mahla (2020) concluded that (ANN-HHOWOA) yielded excellent results with a classification score of 98.6667%. Furthermore, Karthic and Kumar (2019) observed 97 % accurate results. Hence, it can be concluded that the predicted results obtained from ANN by various researchers were consistent with the experimental results. The empirical findings will help in optimizing the performance of the engine.

## 2.10 Outcomes of the Literature Review

Many kinds of literature have been analyzed on the dual fuel engine and techniques related to it during the research. It was found that the dual fuel combustion technique is one of the best combustion techniques used for reducing the emissions of compressed ignition engines without decreasing the power and thermal efficiency. As an outcome of the exhaustive review of literature, the following significant conclusions can be made-.

1. Most of the literature recommends that engines working on dual fuel mode have economic advantages as numerous alternative fuels can be utilized, which is less expensive than conventional petroleum fuels and requires no significant hardware changes. In addition to this, various combinations of liquid and gaseous fuel can be employed in DFE.
2. Various literatures reported that fossil fuels-based exhaust emissions, such as hydrocarbon (HC), carbon monoxide (CO), carbon dioxide ($CO_2$), and smoke/particulate, decrease appreciably at any loads due to less carbon present in gaseous fuel in contrast with neat diesel fuel.
3. Literatures suggested that the brake thermal efficiency (BTE) of the dual fuel engine increases at moderate to high loads, whereas the efficiency decreases at part load conditions.
4. Many engine specialists believe that incomplete combustion of gaseous fuel causes lower in-cylinder pressure in DFE than neat diesel fuel engines, primarily at lower loading conditions. However, with escalating load, the combustion characteristics of the gaseous fuel mixture were improved and effects in higher in-cylinder pressure of the dual-fuel engine, causing enhanced performance.
5. Many engine specialists have stated that the primary prerequisite of any gaseous fuel for DFE is that the gaseous fuel-air blend must not auto-ignite spontaneously. Neglecting to do so can prompt the excessive rise of pressure, high in-cylinder pressure, and overheating of the combustion chamber's wall, causing the engine's failure.
6. Most of the literature reported that unburned hydrocarbon (UHC) emissions, carbon monoxides (CO) inclined at lower loading conditions because of the insufficient combustion of gaseous fuel, eventually leading to the deterioration of the performance compared to petroleum diesel.

7. According to most of the literature, gaseous fuel utilization improved at higher load, which results in higher in-cylinder temperature. Hence, the dual fuel engine exhibits higher NOX emissions compared to the traditional diesel engine. It can be inferred that irregular flame formation at a higher energy ratio, restricted energy proportion of gaseous fuels, and $NO_x$ emissions are the significant challenges of a dual fuel engine.
8. The above-mentioned triple issues could be solved by adopting low-temperature combustion techniques that primarily include water injection, utilizing the oxygenated mixture, and decreasing the engine's compression ratio.
9. Some of the literature suggested that the use of oxygenated blends as pilot fuel improves the performance and emissions characteristics of DFE, which may be due to enhanced physico-chemical properties of pilot fuel at all loading conditions.
10. Some researchers have modified the intake charge conditions or fuel injection technique or optimized injection timings to enhance a dual fuel engine's combustion, performance, and emissions characteristics.
11. Three different techniques have been reported in the literature for utilizing gaseous fuel in DFE. They are Manifold injection, port injection, and direct in-cylinder injection. Among all these strategies, the manifold injection technique offers enormous advantages, like controlling gaseous fuel injection timing, better blending characteristics of fuel/air mixture, low-temperature exposure to the gas injector, and minor modification of the engine's hardware.
12. In general, the availability of CNG, LPG, and hydrogen (gaseous fuel) is still a matter of concern for meeting long-term worldwide energy demands. Besides, the engine running on methane gas ($CH_4$) for a long time may adversely affect the surroundings because of global warming.

## 2.11 Research Gap Analysis

On the basis of an exhaustive literature review, the following research gap was recognized.

1. According to the current literature review, most of the studies have been focused on CNG and hydrogen to be used as a primary fuel in dual fuel engines. However, limited research was found to use acetylene as the primary fuel in dual fuel engines.
2. The literature reported that a limited volume of work had been carried to improve the performance, combustion, and exhaust emission characteristics of dual fuel engine under low to medium loading conditions.
3. The literature suggests that little work has been carried out to study the impact of gaseous flow rate on the performance of dual fuel engine. However, to realize the full potential of gaseous fuel under dual fuel mode variation of quantity of gaseous fuel should be adequately explored.
4. As far as the authors are aware still, higher NOx emission is a massive challenge for dual fuel engine. Hence, reducing NOx emissions without compromising the performance needs to be investigated adequately.
5. The effect of blending oxygenated fuels such as n-butanol and di-ethyl ether (DEE) with pilot diesel on the performance, combustion, and emission characteristics has not been studied comprehensively in dual fuel engine.
6. Indeed, at present, the very little quantum of work is dedicated on predictive analysis of engine parameters using ANN technique in dual fuel engine. Hence, an extensive study of the ANN technique to analyze the performance of dual fuel engine is required.

## 2.12 Problem Statement

After the exhaustive literature review and identification of the research gap, it was concluded that dual fuel combustion is a promising technique for improving a conventional diesel engine's combustion and emission characteristics. The dual fuel approach has economic advantages because it is possible to use different alternative fuels. The pilot fuels are essential in dual fuel engines. The unutilized oxygenated blends such as DEE and n-butanol are identified and selected for pilot fuel use. The artificial intelligence tools (ANN) are used to evaluate and optimize the gaseous fuel flow rate. Very least explored, acetylene as a gaseous fuel was selected as a primary fuel. Therefore, exhaustive trials on dual fuel engines using n-butanol, DEE, and their blends with diesel in various proportions as pilot fuel for acetylene were carried to evaluate the effect of oxygenated blends combustion, performance, and emission characteristics. The present research work aims to utilize acetylene and oxygenated blends in a modified CI engine under dual fuel mode and compared the experimental results with petroleum diesel.

## 2.13 Research Objectives

The following objectives of the current research were envisaged, as mentioned below.

1. Exhaustive literature survey to utilize oxygenated blends such as n-butanol and DEE as a pilot fuel in acetylene fuelled CI engine.
2. To measure the physicochemical properties of n-butanol, DEE, and their blends and to compare the properties with petroleum diesel.
3. To develop a dual fuel engine test rig for conducting exhaustive experiments by varying the gaseous fuel's mass flow rate.
4. To determine the optimum quantity of gaseous fuel based on the performance and emission results.

5. To predict the performance and emission results of a dual fuel engine with the ANN approach.
6. To investigate the effects of n-butanol, DEE, and their blends on the performance, combustion, and emission parameters at an optimized flow rate of gaseous fuel.

# CHAPTER 3

## SYSTEM DEVELOPMENT AND METHODOLOGY

### Overview

*This chapter begins with the selection of fuels and preparation of a pilot fuel during experimentation. Furthermore, the determination of physicochemical properties of all pilot fuels is explained. Moreover, system development for engine testing is explained. Additionally, the ANN architecture for predicting the performance and emission characters of dual fuel engine is also explained.*

### 3.1 Selection of Fuels

The variety of literature has been evaluated in the current study to discover the essential, best combination of primary fuel as well as pilot fuel. Subsequently, the physicochemical properties like density, kinematic viscosity, cetane number, calorific value, and a flashpoint of all pilot fuels were estimated. In the wake of estimating all these properties, the combustion, performance, and emissions profile were evaluated in a dual fuel engine. The modification in conventional diesel was arranged for operating the customized engine with fuel flexibility, i.e., liquid and gaseous fuel. The engine components, properties, and advancement in the engine were explained thoroughly in this chapter.

The selection of primary fuel was a significant task in the current exploration. The choice must be made based on its combustion characteristics such as flame speed, auto-ignition temperature, adiabatic flame temperature, and calorific value. Accessibility, storage, and safety precautions must also be considered while selecting gaseous fuel. Most of the literature highlights acetylene's promising combustion properties compared to other alternative fuels. The fundamental reason behind choosing acetylene as primary fuel was, because of its cleaner and efficient burning affinity, similar combustion properties to hydrogen and less explosive than other available gaseous fuels, economical, renewable in nature, viz. can be easily produced from calcium carbide and water as well as from methane by the thermal cracking technique.

The major sources of calcium carbonate (CaCO3) are available abundantly in India. As mentioned earlier, the greater part of energy in a dual fuel engine is shared by gaseous fuel (primary fuel); however, its ignition is reliant upon pilot fuel. Hence the selection of pilot fuel in such circumstances turns out to be vital and noteworthy. A large portion of the research recommended that diesel utilization as pilot fuel gives improved outcomes at peak loads. However, at lower load, the performance under dual combustion mode is found to be inferior compared to neat diesel. Most of the work has been concentrated on assessing the effect of biodiesel/diesel blends on the performance of dual fuel engine.

It has been reported that performance gets deteriorated, and a rise in exhaust emission levels is also observed with the addition of biodiesel/diesel blends. However, very few researchers have utilized oxygenated additives such as alcohols, ethers, and esters blended with standard diesel to achieve complete combustion and resolve pre-ignition and knock problems in a customized diesel engine employing a dual fuel technique. Hence, in the present study, n-butanol and DEE were selected as oxygenated additives blended with pilot diesel to improve the combustion characteristics of a dual fuel engine. The reasons for selecting fuels are highlighted in table 3.1.

**Table 3.1:** Selection criteria for primary as well as pilot fuels

| | **Acetylene** | **n-butanol** | **DEE** |
|---|---|---|---|
| Production | By the hydrolysis of calcium carbide<br>By thermal cracking of liquid hydrocarbons/ methane gas | From the fermentation of carbohydrates known as ABE fermentation | By the distillation of ethanol with sulphuric acid (acid ether synthesis). |
| Advantages | Higher energy content, high flame speed, short quenching distance, and high diffusivity | Higher lower heating value than ethanol, better miscibility, enhanced blending stability, less prone to moisture, lower carbon build-up in injectors, and less hazardous and renewable bio-fuel | High cetane number (>125), low autoignition temperature, high oxygen content, reasonable energy density, broad flammability limits, high miscibility with diesel fuel, and renewable biofuel |
| Sources | CaCO3Rajasthan (21%) MP& AP(11% each), Chhattisgarh & Karnataka (10% each), Gujarat, TN i& Telangana (8% each), Maharashtra & HP(4% each) | From waste materials,<br>cereal crops, sugar cane, and sugar beet, etc., also produced from cellulosic raw materials | Biomass |

## 3.2 Preparation of Test Fuels (Blends)

The increased rates of research on complex fuel blends in engine applications pose a need for more efficient and accurate fuel blending processes in engine laboratories. The fuel blending system was evaluated in terms of functionality, safety, accuracy, and repeatability. A total of nine pilot fuel samples were prepared, such as D100, BBU05, BBU10, BBU15, BBU20, BDEE05, BDEE10, BDEE15, and BDEE20 respectively were used in this study. DEE and n-butanol were purchased from a local supplier, which was labeled 99.0% pure. Moreover, diesel was purchased from a local petrol pump. To check the miscibility, various proportions of n-butanol and DEE were added to mineral diesel and monitored for 60 days. Test fuels were regularly checked for phase separation and homogeneity, No signs of separation were observed. The test fuel samples were prepared by volume-wise substitution of DEE and n-butanol in the diesel. Blending was done using rigorous agitation at a high rpm hand by a blender. The nomenclature of the fuel sample considered in the present work is shown in table 3.2.

**Table 3.2:** Nomenclature of tested fuels

| **Nomenclature** | **Test Fuel Composition** |
|---|---|
| **ADF2** | Acetylene @2 lpm + diesel as pilot fuel |
| **ADF4** | Acetylene @4 lpm + diesel as pilot fuel |
| **ADF6** | Acetylene @6 lpm + diesel as pilot fuel |
| **ADF8** | Acetylene @8 lpm + diesel as pilot fuel |
| **BBU05** | Diesel 95% + n-butanol 5% |
| **BBU10** | Diesel 90% + n-butanol 10% |
| **BBU15** | Diesel 85% + n-butanol 15% |
| **BBU20** | Diesel 80% + in-butanol 20% |
| **BDEE05** | Diesel 95% + DEE 5% |
| **BDEE10** | Diesel 90% + DEE 10% |
| **BDEE15** | Diesel 85% + DEE 15% |

| | |
|---|---|
| **BDEE20** | Diesel 80% + DEE 20% |
| **D100** | 100% Diesel (Baseline data) |

### 3.3 Physico-Chemical Properties of pilot fuels

The various properties like kinematic viscosity, density, calorific value, cetane index, flash point temperature of baseline diesel, n-butanol/diesel blends, and DEE/diesel blends, were determined by using ASTM standard. The details of the equipment used to measure physico-chemical properties are shown in Table 3.3.

**Table 3.3:** Instrument used for determining the physicochemical properties

| Properties | Measuring Device | Techniques | Manufacture | Operating Range | ASTM Standard |
|---|---|---|---|---|---|
| Kinematic Viscosity | Capillary Tube - High Temperature | Time for the flow of the fuel sample in the capillary tube | Petrotest | + 5°C to +150°C | D445 |
| Density | Density meter | Oscillating U-tube | Anton Paar | 0 g/cm3 to 1.5 g/cm3 | D4052 |
| Flash Point | Automatic Flashpoint Tester | The sample was heated in a test cup | Pensky-Martens | up to 405 °C | D93 |
| Calorific Value | Bomb Calorimeter | Combustion of fuel at constant volume in the presence of oxygen | Parr | 52 to 12000 calorie | D240 |

### 3.3.1 Density

The density meter "DMA 4500 Anto Paar model" was used to measure the density of all pilot fuel during the present study. This instrument is based on the principle of the U-tube oscillation concept. The U-tube oscillation method is preferred for determining the density of liquids and gases by evaluating the frequency of oscillation. The spring-mass system is used to measure the frequency. Plate 3.1 displays the density metre used in the present analysis. The test fuel pipeline was rinsed by injecting 10 ml of toluene into the sample injection port before injecting the sample. Then, a 10 ml sample was inserted through the injection port into the instrument. The procedure for one pilot fuel was replicated at least five times and the average of this calculation was eventually taken as the final value.


**Plate 3.1:** Density meter

### 3.3.2 Kinematic Viscosity

Viscosity is a very important fuel property that can be defined as a measure of the resistance of a fluid that is either deformed by shear stress or tensile stress. Viscosity indicates a fluid's intrinsic resistance to flow and can be interpreted as a measure of fluid friction. In general, scum and deposits on cylinder walls, piston tip, etc. tend to form due to the higher viscosity of the fuel. Hence, it is desirable that there should be minimum fuel viscosity. As per the specification provided in ASTM D445, the various blend samples are prepared by using a kinematic viscometer at 40 °C. The viscometer consists of a capillary tube, which is filled with the sample to be tested. There are two labels marked on the capillary tube. The time for the flow of fuel sample from upper mark to lower mark is measured and kinematic viscosity is calculated using the time taken for each sample. Plate 3.2. shows the plate of the kinematic viscometer apparatus.

The kinematic viscosity of different fuel blends can be calculated as:

$$\upsilon = k \times t$$

Where,
$\upsilon$= kinematic viscosity of sample;

k = constant for viscometer = 0.005675 $mm^2/sec^2$;

t = time taken by the fluid to flow through the capillary tube.

**Plate 3.2:** Viscometer

### 3.3.3 Calorific Value

The calorific value is measured in terms of the amount of heat produced when the unit mass of fuel is completely combusted in a calorimeter under specified conditions. The higher the heating value of the fuel is the total heat produced in kJ per kg or $m^3$. During the combustion process, all fuels containing hydrogen in usable form can react with oxygen to form steam. The resulting steam can condense if the combustion materials are cooled to their original temperature. Thus, maximal heat is abstracted. This heating value is known as the higher heating value. The calorific value of the fuel was calculated as per the ASTM D240.0 specification with the Isothermal Bomb Calorimeter. The combustion takes place in the presence of oxygen at a constant volume in a fully enclosed tank. The fuel samples were then burned in a bomb calorimeter, determining the calorific value of all the samples. The Parr Model 6100EF was used in the laboratory to measure the caloric content of diesel/oxygenated blends. Plate 3.3 displays the Bomb Calorimeter used for measuring the calorific value.


**Plate 3.3:** Bomb Calorimeter

### 3.3.4 Flash Point

The flash point is the lowest temperature at which the oil vapour forms an ignitable mixture when combined with air and produces a momentary flash when a brief pilot flame is added (Sinha et. al, 2007). As seen in Plate 3.4, the Pensky-Martens Mechanical Flash Point apparatus measured the flash point of the test fuel in line with the ASTM D93 standard. The sample was heated in a test cup until the tiny pilot flame was aimed into the cup through the opening created at the top cover at regular intervals. The minimum flash point value defined by EN-14214 and ASTM D-6751 is 120°C and 130°C, respectively.

**Plate 3.4:** Flash Point Apparatus

### 3.5 Long Term Storage Stability Test

Diesel fuel blends stability can refer to storage stability at ambient conditions for extended periods or to thermal stability with short exposure to higher temperatures in end use equipment. Deterioration of the stability of diesel fuels manifests in many ways like color change, development of gum or particulate matter followed by insoluble sediment formation. Changes in the physical and chemical properties of the fuel are also possible. Gum and sediment can cause fuel filter plugging and various engine operating difficulties temperature, oxygen, and storage time play a major role in fuel stability. High temperature, high oxygen content, exposure to light, or prolonged storage time can promote oxidation of the fuel. Also, some metals like copper accelerate fuel aging reactions. Generally, the whole picture of fuel stability is very complex.

Each fuel can have its own mechanism which differs from other fuel types. The type and severity of refining processes affect the stability. However, some polar heterocyclic compounds are known to be deleterious to stability. Nevertheless, many studies agree that instability is attributed to oxidation and/or auto-oxidation reactions. Hydroperoxides are the initial reaction products of the oxidation of hydrocarbons. Nucleation and agglomeration of oxidized products cause sediment formation during storage. The method used in the current study is the visual inspection technique to evaluate the long-term storage stability of diesel fuels. The sample is tested at accelerated conditions: 50 °C for 13 weeks. The colour and the amount of insoluble material of the sample are determined three times during the test period (beginning of the test, after 6 weeks, after 13 weeks). The samples were kept in open bottles in dark conditions. It is believed that the 13-week stability C test correlates with storage at ambient conditions for about one year.

### 3.6 Experimental Set-Up to Measure the Ignition Delay

The ignition delay becomes a very important property in order to maximize the combustion and emission properties of either fuel or its mixture. So, in the present study, a small testing rig was fabricated to test the ignition delay. It consists of a cylinder of stainless steel with a 2l size. The thickness of the selected cylinder was 10 mm to preserve it at a higher pressure and temperature. The ends of the cylinder were closed by stainless steel plates 25mm thick. The welding was carried out between the plates and the cylinder in such a manner that it endured higher pressure and ensuring that the welding was leak-proof. One side of the cylinder, which functions as an intake valve and exhaust valve, was fitted with two needle valves. The air was supplied by an air compressor to the combustion chamber at differing pressures. To test the in-cylinder pressure, a pressure gauge was mounted.

Two electrical heaters with a power of 1.5 kW and 400 W were used to heat the air inside the cylinder. The heater of greater capacity was wrapped outside the cylinder while the heater of limited capacity was placed within the combustion chamber. To calculate the in-cylinder temperature, a K-type thermocouple was also mounted which was attached to a digital temperature controller. The arrangement is done in such a manner that, as the in-cylinder temperature exceeds the desired point, it immediately cuts the power supply of the heaters. The fuel injection system consists of a Bosch fuel injector, the Bosch fuel injection pump, the fuel tank, and the fuel injection mechanism to supply the fuel. A piezoelectric sensor was mounted on the oscilloscope-connected to fuel supply tubing (FLUKE 190-202 oscilloscope with a multimeter). On one side of the combustion chamber, a glass window was provided to detect the combustion.

Moreover, a photosensor was mounted in front of a glass window to detect the ignition. On the other side of the combustion chamber is connected with a hot-surface plate of stainless-steel opposite to the fuel injector. The fuel injector has a pintle nozzle to form a hollow conical spray that impinges on the hot surface. The hot-surface plate and the tip of the pintle-type nozzle were roughly 64 mm apart. A heating coil is placed between two stainless steel plates in the interior of the combustion chamber to expand the exterior temperature of the plates. The heating coil is connected with a transformer's auxiliary coil, which supplies the arrangement of plates and coils with the current. The temperature sensor used in the current work estimates the hot surface temperature inside the combustion chamber of the stainless-steel plate and is a computerized temperature controller. As the temperature goes above a certain limit, it reduces the flow of current to the radiator and hence maintains the temperature of the hot plate below a fixed limit.

The deeply compact air was introduced into the combustion chamber with the help of an air compressor. To sustain the pressure of the combustion chamber, a two-stage high-pressure reciprocating air compressor was used. The pressure generated inside the combustion chamber will vary in the range of 1-25 bars with the assistance of a control valve. A fuel pump consists of a single-acting plunger that reciprocates within a barrel with a continuous stroke. The clearance between the barrel and the plunger is just 2-3 thousandths of a millimetre, which offers perfect sealing without unusual pressing, even at extremely high pressure and low speeds. The basic prerequisite of this investigation depends on the controlling mechanism of the fuel pump. This is to be accomplished by supplying the pump's plunger with reciprocating motion. There are three elements of the whole mechanism: the pump disc, the hitting pin, and the rocker and lever handle. A digital oscilloscope is used in the present inquiry, as seen in Figure 3.1. The ID can be obtained on the oscilloscope screen by changing the time scale by examining the difference between the beginning of the injection and the beginning of combustion.

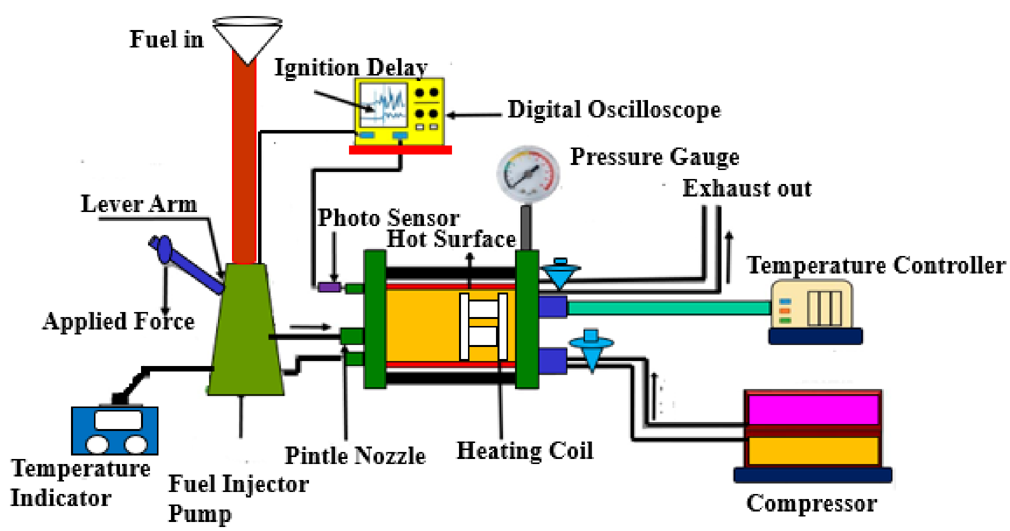


**Figure 3.1:** Ignition delay measurement setup

### 3.7 Selection of Engine

The engine used for the present assessment was a Kirloskar manufactured 4 stroke, DI-CI engine running at a steady speed of 1500 rpm. This engine was air-cooled and commonly used for agricultural purposes in India. The cylinder of the engine was made of cast iron and fitted with a cast iron-liner. The lubrication system employed in this engine setup was wet sump type, and oil was supplied onto the driving shaft for frictionless operation. An overhead camshaft was used to regulate both valves (inlet and exit) controlled by the driving shaft. For loading purpose, an alternator was coupled to the engine cylinder. Moreover, for measuring the engine speed rpm sensor (magnetic type) was connected with the alternator. The details of the engine used for experimentation are mentioned in Table 3.4.

**Table 3.4:** Specifications of the Experimental Test Rig

| **Manufactured by** | **Kirloskar** |
|---|---|
| Model | CAF8 |
| Engine Type | Single Cylinder, Vertical, CI, 4 Stroke |
| BP | 5.5 kW |
| CR | 17.5:1 |
| Rated Speed | 1500 rpm |
| Type of cooling | Air Cooling |
| Bore diameter | 87.5 mm |
| Combustion System | Direct Injection |
| Stroke Length | 110 mm |
| Piston | Hemispherical |
| Cubic Capacity | 0.661 Ltr |

| | |
|---|---|
| Injection Angle | $23^{o}$ BTDC |
| Injection Pressure | 220 bar |
| Lubrication System | Forced Feed |
| Starting | Hand Start with a cranking handle |
| Fuel oil | Commercial High-Speed Diesel |
| Governor | Centrifugal Type |
| **Alternator Specification** | |
| Manufacturer | Kirloskar Electric Co. Ltd., India |
| Alternator Type | Single-phase, 50 Hz, |
| **Valve Timing** | |
| Inlet valve opening | $5^{o}$ CA BTDC |
| Inlet valve closing | $35^{o}$ CA ABDC |
| Exhaust valve opening | $35^{o}$ CA BBDC |
| Exhaust valve closing | $5^{o}$ CA ATDC |

An engine test rig was developed to conduct the experimental trails on the customized CI engine. The acetylene gas was introduced into the intake manifold with a mass flow rate of 2 LPM, 4 LPM, 6LPM, and 8 LPM by the standardized gas flow meter purchased from local vendors, which ranged from 1 - 12 LPM. Initially, an optimized induction length was determined at a minimum flow rate of the acetylene, and it was found that 50 cm. For any gaseous fuel operating under dual fuel mode, it is essential to provide a homogeneous air-gas proportion into the combustion chambers for proper burning and vary the air-gas stream based on the required performance. Hence a slight modification was carried out to install a dedicated gas mixer to the intake manifold end as a passage to introduce acetylene with fresh air into the engine cylinder.

The inlet pressure of acetylene was maintained to sub-atmospheric conditions by a double-stage pressure regulator. The quantity of acetylene was increased manually until the modified engine gives the indications of misfiring. It restrains the extreme flow rate of acetylene during the dual fuel mode of operation. It was noticed that engine speed was slightly elevated because of additional energy shared by acetylene. The engine variables were computed manually as well as the data were verified through an engine soft database installed on a personal computer. Furthermore, investigation on the combustion, emission, and performance characteristics was done at various loading conditions in single fuel and dual fuel modes. The schematic of the engine test rig is shown in Figure 3.2.

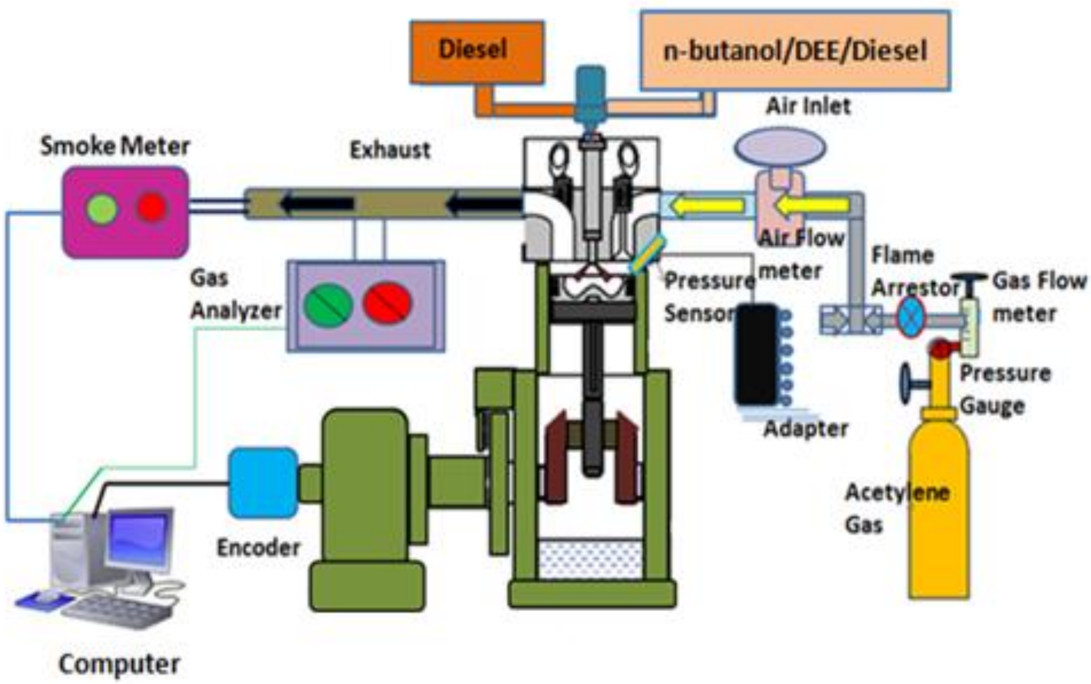


**Figure 3.2:** Schematic diagram of the engine test rig

## 3.8 Gas Flow Arrangement

The gas flow arrangement consists of an acetylene cylinder, pressure regulator, gas flow meter, and flame arrester. All the components are explained below.

### 3.8.1 Acetylene Cylinder

The acetylene gas cylinder comprises the double-stage pressure regulator, flame arrestor, gas flow meter, and induction pipe. It was purchased from a local supplier, and the cylinder contains seven cubic meter of gas compressed in acetone. The purity of gas claimed by the Sigma gas and services in their quotation is 99.99% pure. The gas cylinder was maintained to operate below 15 bar, ensuring safety concern. The actual gas cylinder is shown in Plate 3.5.

**Plate 3.5:** Acetylene gas cylinder

### 3.8.2 Pressure Regulator

The primary function of a double-stage pressure controller is to reduce impractical excessive higher pressure from the gas cylinder to a lower working discharge pressure. Two-phase regulators are deliberately calibrated to bring down the higher pressure in two phases. They require less modification and gives a progressively steady discharge pressure regardless of changes in inlet pressure. They are outstandingly most suitable for higher pressure purpose such as acetylene. Two-phase regulators have two controllers incorporated with the same body. The principal controller is a non-customizable, and to decrease the intermediate pressure, the subsequent regulator is flexible, which can change the discharge pressure within the desired limit. In the present study, the pressure regulator was conjoint between a high-pressure cylinder and a low-pressure induction pipe. Plate. 3.6 shows the actual pressure regulator used in the experiment.


**Plate 3.6:** Double stage pressure regulator

### 3.8.3 Flow Meter

The flowmeter controls and displays the rate of the gas flow in LPM. A gas flow meter was purchased from a local vendor to gauge the mass flow rate of acetylene with which it can induct through the intake manifold. The flowmeter used in this present study is manufactured by Japsin industrial instrumentation, New Delhi. It consists of a valve adjusted by a control knob; a tapered glass tube inscribed with numbers that correspond to gas flow rates; and a float, which rises within the tapered glass tube to indicate the flow rate of gas. As the valve is opened, gas flows into the tapered tube, and the float rises within the tube until the gravitational force acting on the float is balanced by the force of gas entering the bottom of the tube. The more gas that is allowed to enter the tube, the higher the float rises within the tube. The gas passes between the float and the walls of the tube exit from the top of the flowmeter and is delivered to the engine's intake manifold. Flowmeter was attached to the flame arrestor. This gas flow meter range lies between 1 LPM and 12 LPM with the least count of 1 LPM. The measurement error mentioned by the supplier was 1%. The gas flow meter is depicted in Plate 3.7.

**Plate 3.7:** Gas Flowmeter

### 3.8.4 Flame Arrester

A flame arrester or fire arrester is an instrument used in current research for preventing the backflow of the acetylene gas supplied from the cylinder. It will safeguard the equipment against fire and physical damage from the explosion. There are two primary types of flame arrestors, i.e., dry and wet type out of this dry is preferable and dry type flame arrestor is used in this experiment. To ensure safety and smooth experimentation, flame arrestors are equipped with pressure as well as temperature sensitive cut-off valves that have been used in the present study. Temperature sensitive cut-off valve stops the flow of gaseous fuel when a predetermined temperature is exceeded whereas pressure cut-off valves stop the flow of gas when there is backpressure in the direction of opposite to flow. This device's response rate is speedy, and it can quickly quench the flashback in minimal time. The flame arrester is shown in Plate 3.8.

**Plate 3.8:** Flame Arrester

### 3.9 Selection of Engine Test Parameters

The selections of appropriate test parameters are a vital part of engine research. The engine test was done as specified by IS: 10000. The main parameters desired from the engine were selected. Various engine parameters are observed and the calculated parameters.

Observed parameters are enlisted below.

- Engine load
- Engine speed
- Fuel consumption and Air flow rate
- In-cylinder pressure.
- Emissions of HC, CO, and NOx
- Exhaust temperature
- Smoke opacity

Calculated parameters are given below.

- Brake mean effective pressure (BMEP)
- Brake thermal efficiency (BTE)
- Brake specific energy consumption (BSEC)
- Diesel Substitution Rate
- Mass fraction burnt rate (MFB)
- Heat release rate (HRR)

### 3.10 Measurement Methods and Calculations

For data measurement, auto-zoom plots, and digital rendering of data in the system, windows-based Engine software was used. All the inputs obtained through various sensors were processed with the data acquisition system (DAQ) system and converted into digital form. Further, it maintains an infinite number of graphs with facilities to transfer data on an Excel Document. This system can calculate the complex P-θ diagram, heat release rate, and combustion pressure. The detailed measurement techniques for various engine parameters are described below.

### 3.10.1 Load Measurement

The engine load influences the engine's ability to generate power. Generally, the load is the force that opposes the power which the engine generates. Each engine is developed to operate at a specific speed for a given load called the rated load or full load. The laboratory test is carried at different loads; thus, an effective and convenient method for calculating loads is essential. On an engine test rig, a alternator is used to quantify the power produced by an engine. The alternator used in this study is AC type with an air-cooling system. For rapid loading, this offers the benefit of a faster rate of load adjustment. A rotor, shaft, bearings, shell, and bedplate make up the electrical alternator assembly.

The rotor is mounted on a shaft that connects with bearings. As a direct current is supplied, the magnetic field is generated in the casing through the air gap on either side of the rotor, allowing the rotor to rotate into this magnetic field. Overall, the alternator consisted of a shunt-wound AC generator and a load resistance unit whereas, the control panel includes an amp-meter, voltmeter, and switch fuse carrier. It is installed on the engine test rig to compute the power absorbed. The load ranged from 0 ampere to 22.5 ampere in steps of 4.5 ampere. Light sources were connected through the electrical load panel, and the voltage of 220 V was set using a voltage regulator. Plate 3.9 depicts the electric bulb used for loading.

**Plate 3.9:** Electric Bulb used for Loading purpose

### 3.10.2 Measurement of Speed

The engine speed is calculated through an electromagnetic pickup combined with a data-indicator fixed with the panel board. A magnetic pickup with pins is placed at the circumference of the engine flywheel. The signals produced are directed to the display unit that shows the speed in revolutions per minute (rpm). Plate 3.10 shows a speed sensor that is used to measure speed.

**Plate 3.10:** Speed Sensor

### 3.10.3 Measurement of Exhaust Gas Temperature (EGT)

The exhaust gas temperature is measured with a K -type (Chromel/Alumel) thermocouple attached to the engine's exhaust manifold. K-type thermocouples can measure the widest temperature range of -270°C to 1260°C at 6.4 to 54.9 mV output, which is one of the key benefits of k - type thermocouple over other thermocouples or other temperature sensors. They are widely used because of their strong corrosion resistance, durability, and higher accuracy. The capacity to work under extreme environmental environments and in diverse atmospheres renders it a favorite over another temperature transducer. The data acquisition panel board connected to the engine test rig shows the exhaust gas temperature reading. Moreover, an electronic device was also used for measuring room temperature, and it was periodically calibrated. Plate 3.11 illustrates the thermocouple attached to the exhaust end of the engine.

**Plate 3.11:** Thermocouple attached to the exhaust manifold

### 3.10.4 Measurement of In-cylinder Pressure

The in-cylinder pressure was determined by a piezoelectric transducer made by PCB Piezotronics that was water-cooled. Using an appropriate charge amplifier, the transducer's charge output was amplified into an equal voltage signal. The transducer was fixed on the surface of the cylinder head to eliminate passage impact. A charge output is generated by the piezoelectric transducer, which is proportional to the in-cylinder pressure. The device was monitored regularly for drift and corrected accordingly. Using DAQ and a personal computer, the pressure crank angle sequence for 100 consecutive cycles was also recorded. These data were further processed to obtain the average pressure crank angle variations. Plate 3.12 shows a pressure transducer mounted on the head of the engine.

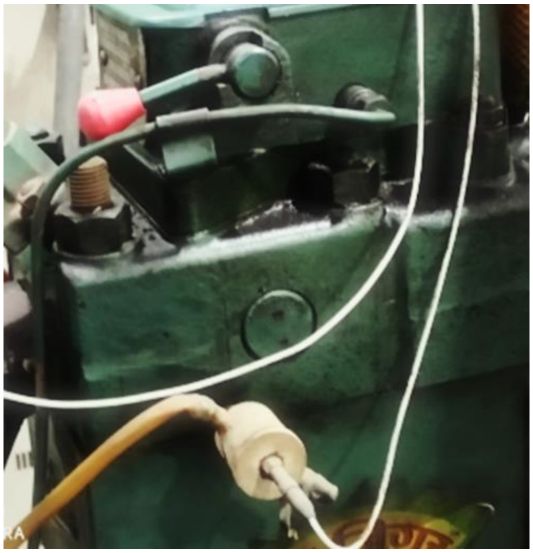

**Plate 3.12:** Pressure transducer attached to the engine

### 3.10.5 Measurement of Crank Angle Position

A Kubler-made electro-optical sensor was installed to indicate TDC's position by providing a voltage pulse exactly when the TDC position was reached. This sensor includes a pair of infrared diodes and a phototransistor, such that when the infrared rays emitted from the diode fall onto the phototransistor, it is not disturbed. To get the correct signal when the piston approaches TDC exactly, a slight cut was created to a continuous disc at the TDC location. The phototransistor output voltage increases to 5 volts at this instant, and it is zero for all the other points. The optical sensor's voltage signals were fed to the analog-digital converter and from there to the data acquisition system for recording the pressure versus crank angle position. Plate 3.13 depicts the crank angle encoder.

**Plate 3.13:** Crank angle encoder

### 3.10.6 Air Flow Measurement

The air flow was measured using an air sensor (turbine type flow meter) installed inside the control panel. In principle, the turbine flow meters use the mechanical energy of the fluid to rotate a “pinwheel” (rotor) in the flow stream. Blades on the rotor are angled to transform energy from the flow stream into rotational energy. The rotor shaft spins on bearings. When the fluid moves faster, the rotor spins proportionally faster. Blade movement is often detected magnetically, with each blade or embedded piece of metal generating a pulse. The transmitter processes the pulse signal to the data acquisition system that determines the flow of the fluid. The air flow data was available in the sensor indicator of the control panel and the engine soft database. Moreover, there was another method available to validate the sensor data. It was based on the orifice and the air box method. The differential pressure across the orifice inserted in the air flow channel provides the air flow rate using the formula in equation 3.2.

**Air Consumption** (kg/s):

The mass flow rate of air can be calculated as given below:

The mass flow rate of air (ma) = Cd × Q × $\rho$a

$$Q = \sqrt{\frac{2gH\rho_W}{\rho_a}} \qquad (3.1)$$

Where Q = Volume flow rate of air.

Cd = coefficient of discharge of orifice.

Density of air $\rho$a = Patm /RT atm (kg /$m^3$)

H = difference in the manometric head in the water column.

$\rho_W$ = density of water.

A = Area of an orifice.

$$m_a = C_d . A\sqrt{2gH\rho_W . \rho_a} \qquad (3.2)$$

Cd = Co-efficient of discharge of orifice (0.65 in the present case)

$\rho a$ =1.225 kg/m$^3$

$\rho_W$ =1000 kg/m$^3$

Diameter of orifice meter= 0.015 m

g = 9.81 m/s

Sp. gravity of fuel = 0.86

### 3.10.7 Pilot Fuel Mass fuel consumption (mf)

The mass flow rate of pilot fuel is calculated by determining the time required to consume a fixed amount of fuel, i.e., ten cc through stopwatch. The consumption of the fuel under various loading conditions was obtained by using the following equation:

Mass of fuel consumption (kg/s)

$$m_f = \frac{Y_{CC} \times \rho}{T} \tag{3.3}$$

Where,

Ycc is 10 cc of volume consumed

T is the time taken in seconds

ρ is the density of the liquid fuel

Sp. gravity of fuel = 0.836

The engine performance is dependent upon several factors such as torque, brake power (BP), brake specific energy consumption (BSEC), brake mean effective pressure (BMEP), brake thermal efficiency (BTE), etc. These engine characters were calculated manually as well as the data were verified by Engine soft software. The various performance parameters are presented below:

### 3.10.8 Brake Power (BP)

The power delivered at the crankshaft is referred to as brake power. Usually, by attaching a power absorption unit to the engine drive shaft, the brake pressure is calculated. In the case of electrical loading, the brake power of the engine is calculated by using the formula given below:

$$BP(kW) = \frac{V \text{ x } I}{\eta_G \text{ x } 1000} \tag{3.4}$$

Where,

I = Ammeter reading in Amps.

$\eta_G$ = efficiency of the generator

V = Voltmeter reading in Volts.

BP = Brake power in kW.

### 3.10.9 Brake Mean Effective Pressure (BMEP)

Mean effective pressure is theoretical pressure, which is supposed to be acting on the piston during the power stroke. It is an important parameter that is used to measure an engine's performance and compare various engines. However, it does not represent the real pressures inside the combustion chamber. It measures the capacity of an engine to do work. BMEP is the mean effective pressure measured from the alternator power. It is the actual output power at the crankshaft.

Mathematically,

$$\text{BMEP (bar)} = \frac{\text{BP x 60 x } 10^3}{\text{L x A x N x n}} \tag{3.5}$$

Where L=stroke length

A= Area of crossection

N = Rotational speed of engine RPM

n= number cylinders

### 3.10.10 Brake specific energy consumption (BSEC)

BSFC is how much fuel is consumed to generate one kilowatt of brake power in one hour (i.e., power available at the engine shaft not produced power into the engine block). BSFC is the ratio of fuel consumption per hour to the brake power, whereas BSEC is the product of BSFC and the calorific value of a fuel. This implies the quality of fuel energy derived from the specified fuel. BSEC at various loading conditions was found by using the following equation:

$$\text{BSEC of normal Diesel}(\text{BSEC}_\text{d}) = \frac{\dot{\text{m}}_\text{d}\ x\ 3600}{\text{BP}} x\ CV_d \tag{3.6}$$

$$\text{BSEC of Dual Fuel Engine}(\text{BSEC}_\text{df}) = \frac{(\dot{\text{m}}_\text{d} + \dot{\text{m}}_\text{ace.})\ x\ 3600}{\text{BP}}\ x\ CV_{eq} \tag{3.7}$$

Where,

$\dot{m}_d$ is the mass of standard diesel consumed in kg/sec.

$\dot{m}_{ace.}$ is mass of acetylene consumed in kg/sec.

$CV_d$ is the calorific value of diesel

$CV_{eq}$ is the calorific value of the equivalent of dual fuel

BP is brake power in kW.

BSFC = Brake specific fuel consumption, g/kW–hr

### 3.10.11 Brake Thermal Efficiency (BTE)

The engine's brake thermal efficiency provides an understanding of the engine's output power from the heat supplied by the fuel. The BTE was calculated using the following equation at various loading conditions:

$$\text{BTE of Diesel Engine } (\eta_d\%) = \frac{BP}{\dot{m}_d CV_d} \tag{3.8}$$

$$\text{BTE of Dual Fuel Engine } (\eta_{df}\%) = \frac{BP}{\dot{m}_{ace.} CV_{ace.} + \dot{m}_d CV_d} \tag{3.9}$$

Where,

$CV_d$ is the Calorific value of diesel used, kJ/kg

$CV_{ace.}$ is Calorific value of acetylene used kJ/kg

### 3.10.12 Diesel Substitution rate

The diesel substitution rate was calculated by the given formula.

$$\text{Diesel Substitution } (\%) = \frac{\dot{m}_d - \dot{m}_{PF}}{\dot{m}_d} x\ 100 \tag{3.10}$$

Where,

$\dot{m}_{PF}$ is energy shared by diesel during dual fuel mode.

### 3.10.13 Energy shared by acetylene

The proportion of energy shared by acetylene id dual fuel engine is given by:

$$\text{Energy Share of Acetylene } (\%) = \frac{\dot{m}_{ace.}}{\dot{m}_{ace.} + \dot{m}_d} x\ 100 \tag{3.11}$$

### 3.10.14 Calculation of Heat Release Rate (HRR)

HR calculations are calculated by considering thermodynamic equilibrium during in-cylinder combustion. However, temperature gradients, pressure waves, nonequilibrium conditions, fuel vaporization, mixing, etc. are not considered. From the first law of thermodynamics:

$$\frac{dU}{dt} = \dot{Q} - \dot{W} \tag{3.12}$$

$$mC_V \frac{dT}{dt} = \dot{Q} - P\frac{dV}{dt} \tag{3.13}$$

$$PV = mRT \tag{3.14}$$

$$\frac{dT}{dt} = \frac{1}{mR}\left[P\frac{dV}{dt} + V\frac{dP}{dt}\right] \tag{3.15}$$

$$\dot{Q} = \left[\frac{C_V}{R} + 1\right] P\frac{dV}{dt} + \frac{C_V}{R} V\frac{dP}{dt} \tag{3.16}$$

$$\gamma = \frac{C_P}{C_V} \tag{3.17}$$

$$\frac{dQ}{d\theta} = \left[\frac{\gamma}{\gamma-1}\right] P \frac{dV}{d\theta} + \left[\frac{1}{\gamma-1}\right] V \frac{dP}{d\theta} \tag{3.18}$$

Where $\dot{Q}$ is combined heat release rate and heat transfer rate, $\dot{W}$ is the rate of work done by the system, U is the internal energy, P is the pressure, V is volume, T is temperature, R is the gas constant, m is the mass of gas, $\gamma$ is the ratio of specific heat, and θ is the crank angle.

### 3.10.15 Rassweiler and Withrow model for calculating Mass Fraction burnt

Burn rate analysis is commonly used with spark-ignition engines to determine the mass fraction burned. Rassweiler and Withrow developed a technique in 1938 that is still considered today to be both accurate and computationally efficient. During combustion, the pressure rise, ΔP, during a crank interval, Δθ, is considered to consist of pressure rise due to combustion, $\Delta P_c$, and pressure change due to change in volume, $\Delta P_v$.

$$\Delta P = \Delta Pc + \Delta Pv \tag{3.19}$$

As the crank angle increments from $\theta_i$ to $\theta_{i+1}$ the volume changes from $V_i$ to $V_{i+1}$ and the pressure from $P_i$ to $P_{i+1}$. Assuming that the change in pressure due to volume change can be calculated from a polytropic process of constant k:

$$P_{i+1} - P_i = \Delta P_c + P_i[(\frac{V_i}{V_{i+1}})^k - 1] \tag{3.20}$$

$$\Delta P_c = P_{i+1} - P_i(\frac{V_i}{V_{i+1}})^k \tag{3.21}$$

Because the combustion process does not occur at constant volume, the pressure rise rate due to combustion is not directly proportional to the mass of fuel burned. Therefore, the pressure rise due to combustion must be referenced to a datum volume, such as that at TDC, $V_{tdc}$.

$$\Delta P_c{}^* = \Delta P_c \frac{V_i}{V_{tdc}} \tag{3.22}$$

By identifying the end of combustion and the number of crank angle intervals between start and finish of combustion, N, the mass fraction burned can be calculated:

$$\text{mfb} = \frac{\sum_0^i \Delta P_c{}^*}{\sum_0^N \Delta P_c{}^*} \tag{3.23}$$

For the purpose of cycle-to-cycle analysis, the crank angle at which burn rate percentages of 1%, 2%, 5%, 10%, 20%, 25%, 50%, 75%, 80%, 90%, 95%, 98% and 99% mass fraction burned are determined. Additionally, the ignition delay and combustion duration are determined from mass fraction burned curves. The ignition delay is the crank angle between the start of combustion and typically 1,2 or 5% mfb. Combustion duration is calculated as the crank angle between the end of the ignition delay and typically 90, 95, or 99% mfb. Determining small or large percentages such as 1 or 99% can be difficult due to the susceptibility of the calculation to the effects of noise with small pressure changes. This method of calculating mass fraction burned was chosen because of its proven reliability and its widespread use. Also, research has shown that more complex models provide little additional accuracy.

### 3.10.16 Measurement of Exhaust Emissions

The exhalations such as NOx, HC, CO, and smoke levels are significant variables for evaluating the vehicular pollution. The AVL 444 di-gas analyzer and AVL 437-smoke meter were utilized to quantify the exhaust levels. Before taking measurements, the analyzer's probe was placed into the exhaust end of the engine. The readings from the gas analyzer and smoke meter were recorded at a steady condition. Initially, the NOx, HC, and CO levels were measured by AVL 444 di-gas analyzer, and then the smoke opacity was measured by AVL 437 smoke meter. The detailed information of both instruments is demonstrated in Plate 3.13. The computerized data acquisition system collects the data used to analyze engine exhaust gas emissions in excel format. Plate 3.14 shows the computer along with the di-gas analyzer.

**Plate 3.14:** Exhaust gas Analyser and computer

**Table 3.5:** Specifications of the di-gas analyzer and smoke meter.

| Instrument Name | Measurement Parameters | Measurement Technique | Resolution | Measurement Range |
|---|---|---|---|---|
| AVL Di- Gas Analyzer | CO | Non-dispersive infrared sensor (NDIR) | 0.01% Volume | 0-10% Volume |
| | HC | NDIR | 1ppm | 0-20000 ppm Volume |
| | NOx | Chemin-luminescence principle, electrochemical sensor | 1ppm | 0-10000 ppm Volume |
| AVL- Smoke Meter | Smoke Opacity | Photochemical | 0,1 % | 0-100 % |

### 3.11 Uncertainty Analysis

Holman technique is used to determine the errors caused by the errors that arise from the required instrument, experimental condition, and observation of results, evaluation, and interpretation of test results. All the instruments were tuned regularly. The precision of the measuring devices used, and the uncertainty of the results measured are given in Table 3.6. The total uncertainty percentage of this experiment was approximately two, which consists of numerous constraints.

The total percentage uncertainty of this experiment is calculated by:

$$= \sqrt{\begin{gathered}[(Uncertainity\ of\ BMEP)^2 + (Uncertainity\ of\ BTE)^2 + (Uncertainity\ of\ BSEC)^2 + \\ (Uncertainity\ of\ HC)^2 + (Uncertainity\ of\ CO)^2 \\ +(Uncertainity\ of\ NOx)^2 + (Uncertainity\ of\ Smoke)^2 + \\ (Uncertainity\ of\ Pressure\ transducer)^2 + \\ (Uncertainity\ of\ Crank\ Angle\ encoder)^2 + (Uncertainity\ of\ Speed\ sensor)^2 + \\ (Uncertainity\ of\ burrette)^2 \\ +(Uncertainity\ of\ stop\ watch)^2 + (Uncertainity\ of\ EGT)^2]\end{gathered}}$$

(3.24)

**Table 3.6:** Uncertainty Table

| Instruments | Measurement Technique | Range | | Accuracy | Uncertainties (%) |
|---|---|---|---|---|---|
| Pressure Transducer | Piezoelectric | 0 - 100 bar | | ± 0.1 bar | ± 0.20 |
| Crank Angle Encoder | Optical | 0 – 720$^{o}$ | | | ± 0.20 |
| Speed Measuring Unit | Magnetic pickup | 0-10000 rpm | | ± 10 rpm | ± 1 |
| Burette Measurement | Level indicator | 0-100 cc | | ± 0.1 cc | ± 1 |
| Gas Flowmeter | Positive displacement | 0-12 LPM | | ± 0.1 LPM | ± 1 |
| Digital Watch | Time in second | | | ± 0.6 s | ± 0.50 |
| Calculated BMEP | - | | | | ± 0.60 |
| Calculated BSEC | - | | | | ± 0.50 |
| Calculated BTE | - | | | | ± 0.60 |
| Exhaust Gas Temperature | K-type thermocouple | -270°C - 1260°C | | ± 1 $^{o}$C | ± 0.20 |
| Smoke-meter | Photochemical | 0-100% | | | ± 0.5 |
| | NDI | HC | 0 -10000 ppm | ± 20 ppm | ± 0.5 |

| | | | | | |
|---|---|---|---|---|---|
| Exhaust Gas Analyzer | NDI | CO | 0-10,000 ppm | ± 20 ppm | ± 0.5 |
| | Electrochemical | NOx | 0 - 5000 ppm | ± 50 ppm | ± 1 |

### 3.12 Baseline Tests

To ascertain a reason for the correlation of obtained results and to ensure the consistency of the test results, baseline data were prepared by running the engine on normal diesel mode. Initially, the engine was warmed and operated for at least 20-30 minutes for reaching a stable and steady-state. The engine was operated from no load (0%) to full load (100%) conditions in the steps of 20% for at least three times for each load. The engine was operating at a steady speed of 1500 rpm on pure diesel for collecting baseline data. The performance analysis was conducted using diesel as fuel. For the baseline test, the following detailed procedure was deployed:

- Switch on the electric panel for supplying electricity.
- Make sure the lubricating oil is proper in the engine.
- Verify the fuel level in the tank.
- Make sure the load is released, if any, on the alternator.
- Open the cock such that fuel flows to the engine.
- The engine should be started by cranking.
- Enable the engine to run for 20-30 minutes in idling conditions (no load) to ensure stable working conditions.
- Noting down all the performance parameters for a no-load condition via a data acquisition system
- Repeat the experiment and record the results by varying loads (from lower to higher and

vice versa a minimum of three times)

- The load should be released after completion of the readings and then stop the engine.

### 3.13 Acetylene Dual Fuel Tests

Experiments were performed to analyze the engine performance, combustion, and emission characteristics in the following ways:

- Testing the customized diesel engine under dual fuel mode with diesel as the pilot fuel during the manifold injection.
- Testing the customized diesel engine under dual fuel mode with diesel/n-butanol blends as the pilot fuel during the manifold injection.
- Testing the customized diesel engine under dual fuel mode with diesel/DEE blends as the pilot fuel during the manifold injection.

The experimental trials carried out during the current study is shown in table 3.7

**Table 3.7:** Test Matrix

| Variables | Fuel used | Analysis |
|---|---|---|
| **Normal operation** | | |
| Maintained constant load at no load, 20%, 40%, 60%, 80%, and 100% at the rated speed of 1500 rpm. | Diesel | Comparison of performance, emissions, and combustion characteristics of neat diesel fuel |
| **Manifold injection technique** | | |
| Maintained constant load at no load, 20%, 40%, 60%. 80% and 100% at the rated speed of 1500 rpm for various gas flow rates (2 LPM, 4 LPM, 6 LPM, and 8 LPM) up to the knock limit. | Acetylene + diesel | Comparison of performance, emission, and combustion characteristics of acetylene diesel with diesel fuel. |
| **Manifold injection technique with n-** | | |

| | | |
|---|---|---|
| **butanol/diesel blend** | | |
| Maintained constant load at no load, 20%, 40%, 60%, 80%, and 100% at the rated speed of 1500 rpm. Acetylene gas was maintained at a maximum flow rate of 4 LPM. 5 % to 20 % n-butanol diesel blends were selected as pilot fuel. | Acetylene + diesel+ n-butanol | Comparison of performance, emission, and combustion characteristics of n-butanol/diesel blends as a pilot source in acetylene dual fuel engine and neat diesel fuel. |
| **Manifold injection technique with DEE/diesel blend** | | |
| Maintained constant load at no load, 20%, 40%, 60%, 80%, and 100% at the rated speed of 1500 rpm. Acetylene gas was maintained at a maximum flow rate of 4 LPM. 5 % to 20 % DEE/diesel blends were selected as pilot fuel. | Acetylene + diesel + DEE | Comparison of performance, emission, and combustion characteristics of DEE assisted with diesel as an ignition source in acetylene dual fuel engine and neat diesel fuel. |

### 3.14 Construction of an Artificial Neural Network Model

In this current study, performance, and exhaust emissions of acetylene fuelled CI engine was predicted with the help of an artificial neural network (ANN) architecture. The backpropagation method with feed-forward networks was utilized as a learning algorithm of ANN. This network is equipped with five input parameters, single hidden layers, and six output parameters. The tansig activation function was utilized to evaluate the model, whereas the Levenberg-Marquardt method (trailm) was used to train the data set. The model was scaled between −1 to 1 to ensure that the inputs weigh equal contribution to the network. The ratio of 70:15:15 data was used for training, testing, and validation purposes.

Mean square error (MSE) function and regression correlation (R) was selected as the objective function in the current study. The MSE determines the average square difference between the observed and expected output. A value closer to 0 reveals the optimal performance for the network. The R coefficient determines the intensity of the relationship between measured and predicted values. An amount that is equivalent to 1 implies a strong relationship. The selected input parameters for optimizing performance were variable gas flow rate (2,4,6 and 8 LPM), variable loading condition (0,20,40,60,80 and 100%), diesel substitution rate (DSR), energy shared (%) by liquid fuel as well as a gaseous fuel. The output parameters were BSFC, BTE, EGT, HC, CO, and NOx.

Figure 3.3 shows the architecture of ANN considered during the present analysis. Finally, the regression analysis was carried out between the network's response and the corresponding targets to analyze the network reaction in more depth. The MATLAB 2018 computer software was used for the construction of ANN by exploring the neural network toolbox in the command window. The performance index (MSE) is calculated by the given formula:

$$MSE = \frac{1}{n}\sum_{i=1}^{n}(M_i - P_i) \qquad (3.25)$$

Where $M_i$ is the measured value, and $P_i$ is the predicted value.

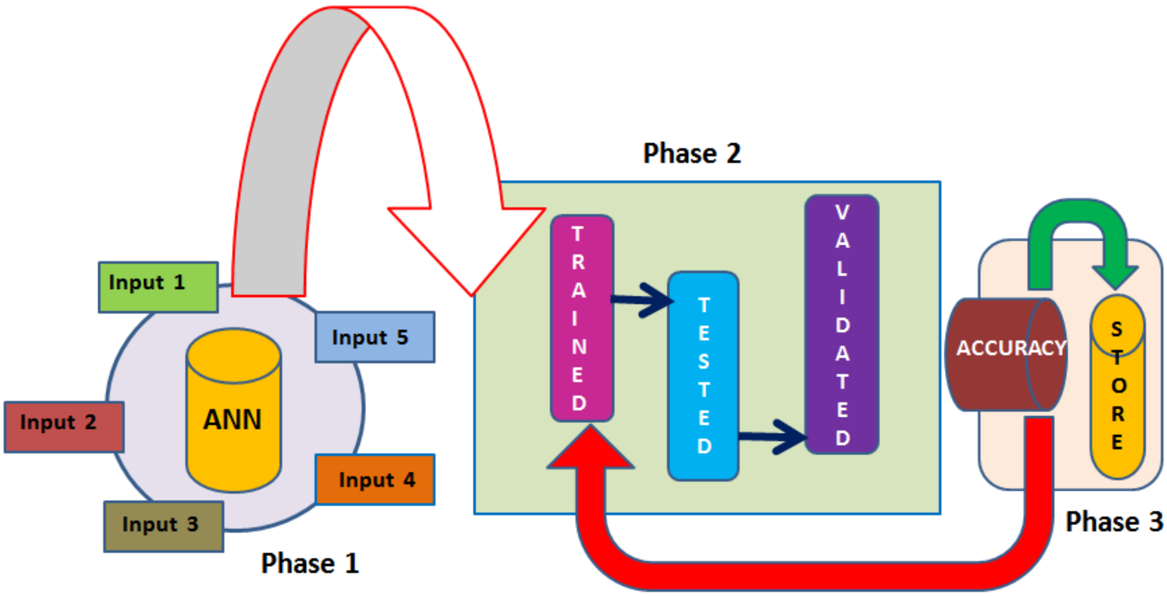


**Figure 3.3**: ANN Architecture

Step by step procedure for predicting the engine variables are mentioned below:

- In the MATLAB workspace, two variables, input, and target, were named.
- Data were imported from an excel sheet prepared from the original experiment.
- The input and target data were transposed.
- Using the ANN toolbox in the command window, the input, output, and target were assigned.
- The network was created with a feed-forward backpropagation algorithm.
- Training function -trainlm, adaption -learngdm and performance function-MSE was selected with a number of neurons-10
- After creating the network, training of the network was done with 1000 epochs. A performance plot was obtained.
- If the training graph is in descending order, i.e., MSE is low, it can be confirmed that the

nature of data is accurate, and all the data are fit for consideration.

- At last, predicted results are obtained by exporting the data from the validated network.
- Finally, the actual and predicted value were compared.

CHAPTER 4

# RESULTS AND DISCUSSION

## Overview

*In this chapter, initially, the physicochemical characteristics of pilot fuel blends are determined. Then experimental observations of the acetylene fuelled CI engine under dual fuel mode are discussed at a variable mass flow rate. At first, neat diesel is utilized as pilot fuel under dual fuel combustion mode to get the optimized flow rate. Furthermore, the software analysis through the ANN technique is carried to verify the experimental results. Finally, the experimental trials are conducted with oxygenated blends, namely DEE/ diesel and n-butanol/ diesel blends, as pilot fuel at the optimized flow rate of gaseous fuel for various loading conditions. For the same power output, a comparative assessment of the combustion, performance, and emission characteristics of diesel and a dual fuel engine is also carried.*

## 4.1 Physicochemical Characteristics of Pilot Fuel Blends

The physico-chemical properties (the kinematic viscosity, density, calorific value, flash point, and oxygen content) of neat diesel (D100), BU100 DEE100, BBU05, BBU10, BBU15, BBU20, BDEE05, BDEE10, BDEE15, and BDEE20 were evaluated according to ASTM standards.

### 4.1.1 Density

Fuel density represents an important fuel property with respect to volumetric fuel economy and maximum power. The density of the liquid is defined as mass per unit volume. The fuel density affects engine calibration and power as the mass of fuel injected per stroke varies with fuel density. Moreover, high-density fuels also have a higher viscosity thus, influence injection characteristics. Subsequently, the fuel density affects engine combustion and emissions characteristics. Hence density of fuel must be considered while evaluating engine performance. In the present study density of diesel was found 836 kg/m$^3$ whereas the density of DEE/diesel and n-butanol/diesel blends was found to be lower than standard diesel. The density of BDEE05, BDEE10, BDEE15, BDEE20, BBU05, BBU10, BBU15 and BBU20 were 829 kg/m$^3$, 823 kg/m$^3$, 817 kg/m$^3$, 811 kg/m$^3$, 834 kg/m$^3$, 831 kg/m$^3$, 829 kg/m$^3$ and 827 kg/m$^3$, respectively. The variation of density for all pilot fuels is shown in Figure 4.2

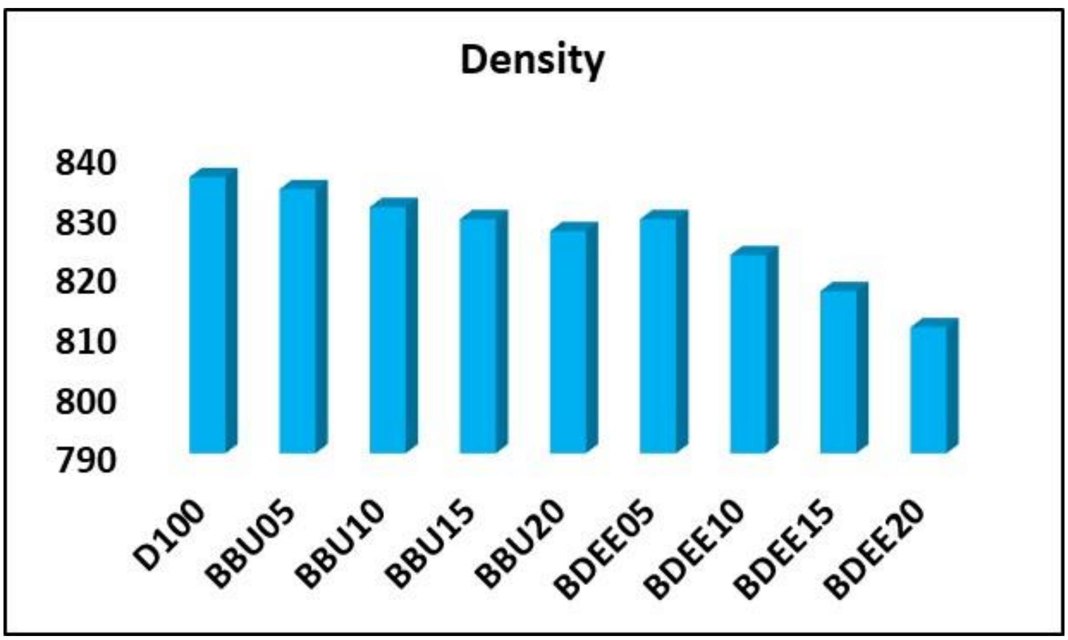


**Figure 4.1:** Variation of density for all pilot fuels

### 4.1.2 Kinematic viscosity

For any fuel kinematic viscosity is an important property. Kinematic viscosity is the internal resistance of a fluid to flow. It affects the injection as well as atomization characteristics. The kinematic viscosity of fuel is the function of temperature. It decreases with increasing temperature. The fluidity of any fuel is a concern especially at a lower temperature because they have high viscosity at a lower temperature. The main concern of the liquid fuel is the higher kinematic viscosity so it must meet the norms. Many times, it is observed that because of the higher viscosity of liquid fuels premature injector fouling takes place causing inferior atomization, which further results in higher soot emissions. The high kinematic viscosity can cause choking inside the fuel supply manifold. It is found that DEE/diesel blends and n-butanol/diesel has a lower kinematic viscosity compared to diesel (2.72 $mm^2/s$). The kinematic viscosity of all pilot fuels is shown in Figure 4.2.

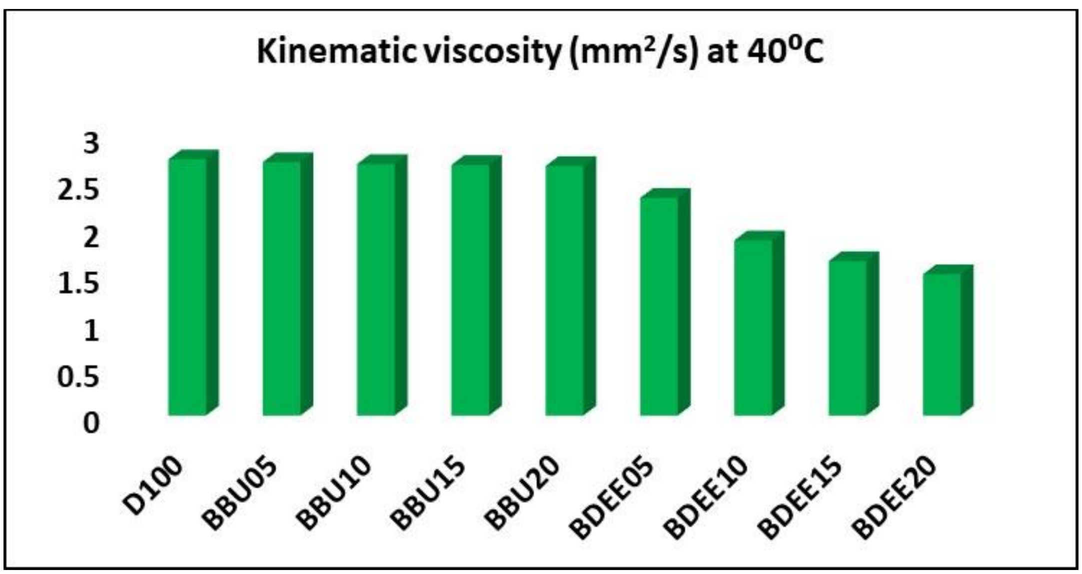


**Figure 4.2:** Variation of kinematic viscosity for all pilot fuels

### 4.1.3 Calorific value

Calorific value is the amount of energy released or produced when 1 kg of fuel is burnt in the presence of oxygen and the products of combustion are cooled to STP. Its SI unit is kJ/kg. If water is present in the products of combustion as a liquid, then the ‘higher calorific value’ (HCV) is obtained. If the water is present as a vapour then the ‘lower calorific value’ (LCV) is obtained. Most commonly lower heating value of fuels was used in internal combustion engines. The calorific value of all pilot fuels was measured with the help of a bomb calorie meter, whose working principle was already explained in an earlier section. Diesel fuel has the calorific value of 42.7 MJ/kg which was the highest among all pilot fuels. The calorific value of BDEE05, BDEE10, BDEE15, BDEE20, BU05, BBU10, BBU15 and BBU20 was 42.3 MJ/kg, 41.8 MJ/kg, 41.4 MJ/kg 40.9 MJ/kg, 41.2 MJ/kg, 40.7 MJ/kg, 40.3MJ/kg and 39.8 MJ/kg, respectively. It was observed that DEE/diesel blends have a higher calorific value than n-butanol/diesel blends. The variation in the calorific value of all pilot fuel is shown in Figure 4.3.

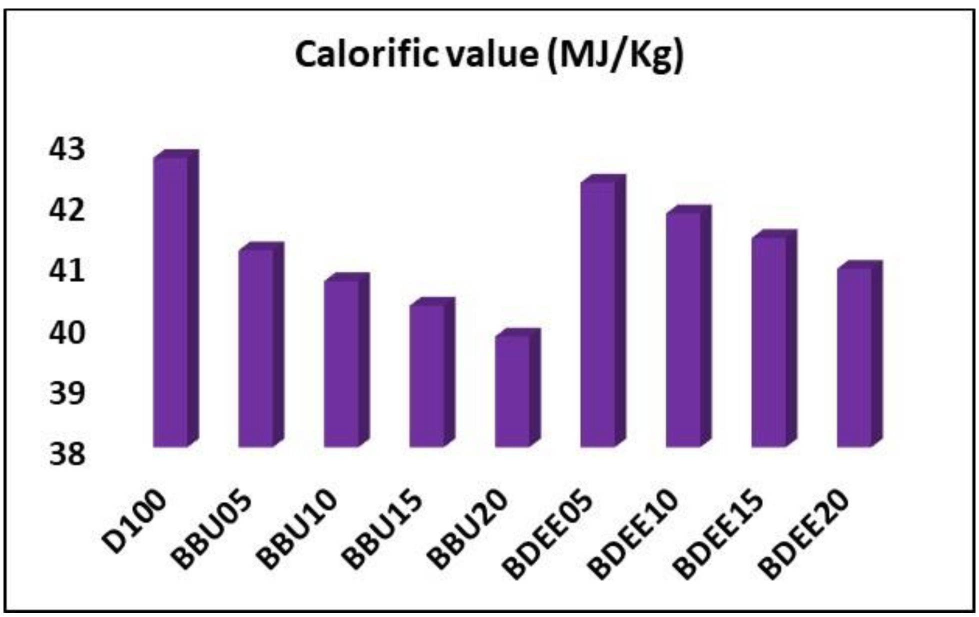


**Figure 4.3:** Variation of calorific value for all pilot fuels

### 4.1.4 Flash point

The flash point is the lowest temperature at which vapours arise from oil ignites. It indicates the maximum temperature at which a fuel can be stored without serious hazard. Flash point is a significant property not for the operability of a diesel fuel, but its storage and handling. Diesel fuels are classified as nonvolatile fuels, and their storage does not need specific precautions. Flash point is an excellent indication of diesel fuel contamination with more volatile products. The flash point of pilot fuel was measured as per ASTM standard. The flash point of neat was determined to be 70 ° C, which was higher among pilot fuels. The flash points of BDEE05, BDEE10, BDEE15, BDEE20, BBU05, BBU10, BBU15 and BBU20 were 67 ° C, 65 ° C, 63 ° C, 62 ° C, 69 ° C, 68 ° C, 67 ° C and 66 ° C, respectively. The comparative result on flash point of all pilot fuels is shown in Figure 4.4.

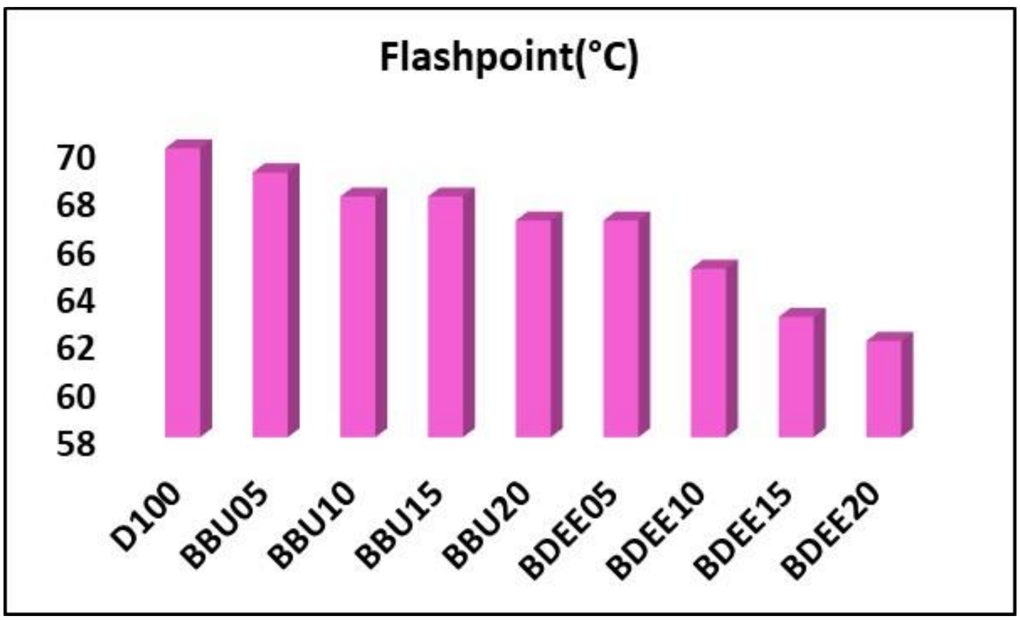


**Figure 4.4:** Variation of flash point for all pilot fuels

### 4.1.5 Cetane number

The cetane number (CN) is an indicator of the ignitibility of diesel fuels. It provides information about the ignition delay, i.e. the speed of self-ignition of diesel fuel when injected into hot air through the fuel injector. Cetane number is based on two compounds, namely hexadecane with a cetane of 100 and heptamethylnonane with a cetane of 15. In the CI engine, the cetane number is authoritative for describing the quality of the combustion process. Higher cetane numbers mean shorter ignition delay and at the same time a better performance of the diesel engine. However, if the ignition delay is high, for example, due to a low cetane number, most of the injected fuel will burn explosively. This may lead to the typical combustion noise called "knocking".Hence cetane number is a very valuable parameter while examining the performance of the diesel engine. It was found that DEE/diesel blends have a higher cetane number followed by diesel and n-butanol/diesel blends. The cetane number of diesel, BDEE05, BDEE10, BDEE15, BDEE20, BBU05, BBU10, BBU15 and BBU20 were 51, 54.6, 58.1, 61.7, 65.3, 49.6, 48.3, 47.2 and 45.9, respectively. The cetane number of all pilot fuel is shown in Figure 4.5.

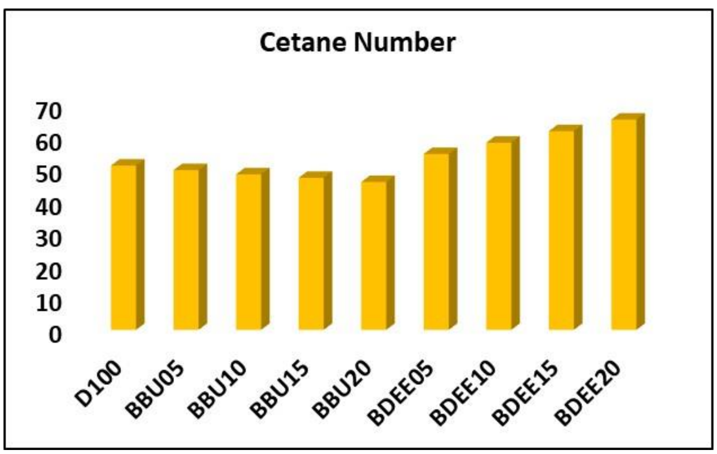


**Figure 4.5:** Variation of cetane number for all pilot fuels

### 4.1.6 Oxygen Content

Oxygen-enriched combustion is one of the attractive combustion technologies to control pollution and improve combustion in diesel engines. Moreover, increasing the oxygen content in the air leads to faster burn rates and increases the combustibility at the same stoichiometry (oxygen-to-fuel ratio). These effects have the potential to increase the thermal efficiency and specific power output of a diesel engine. Hence in the present study oxygenated fuels were used to enhance the combustion characteristics. The oxygen content was calculated by the mathematical relation explained in the appendix. The oxygen content (%) of BDEE05, BDEE10, BDEE15, BDEE20, BBU05, BBU10, BBU15 and BBU20 were 0.93, 1.87, 2.83, 3.8, 1.05, 2.1, 3.15, 4.21 and 0.93, respectively. The oxygen content of all pilot fuel is shown in Figure 4.6.

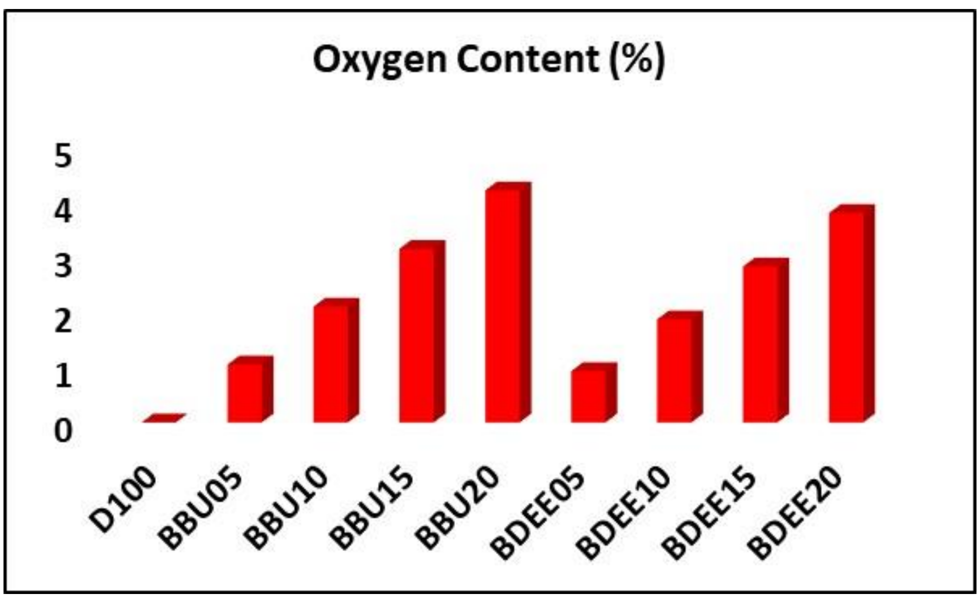


**Figure 4.6:** Variation of cetane number for all pilot fuels

All the physicochemical properties of pilot fuels and their impact on the performance are determined in the "Centre for Advanced Studies and Research in Automotive Engineering

(CASRAE), Delhi Technological University (DTU)," New Delhi. The comparison of the physico-chemical properties of diesel and blend is represented in Table 4.1.

**Table 4.1:** Physicochemical properties of pilot fuels

| **Parameter** | **D100** | **BU100** | **DEE100** | **BBU05** | **BBU10** | **BBU15** | **BBU20** | **BDEE05** | **BDEE10** | **BDEE15** | **BDEE20** |
|---|---|---|---|---|---|---|---|---|---|---|---|
| Density @15ºC (Kg/m$^3$) | 836 | 810 | 713 | 834 | 831 | 829 | 827 | 829 | 823 | 817 | 811 |
| Kinematic viscosity (mm$^2$/s) at 40ºC | 2.72 | 2.63 | 0.23 | 2.69 | 2.67 | 2.66 | 2.65 | 2.31 | 1.86 | 1.64 | 1.50 |
| Calorific value (MJ/Kg) | 42.7 | 33.1 | 33.9 | 41.2 | 40.7 | 40.3 | 39.8 | 42.3 | 41.8 | 41.4 | 40.9 |
| Cetane number | 51 | 25 | 125 | 49.6 | 48.3 | 47.2 | 45.9 | 54.6 | 58.1 | 61.7 | 65.3 |
| Flashpoint(°C) | 70 | 35 | -45 | 69 | 68 | 68 | 67 | 67 | 65 | 63 | 62 |
| Oxygen Content (%) | 0 | 21.6 | 21.6 | 1.05 | 2.1 | 3.15 | 4.21 | 0.93 | 1.87 | 2.83 | 3.8 |

**4.2 Optimizing Flow Rate of Acetylene**

The combustion, performance, and emission characteristics of the acetylene dual fuel engine are investigated. The experimental results are further examined and compared with neat diesel at various loading conditions. Based on the experimental outcome, the optimum flow rate of acetylene is determined.

### 4.2.1 Combustion Analysis

The typical manifestation of combustion is essentially the appearance of the visible flame seen by an open eye in the combustion chamber. Combustion is an exothermic chemical reaction. There are four different phases of combustion. They are ignition delay, rapid combustion (premixed combustion, controlled combustion (diffusion combustion), and late combustion stage. The first stage of combustion is called ignition lag or ignition delay period. It is the time required for preparing the air-fuel combustible mixture. When fuel is injected inside the cylinder, it does not burn instantaneously because of its liquid nature and fuel composition. It requires a finite time interval to dissociate into fine particles and to vapourize. In the CI engine, the region where ignition occurs is changing abruptly; hence combustion is difficult to control. A tremendous amount of heat is generated due to the increased temperature of the compressed charge, causing preignition leading to detonation or knocking phenomena at higher output.

Hence the speed of the main flame is essential to avoid knocking issues. Factors affecting the flame propagating are air-fuel ratio, inlet pressure, and turbulence. When the flame moves relatively in a uniform manner, it is called controlled combustion. Though most of the time distribution of fuel-air mixture is non -uniform where some amount of fuel is still in liquid form, which is being injected through the nozzle, and because of liquid inertia, it is impossible to convert all fuel droplets into a gaseous state instantly. Hence combustion in the CI engine depends upon the local state of fuel-air mixture, local pressure, and temperature of the charge. Rapid combustion is strictly due to rich fuel, which has time to be evaporated and mixed with air during the delay period. The rate and extent of burning during this period are closely associated with the length of the delay period and its relation to the injection process. The last stage is combined with controlled combustion and late combustion.

It is the period from the maximum pressure rise to the point where the combustion is measurably completed. In a dual fuel engine, two phases of combustion are observed, first phase is due to the combustion of liquid or pilot fuel whereas the second phase is the combustion of gaseous fuel initiated by igniting liquid fuel. Hence, the rapid burning of a gaseous-air fuel mixture is seen in the premixed stage of a dual fuel engine due to earlier combustion. The reason may be due to the increased number of artificial ignition centres during dual fuel mode. When the piston moves gradually in a downward direction, the pressure and temperature start to reduce, and a fresh charge is no longer fresh; it is exhausted. Thus, controlled combustion takes place during expansion work and thus power output is extracted.

#### 4.2.1.1 Heat Release Rate

In a conventional diesel engine, only the air is sucked during suction stroke, and this particular quantity of air is further compressed to attain higher pressure and temperature. At the end of the compression stroke, the diesel fuel is sprayed inside the engine cylinder through the nozzle. The fuel particles must be reasonably small to penetrate inside the air. Moreover, the heat generated by the compressed air ignites the injected fuel. Further combustion of the air-fuel mixture takes place, generating uncontrolled heat energy, which further increases the pressure and temperature of the engine cylinder rapidly. The HRR is calculated by the thermodynamics equation, which is explained in the appendix.

However, in a dual fuel engine, combustion is influenced by pilot fuel, ignition lag, compression ratio, and gaseous fuel flow rate (Lakshmanan and Nagarajan 2011b). The fluctuation of the heat release rate (HRR) by varying the crank angle is portrayed in Fig. 4.7. It is found that when the engine is operated from zero to partial loads, the acetylene fuel-air mixture is rich, which causes abnormal combustion, which further increases HC and CO emissions at low loading condition. Furthermore, when the load is raised to the maximum level, the charge quantity is slightly lean, and the burning rate is more stable than lower loads. Due to the stable combustion characteristics at higher loads the performance of both diesel and dual fuel engine improves (Behera, Murugan, and Nagarajan 2014).

Moreover, when the flow rate of acetylene is increased from 2 LPM to 8 LPM in a step count of 2 LPM, the combustion pattern is irregular, which attributes to higher flame speed and higher calorific value of the acetylene as compared to standard diesel. It is further noticed that the delay period of pilot fuel is also affected due to the advanced ignition of acetylene gas, which again confirms the rapid combustion initiated by gaseous molecules (Srivastava et al. 2017). The maximum HRR for acetylene DFE at 2 LPM, 4 LPM, 6 LPM, and 8 LPM gas flow rate is observed 57 J/$^o$CA, 60 J/$^o$CA, 63 J/$^o$CA, and 66 J/$^o$CA respectively at full load.

Furthermore, for neat diesel the HRR is observed slightly lower i.e., 52.2 J/$^o$CA at full load condition, which suggests diesel fuel burnt progressively compared to acetylene gas (Sudheesh and Mallikarjuna 2015). Moreover, a neat diesel engine shows a dominant diffusion stage compared to a dual fuel engine which further shows the improvement in the performance of the single-mode combustion engine. Similar results have been stated by various researchers (Saravanan et al., 2007; Basha et al., 2016). The major reasons for increased HRR during dual fuel mode are instantaneous combustion and more fuel burned during the premixed burning period leads to an increase in the heat release rate (HRR) at the beginning of combustion and a faster rate of pressure rise. It is clearly seen from Fig. 4.7 that with an increase in the flow rate of acetylene from 2 LPM to 4 LPM, HRR is closer to TDC due to increased accumulation of fuel during the relatively longer delay period resulted in a higher rate of heat release.

However further increasing the flow rate of acetylene to 6 LPM and 8 LPM the HRR peak shifts away from the TDC, owing to a longer ignition delay which will lead to more premixed fuel available at the start of the premixed burning period lead to excessive rates of energy release finally causing a high rate of pressure rise as mentioned by Behera, Murugan, and Nagarajan 2014 in their study. Hence it can be concluded that due to rapid HRR during acetylene induction, combustion temperature and engine cylinder pressure increases that further escalates NOx emissions, which is explained during emission analysis. Few engine experts have also stated the reasons for higher HRR at an elevated flow rate are due to the very high flame propagation speed of gaseous air mixture (Karim et al., 2010). The continued increase of the gaseous fuel concentration leads to knocking which further reduces the thermal efficiency of the dual fuel engine as explained in performance analysis.

At a higher flow rate, the preignition reaction of the gaseous fuel around the pilot fuel region becomes so active that once the pilot ignites, combustion takes place both inside and outside the pilot fuel and gaseous fuel can then burn so rapidly and almost simultaneously that extremely high energy release rates and sharp pressure rise result. Hence, the heat release rate for acetylene induction shows a significant premixed combustion phase, followed by a marginally decreased diffusion combustion phase. Subsequently, it can be concluded that HRR depends on various factors such as ignition delay, the start of combustion, a fraction of fuel burnt in the un-controlled combustion stage, and a difference in the combustion rate of a pilot and liquid fuels.

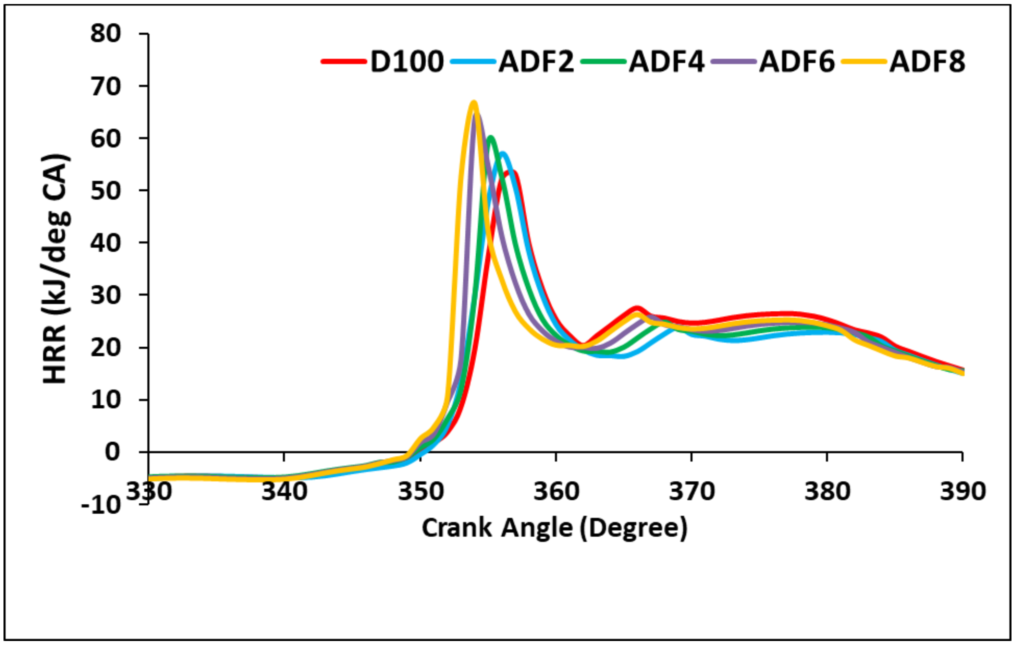


**Figure 4.7:** Variation of HRR with the change in the crank angle (Deg)

#### 4.2.1.2 Cylinder Pressure

The pressure of the engine cylinder explains the combustion characteristics in the cylinder body, and it is directly proportional to the engine load. The pressure versus crank angle graph is plotted with the help of the information recorded in the DAQ system using a pressure transducer and crank angle encoder. Most of the gaseous fuel is burnt in the premixed phase of combustion which causes a rapid combustion rate (Lakshmanan and Nagarajan 2011b) eventually increasing the peak pressure of the cylinder. Generally, when the piston is moving in an upward direction, the pressure, as well as temperature, is very high and the charge accumulated in the cylinder is still fresh, which generates higher flame (Srivastava et al. 2017). Due to rapid combustion, higher heat will be released, causing higher cylinder pressure in the case of a dual fuel engine.

When the gas is supplied at low loads, the fuel mixture tends to be rich, causing incomplete combustion, and some of the fuel goes un-utilized, causing emissions glitches at lower loads (Lakshmanan and Nagarajan 2011c). Furthermore, when the load is increased at an optimum level (80%), the combustion is improved considerably i.e., efficient utilization of gaseous fuel-air mixture occurs. At this instant, the cylinder pressure may go on rising due to a higher heat release rate, and the performance could be improved due to the complete combustion of charge present in the combustion chamber compared to partial load. Fig. 4.8. shows the variation of the change in-cylinder pressure with the varying crank angle at high load. It is found that with the increased flow rate of acetylene, the pressure inside the combustion chamber rises abruptly because of higher heat released during the premixed combustion phase.

This will cause higher thermal losses through the conduction of gaseous molecules and consequently, the performance of the dual fuel engine deteriorates at a higher flow rate of acetylene. The in-cylinder pressure for acetylene engine is 72 bar, 74 bar, 79 bar, and 80 bar respectively at 2 LPM, 4 LPM, 6 LPM, and 8 LPM of acetylene induction. For neat diesel, the peak cylinder pressure is slightly lower than all experimented fuels, i.e., 69 bar. Similar outcomes have been observed by many researchers (Behera, Murugan, and Nagarajan 2014 and Sudheesh and Mallikarjuna 2015). The peak pressure is observed at 1 to 3$^{o}$ CA ATDC for 2 LPM and 4 LPM whereas for 6 LPM and 8 LPM it is found 2$^{o}$ to 3$^{o}$ CA BTDC due to earlier start of combustion during the cycle by cycle and advancement in energy release.

It is also observed that at a higher flow rate of acetylene (6 LPM and 8 LPM) engine has a knocking tendency, and slight vibration is also noticed. Thus, optimum quantity of gaseous fuel should be injected for improving the performance of dual fuel engine and as a matter of concern, the optimization of flow rate has been studied. However, vibration analysis has not been studied in the present research but can be considered future work. The peak pressure rate principally depends on the rate of combustion in the initial stage and the fuel-air mixture taking part in the uncontrolled combustion stage. The increased burn rate of acetylene leads to higher in-cylinder temperatures and pressures during combustion. This can lead to excessive rates of energy release, causing a high rate of pressure rise.

At a higher flow rate of gaseous fuel, the excessive heat release occurs due to lengthening of ignition delay, which causes instantaneous pressure rise. Hence, the acetylene-air mixture is ignited prior to TDC which causes a part of the fuel energy being wasted in countering the compression work, hence there is a considerable reduction in effective work during the power stroke for dual fuel engine in comparison to conventional fuel. Thus, the performance of dual fuel engine deteriorates in comparison to neat diesel.

Moreover, when acetylene gas is supplied at 4 LPM, performance and emissions trends are comparable to that of standard diesel, which again suggests that relatively stable combustion takes place at 4 LPM because peak pressure is slightly closer to TDC in comparison to 6 LPM and 8 LPM. Overall. It can be concluded that the rate of pressure rise is higher for acetylene operated dual fuel engine compared to diesel operated engine due to the increase in ignition delay of diesel fuel. The advancement in peak pressure for acetylene combustion may be due to the instantaneous combustion of acetylene compared to diesel fuel (Lakshmanan and Nagarajan 2011c).

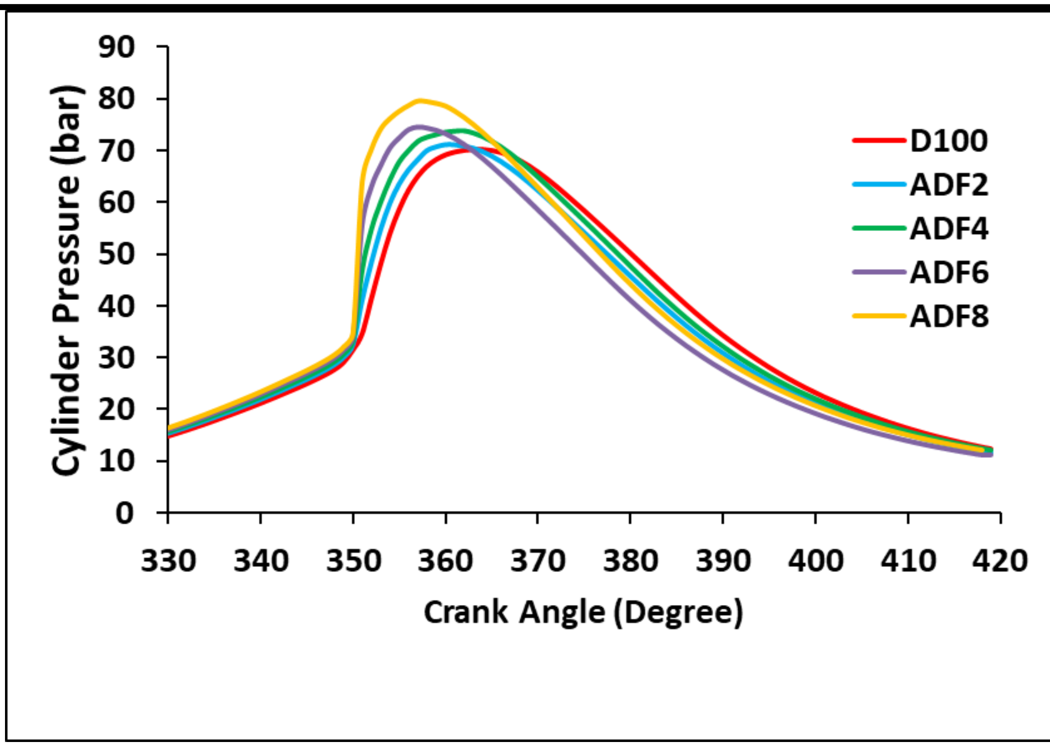


**Figure 4.8:** Variation of cylinder pressure with the change in crank angle (Deg)

#### 4.2.1.3 Mass fraction burnt

The mass fraction burnt characterizes the amount of fuel burnt during the combustion process. It is calculated by using Rassweiler and Withrow model, which has been used by many researchers. The differences in mass fraction burnt at full load with the change in the crank angle are demonstrated in Fig. 4.9. It is found that the mass fraction burnt for dual fuel engine is lower in comparison to neat diesel. The reason may be due to the complex combustion phenomena of dual fuel engine which is quite different than conventional diesel fuel. The instantaneous burring of acetylene causes abrupt combustion, and the earlier energy release rate caused a rise in peak pressure of the engine cylinder.

The calorific value of acetylene is higher than neat diesel which causes higher HRR away from TDC however, the combustion of primary fuel is dependent on liquid fuels. Thus, with the increase in the flow rate of gaseous fuel ignition delay increases for pilot diesel which is the reason why combustion is inefficient during dual fuel engine at almost every load. Hence at the beginning of the combustion, the mass fraction burnt for standard diesel fuel is higher because of the lower auto ignition temperature of diesel in comparison to acetylene; however, the difference is reduced substantially during the commencement of power stroke because of higher combustion temperature more ignition centres are formed to combust gaseous fuel molecules.

Overall, during gaseous fuel induction, a higher mass fraction burnt is observed at 4 LPM, which may attribute to stable flame propagation, causing improved performance compared to other flow rates. The mass fraction burnt for diesel is approximately 99%, whereas, for acetylene dual fuel engine at 2 LPM, 4 LPM, 6 LPM, and 8 LPM, it is around 96%, 97%, 94%, and 93%, respectively at optimum load. Similar observations are stated by many engine scientists (Behera, Murugan, and Nagarajan 2014; Srivastava et al. 2017).

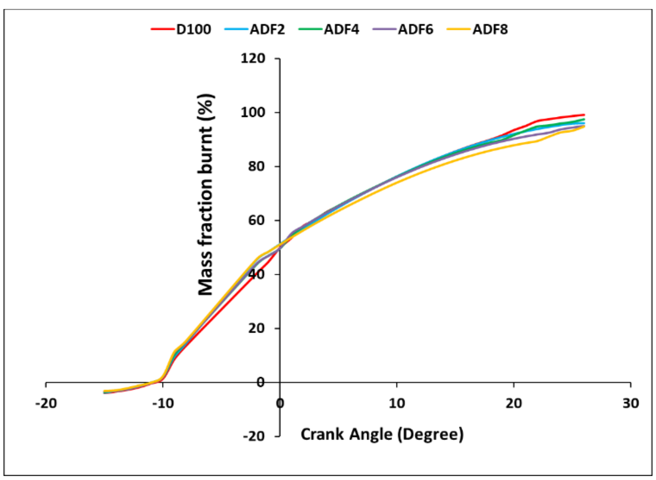


**Figure 4.9:** Variation of mass fraction burnt with the change in crank angle (Deg)

#### 4.2.1.4 Ignition Delay

The ignition delay (ID) period is defined as the time interval between fuel injection and the ignition of fuel (Banapurmath et al., 2014). Fig. 4.10. portrays the deviation of ID with change in load (%). The delay period shortens with a rise in load; it signifies that as the load is increased, pressure and temperature escalate, leading to complete combustion at higher loads in comparison to partial loads (Wei and Geng, 2016). Hence, the reduction of ID attributes to the higher combustion temperature and dilution of exhaust gas at a higher load. The ignition delay increases for pilot fuel due to the presence of gaseous fuel in the air, which affects the combustion during dual fuel mode. Hence, when the flow rate of acetylene is increased from 2 LPM to 8 LPM ignition delay is prolonged.

Moreover, due to the fuel-air mixture that gets accumulated in the combustion chamber during the time between fuel injection and the start of combustion undergoes rapid combustion and releases heat rapidly. This leads to steep pressure rise which can be clearly seen in the pressure-crank angle graph and may cause rough running of the diesel engine, accompanied by knocking condition at higher mass flow rate. Similar results have been observed by many engine experts (Karim, 2015a). The ignition delay becomes higher for dual fuel engine than that of standard diesel for intermediate loadings whereas a reverse trend was observed for higher loads. The reasons behind the reduction of the ignition delay with increasing flow rate may be due to overlapping of valve openings and high diffusion rate of acetylene at higher loads (Behera, Murugan, and Nagarajan 2014).

Engine scientists have also stated that the increase of cylinder pressure, temperature and equivalence ratio are the reasons for reduced ignition delay at full load. For diesel engine delay period varied from 16º to 11º from no load to full load conditions. For dual fuel engine, ignition lag is observed 12.2º, 12º, 11.5º, and 11º at 2 LPM, 4 LPM, 6 LPM, and 8 LPM when the engine is operated at 80% load. The results are consistent with previous research work carried by Nagarajan and Nagalingam, 2008. Few engine experts have also noticed that the ignition delay of dual fuel engine depends not only on the type of gaseous fuels and their concentrations but also on charge temperature, pressure, and oxygen concentration (Lata and Mishra, 2011).

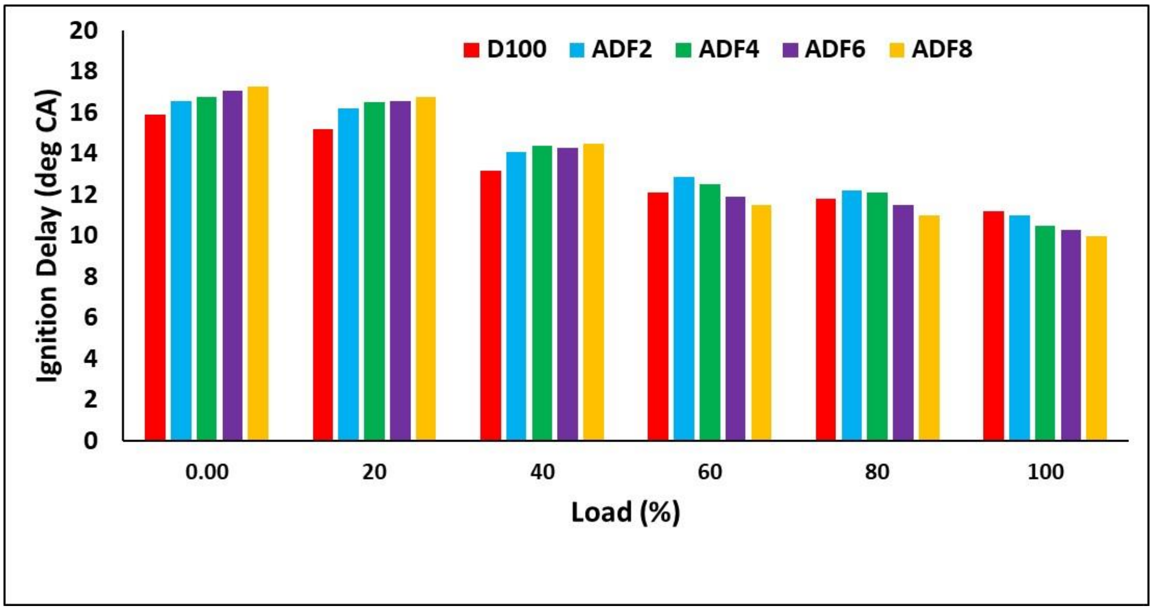


**Figure 4.10:** Variation of ignition delay with the change in load (%)

### 4.2.2 Performance Analysis

The performance of a dual fuel engine is evaluated by measuring various parameters such as BTE, BSEC, and EGT under different loads. The results are further compared with that of neat diesel.

#### 4.2.2.1Brake Thermal Efficiency

The variation of BTE with the change in BMEP is depicted in Fig. 4.11. It is noticed that BTE is increasing with the inclination of loads in all cases because of the lean air-fuel ratio and improved quality of charge formed at higher output. However, with acetylene introduction, the efficiency is always lower than diesel, but tends to increase with an increase in gas flow rates at higher loads up to 4 LPM. Moreover, while increasing the mass flow rate further to 8 LPM, the BTE is reduced compared to the lower flow rate. The reduction in BTE may be due to heterogeneous mixture formation at a higher flow rate and incomplete combustion explained earlier in the combustion study. Few experts have also stated the reason being the higher flame velocity of acetylene, increases the heat transfer losses further causing a reduction in BTE at a higher flow rate (Lakshmanan and Nagarajan 2011a).

The BTE noticed available at 2 LPM,4 LPM, 6 LPM, and 8 LPM are 28%, 28.90%, 27.50%, and 27% while introducing acetylene, at optimum load (80%). Diesel fuel shows the highest thermal efficiency of 30.59% at optimum loading condition. However, at peak load, all fuels show slight deterioration in thermal efficiency due to inefficient combustion, unable to combust gaseous-air mixture completely in comparison to 80% load. Similar results are described in the various literature (Lakshmanan and Nagarajan 2011b and Srivastava et al. 2017). Overall, at 4 LPM flow rate of acetylene, the dual fuel engine has comparable BTE to baseline diesel. The reasons may be due to enhanced mixing, improved combustible mixture formation, and increased combustion efficiency, leading to maximum utilization of acetylene energy at optimum load (80% load).

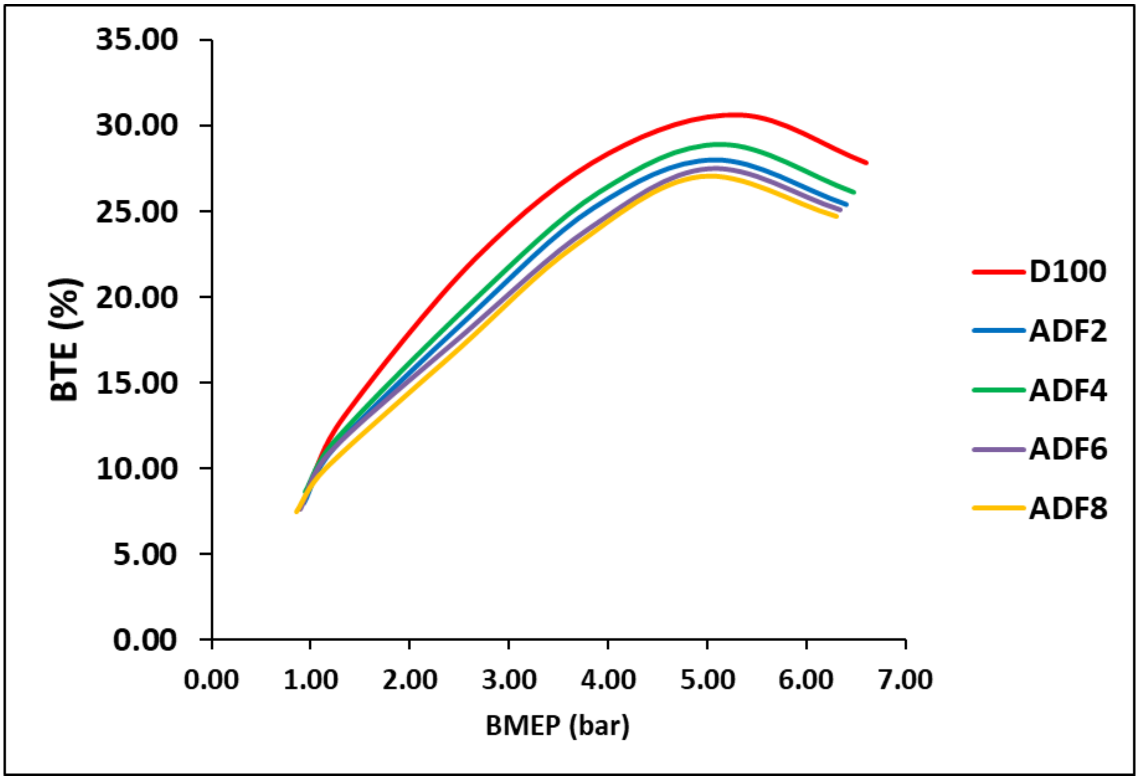


**Figure 4.11:** Variation of BTE with the increase in BMEP

#### 4.2.2.2 Brake Specific Energy Consumption

BSEC measures the fuel conversion rate for producing unit power (Brusca et al., 2014). Fig. 4.12. indicates the variation of BSEC with a rise in BMEP for all experimented fuels. For standard diesel fuel, BSEC varies from 66 MJ/kWh to 23 MJ/kWh at optimum load. Furthermore, when acetylene is supplied at the flow rate of 2 LPM, 4 LPM, 6 LPM, and 8 LPM at full load, the BSEC is found to be 31 MJ/kWh, 28 MJ/kWh, 33 MJ/kWh, and 37 MJ/kWh respectively. It is observed that with an increased flow rate of acetylene gas, BSEC increases due to rapid combustion caused by higher flame velocity of acetylene suppressing ignition centres

(Parthasarathy et al., 2014). However, few engine experts have also stated that the reduction in effective power in comparison to diesel engine may be one of the reasons for increasing BSEC (Choudhary et al., 2018).

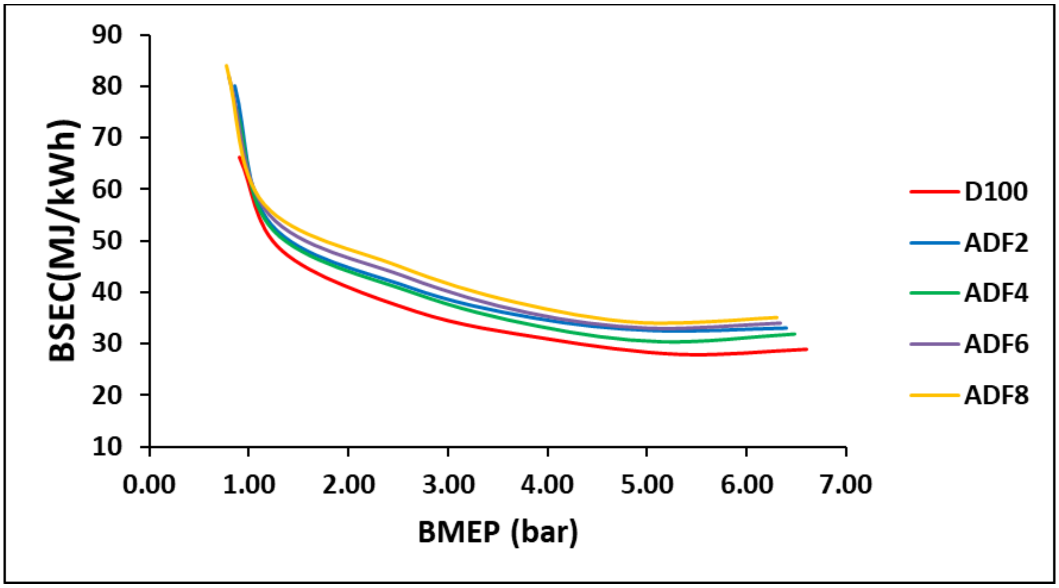


**Figure 4.12:** Variation of BSEC with the change in BMEP

The results obtained are similar to previously published work (Mahla et al., 2012 and Lakshmanan and Nagarajan 2009, 2010a). Furthermore, it is noticed that at lower loads higher quantum of energy goes unutilized, causing an increase in BSEC. However, at intermediate to high loading conditions, effective utilization of input energy shared by acetylene is observed, reducing BSEC at elevated load, finally increasing the thermal efficiency of dual fuel engine at optimum load. In the case of 4 LPM acetylene, BSEC's trend is closer to that of neat diesel fuel may be because of improved combustion characteristics at higher output.

#### 4.2.2.3 Exhaust Gas Temperature

The exhaust gas temperature (EGT) variation with a change in BMEP is shown in Fig. 4.13. EGT increases with an increase in load percentage. Similar trends are observed in the various kinds of literature (Srivastava et al., 2017 and Behera, Murugan, and Nagarajan 2014), this is attributed to an increase in combustion temperature at elevated load. Furthermore, it is noticed that with an increasing mass flow rate of acetylene, the EGT reduces in the case of 6 LPM and 8 LPM compared to 2 LPM and 4 LPM. It is due to a higher heat transfer caused by the conduction of gaseous molecules at the elevated flow rate eventually reducing the thermal efficiency explained earlier.

Similar reasons have been stated by various engine experts (Karim et al., 2010 and Koli and Rao, 2020). For diesel operation, the EGT increases to 757 K, whereas, for acetylene induction at 2 LPM, 4 LPM, 6 LPM, and 8 LPM, EGT is observed 737 K, 748 K 717 K, and 708 K respectively, at full load. The increment in EGT at 2 LPM and 4 LPM in comparison to 8 LPM again reflects the improved BTE, which may be due to effective utilization of heat released and stable combustion. The decrease in the EGT in dual fuel engine will increase the durability of the engine components further increasing the reliability of customized engine than conventional diesel engine (Lakshmanan and Nagarajan, 2011b).

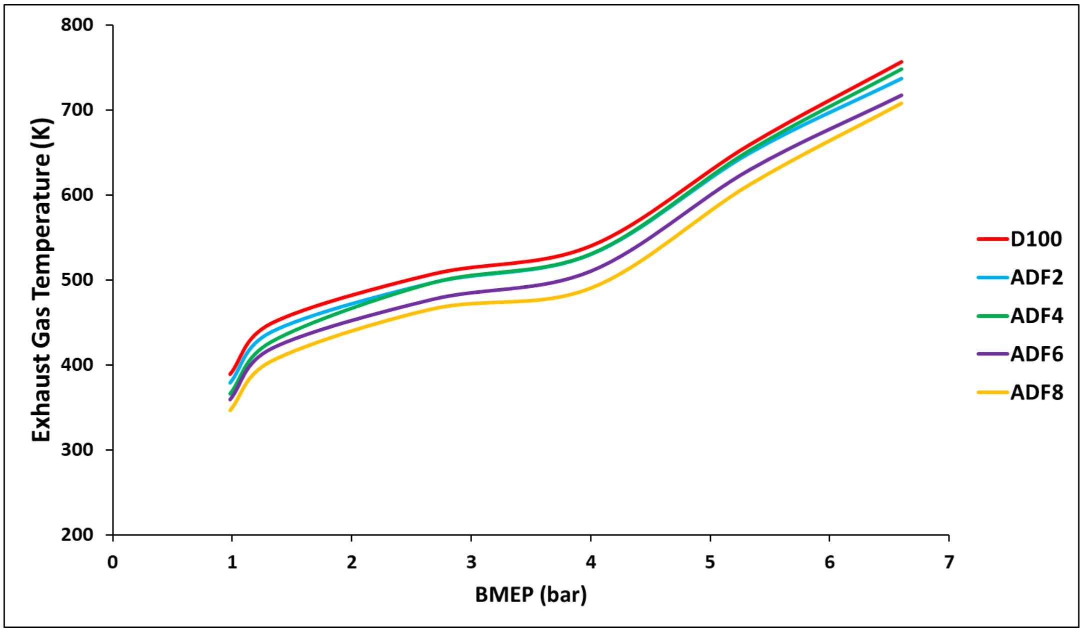


**Figure 4.13:** Variation of EGT with the increase in BMEP

### 4.2.3 Emission Characteristics

Harmful contaminants, including hydrocarbons (HC), carbon monoxide (CO), smoke, and nitrogen oxides (NOx), are the main difficulties faced by diesel engines. They have directly impacted the environment and human health. The researcher has established firm conclusions from the outcomes and comparative study with the neat diesel.

**Hydrocarbons**

When the unburnt fuel particles are trapped in between the wall and engine cylinder, hydrocarbon (HC) is emitted. HC is mainly produced due to a lack of proper diesel fuel oxidation. The HC emission variation with change in BMEP is seen in Fig. 4.14. HC emissions ranged from 0.41 g / kWh to 0.09 g / kWh from no load to full loads for pure diesel. Moreover, HC emissions reduced drastically with acetylene induction at all loading parameters excluding at full load. At higher engine loads the peak cylinder pressure, as well as temperature increases hence complete combustion, is attained consequently results in higher BTE and lower carbon-based emissions (Lakshmanan and Nagarajan, 2010). During acetylene dual-fuel combustion, the substantial reduction in HC emissions is attributed to the increase in the mixture's flame velocity (1.5 m/s), which further improved the combustion characteristics.

The HC exhalations for the dual fuel engine at full load are 0.091 g/kWh, 0.072 g/kWh, 0.063 g/kWh, and 0.098 g/kWh for 2 LPM, 4 LPM, 6 LPM, and 8 LPM gaseous flow rate, respectively. Similar trends have been observed by Choudhary, Nayyar, and Dasgupta, 2018 by utilizing acetylene. The reduction in HC emission in the case of dual fuel mode up to 80% load is observed due to the higher burning velocity of acetylene that enhances the burning rate and slightly leaner mixture with an increase in gas flow rate. However, at full load HC emissions is slightly higher with an increased flow rate of gaseous fuel owing to rich fuel-air mixture in comparison to lower flow rate.

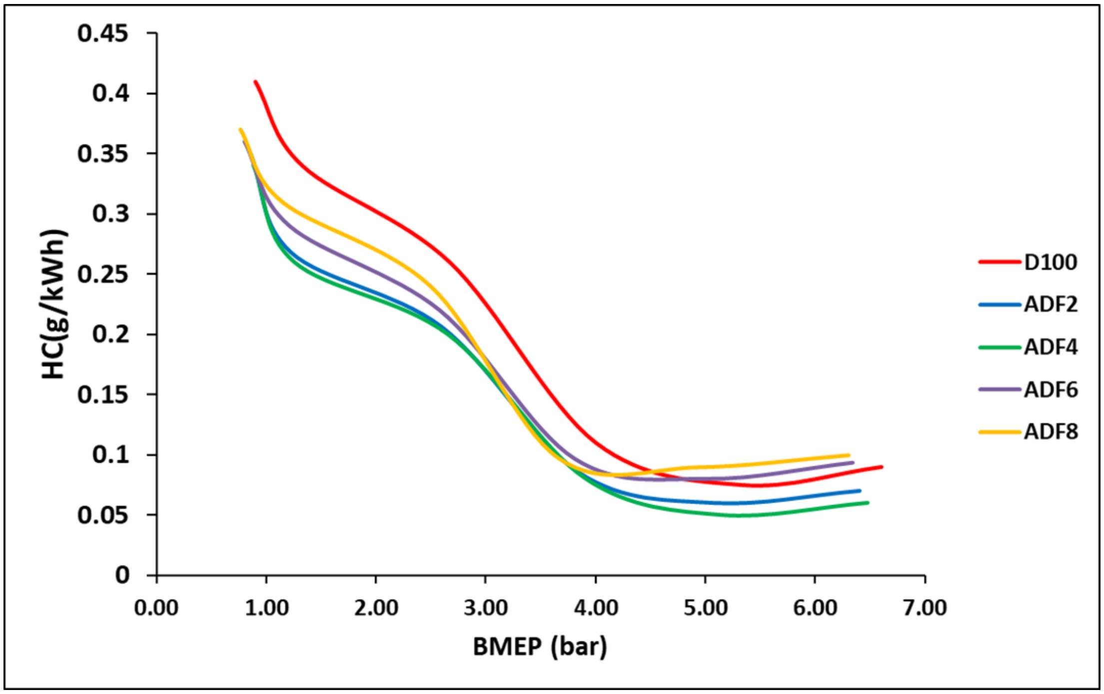


**Figure 4.14:** The fluctuation of HC with the change in BMEP

However, it is noticed that HC emissions are slightly inclined at peak load due to an increase in the amount of HC molecules at a higher flow rate. This shows that when the quantity of acetylene is increased, the specific amount of air is subsequently displaced by gaseous fuels and leads to incomplete combustion. Many scientists have observed similar conclusions (Lakshmanan and Nagarajan 2010b, 2010a). At 4 LPM acetylene induction, the lowest HC level is observed, i.e., 30 % lower than standard diesel. Consequently, the reasons for lowering HC emissions may be due to the wide ignition limit, higher flame velocity, and higher energy content of the gaseous fuels responsible for improved combustion (Mahla et al., 2012).

#### 4.2.3.2 Carbon Monoxide

The crucial factor for the formation of carbon monoxide (CO) is an inadequate oxygen supply (Lata, Misra, and Medhekar, 2012). Some scientists have reported that heterogeneous air-fuel mixture caused an increased rate of CO emissions (Karim et al., 1999). The CO emission deviation with a changing BMEP is seen in Fig. 4.15. In all loading conditions, acetylene DFE displays a lower CO exhalation level than regular diesel mode. In DFE, this is because of the leaner operational range relative to pure diesel. With the change in load, similar patterns have been observed for CO emissions by many engine scientists(Srivastava et al. 2017 and Behera, Murugan, and Nagarajan 2014).

Furthermore, it is observed that for rich mixtures, CO formations are comparatively higher due to inadequate oxygen accessibility for the oxidation process (Tsolakis et al., 2014). The lower emissions rate of CO during the acetylene enriched mixture suggests that the charge is homogenous, and the modified engine is operating on a lean mixture. However, at an elevated flow rate, some of the fuel goes unburnt due to incomplete combustion which causes higher CO emissions (Sonachalam and Manieniyan, 2020).

Hence, the CO emission is reduced significantly with acetylene induction. The variation of CO from no-load to maximum load is observed to be 11 g/kWh to 0.41 g/kWh for neat diesel, whereas, for dual fuel mode, it is 8 g/kWh to 0.30 g/kWh for 2 LPM, 7 g/kWh to 0.25 g/kWh for 4 LPM, 9 g/kWh to 0.35 g/kWh for 6 LPM, 10.5 g/kWh to 0.39 g/kWh for 8 LPM. CO decreases by 39.20 % at full load for 4 LPM acetylene induction owing to the increased combustion temperature as well as improved combustion efficiency at elevated load, relative to neat diesel.

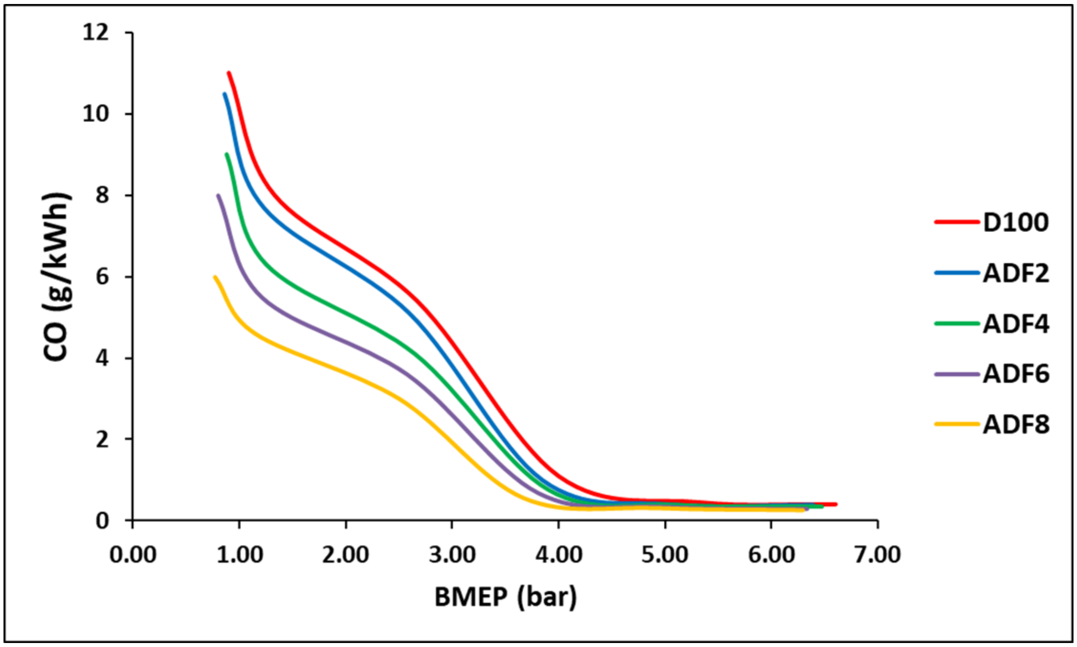


**Figure 4.15:** Deviation of CO with the increase in BMEP

#### 4.2.3.3 Oxides of Nitrogen ($NO_x$)

For all experimental fuels, NOx increases with increasing load. It may be attributed to increased load; more fuel must be added, which raised the reaction temperature and the cylinder's peak pressure and thus increases NOx levels (Chandra et al., 2011). Moreover, NOx levels are smaller in dual fuel engine than those of pure diesel at lower loads, owing to the decreased intensity of premixed flames (Karabektas et al., 2014). It can be observed from Fig. 4.16. while introducing acetylene; there is a rapid increase in NOx formation from intermediate to peak loading conditions. Moreover, it has also been noticed that the NOx emission rate rises with an increased flow rate of acetylene gas (Wang et al., 2007).

It may be attributed to the increase in peak cycle temperature level because of faster energy release, which is confirmed by the increased peak cycle pressure in pressure -crank angle graph. The level of NOx is 12.67 g / kWh for the neat diesel while 13.5 g / kWh, 14.76 g / kWh, 16.6 g / kWh, and 19.1 g / kWh at 2 LPM, 4 LPM, 6 LPM, and 8 LPM respectively at fuel at maximum loading conditions. Different engine experts (Lakshmanan and Nagarajan 2010b, 2010a, 2010c, 2009) observed similar trends, and the present results are corroborated with the previous works of literature. At 4 LPM gaseous fuel induction, the NOx increases by 16% compared to standard diesel. It may be attributed to higher flame velocity and elevated combustion temperatures with higher HRR as explained earlier in the combustion section.

As stated by Zeldovich mechanism high reaction temperatures, adequate reaction time, and excess oxygen supply in the combustion chamber are the primary reasons for NOx formation. Overall, it can be concluded that as the mass flow rate is increased, the rise in NOx level is observed due to faster and uncontrolled combustion release eventually increasing the peak cylinder pressure seen in the pressure versus crank angle diagram. Operating the acetylene engine with lean mixtures can reduce NOx emissions. The lean mixture results in lower temperature and slows the chemical reaction, which weakens the kinetics of NOx formation.

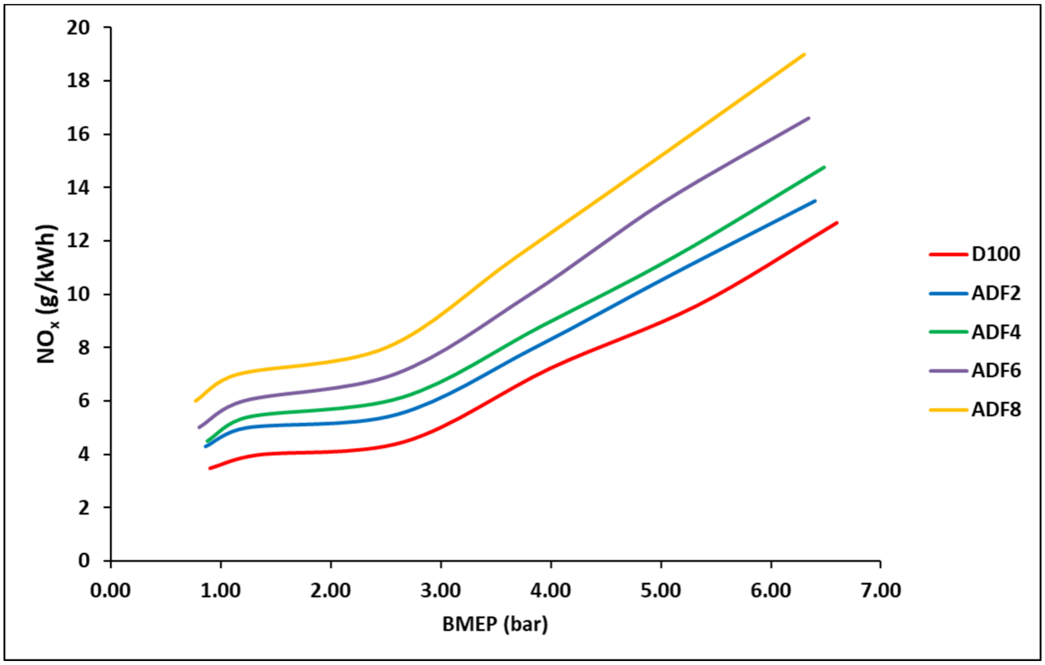


**Figure 4.16:** The fluctuation of $NO_x$ with varying BMEP

#### 4.2.3.4 Smoke Emissions

The smoke meter is used to measure the black smoke coming from the exhaust of the CI engine. The variation of the smoke level versus BMEP is seen in Fig. 4.17. The formation of smoke is primarily due to the hydrocarbon pyrolysis effect during the rich fuel mixture, leading to excessive combustion, mostly under low to moderate loading (Karim et al., 1999). With the rise in load, the smoke level increases, and for regular diesel, it is almost 70% at maximum load. The non-uniform air-fuel mixture formation may be the reason for rising smoke levels in a conventional diesel engine (Ahmed et al., 2017).

However, many engine experts reported that during the diffusion stage, maximum smoke formation occurs (Marlena et al., 2019). It is observed that the smoke percentage decreases by

20.9%, 25.6%, 35.1%, and 41.4% while operating engine at 2 LPM, 4 LPM, 6 LPM, and 8 LPM acetylene induction compared to the neat diesel. It may be because of higher energy released during the premixed combustion phase as well as higher diffusivity of gaseous molecules promoting homogenous mixture in dual fuel mode. Many scientists have observed a similar pattem (Srivastava et al. 2017 and Behera, Murugan, and Nagarajan 2014). Few engine scientists have stated the reasons for the reduction in smoke emissions are instantaneous combustion which results in a decrement in the diffusion combustion stage and thus acetylene is not exposed to pyrolysis (Mahla et al., 2012). Overall, the percentage of smoke levels strictly depends on the fuel flow rate, injection timing, and mixture composition required for efficient combustion.

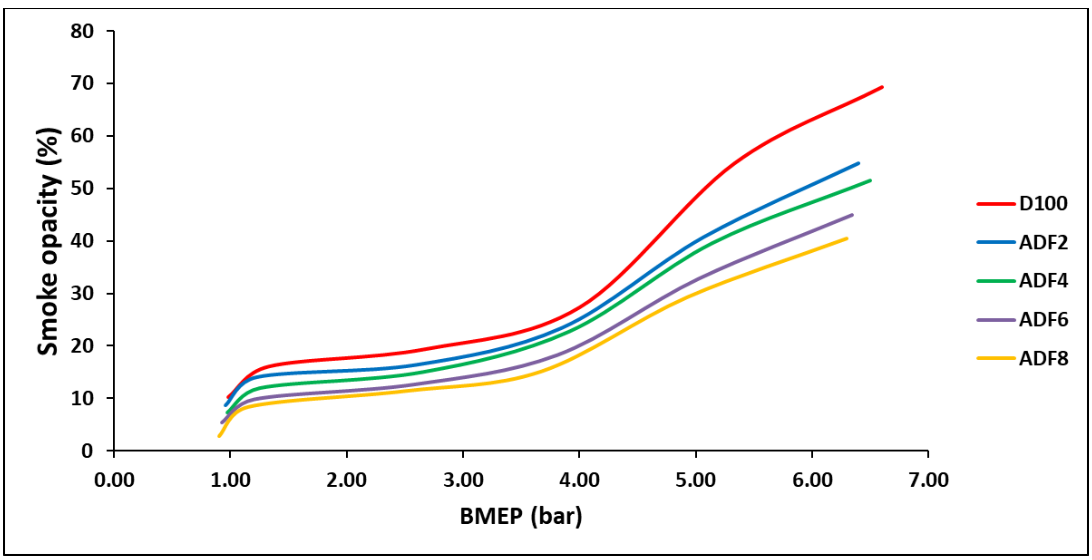


**Figure 4.17:** Variation of Smoke opacity with the increase in BMEP

The summary of the test results at full load is given in table 4.2 whereas the comparative analysis of the present study with previous work is given in table 4.3.

**Table 4.2:** Summary of the results at full load

| Parameters | Neat Diesel | Acetylene @2 LPM | Acetylene @4 LPM | Acetylene @6 LPM | Acetylene @8LPM |
|---|---|---|---|---|---|
| MaximumBTE (%) | 30.59 | 28 | 28.9 | 27.5 | 27 |
| Variation with diesel (%) | -- | -8.47 | -8.27 | -10.10 | -11.74 |
| EGT(K) | 757 | 737 | 748 | 717 | 708 |
| Variation with diesel (%) | -- | -2.64 | -1.18 | -5.28 | -6.47 |
| Peak Pressure | 69 | 72 | 77 | 79 | 81 |
| Variation with diesel (%) | -- | +4.35 | +11.6 | +14.5 | +17.4 |
| NOx (g/kWh) | 12.6 | 13.5 | 14.7 | 16.6 | 19 |
| Variation with diesel (%) | -- | +7.14 | +16.67 | +31.74 | +50.79 |
| HC (g/kWh) | 0.09 | 0.072 | 0.063 | 0.091 | 0.098 |
| Variation with diesel (%) | -- | -20 | -30 | +1.12 | +8.88 |
| CO (g/kWh) | 0.41 | 0.30 | 0.25 | 0.35 | 0.39 |
| Variation with diesel (%) | -- | -26.82 | -39 | -14.63 | -4.87 |
| Smoke Opacity (%) | 69.26 | 54.75 | 51.48 | 44.9 | 40.54 |
| Variation with diesel (%) | -- | -20.95 | -26.16 | -35.17 | -41.46 |

Decrement (-); Increment (+); Nil (--)

**Table 4.3:** Comparative study of experimental results (various gas flow rate)

| **Parameters for comparison** | Results in previous works of literature (Lakshmanan and Nagarajan, 2010, 2011a,2011b; Mahla et al., 2012; Behera et al., 2014; Choudhary et al., 2018; Koli and Rao,2020) | Results in this study |
|---|---|---|
| **BTE** | Increases*<br>Increases/Decreases** | Increases*<br>Increases up to 4 LPM and at higher LPM decreases |
| **BSEC** | Decreases*<br>Increases** | Decreases*<br>Decreases at 4 LPM and at higher LPM increases |
| **EGT** | Increases*<br>Decreases** | Increases*<br>Decreases** |
| **HC** | Increases*<br>Decreases** | Increases *<br>Decreases at 4 LPM and slightly higher at another flow rate |
| **CO** | Increases*<br>Decreases** | Increases*<br>Decreases at 4 LPM and slightly higher at another flow rate |
| **NOx** | Increases*<br>Increases ** | Increases*<br>Increases ** |
| **Smoke** | Increases*<br>Decreases * | Increases*<br>Decreases ** |

* (concerns to increase in load); ** (increase in the mass flow rate of gaseous fuel)

### 4.3 Predicting the Engine Parameters by using ANN technique

In this current study, performance, and exhaust emissions of acetylene fuelled CI engine is predicted with an artificial neural network (ANN) modeling. The input parameters are variable gas flow rate (2,4,6 and 8 LPM), variable loading condition (0,20,40,60,80 and 100%), diesel substitution rate (DSR), discharge pressure, energy shared (%) by liquid fuel as well as a gaseous fuel. In contrast, target data are BTE, BSEC, EGT, HC, CO, and NOx. The software details are provided in Table 4.4.

**Table 4.4:** Software Configuration

| Software (Matlab-2018a)/Algorithms | |
|---|---|
| Data Division | Random (dividerand) |
| Training | Levenberg-Marquardt (trainlm) |
| Performance | Mean Squared Error (mse) |
| Calculations | MEX |
| Epoch/Validation | 1000 iterations |

It is found that more accurate results are obtained with the usage of ANN to predict desired performance and exhaust emissions. The input parameters, as well as output parameters, are shown in Figure 4.18. The MSE for 70% of the training is 0.00104, for 15% validation is 0.0014, and 15% testing is 0.00092. Moreover, it is demonstrated that the R values are reasonably close to 1, which shows the reliable and exactness of the anticipated data for the network range with 0.9977 for training, 0.9926 for validating, and 0.9959 for testing.

Similar results have been observed by many researchers (Mehra et al., 2018; Singh, Jain, and Mahla, 2020; Ramalingam et al., 2019). The ANN approach can predict the precise and accurate value of the target assigned for a dual fuel engine very easily and in less time. The predicted accuracy is 99.77%, 99.92%, 99.59%, 99.76%, 99.59%, and 99.12%, for BTE, BSEC, EGT, HC, CO, and NOx, respectively. The proposed network shows a perfect agreement with the experimental results which can be seen in Fig. 4.19.

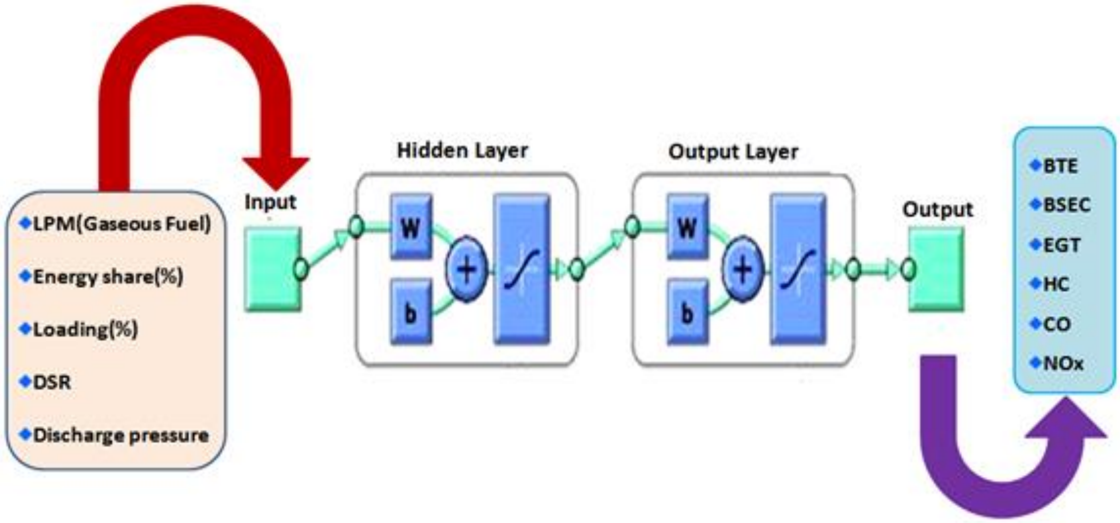


**Figure 4.18:** Predicted Parameters evaluated during ANN analysis

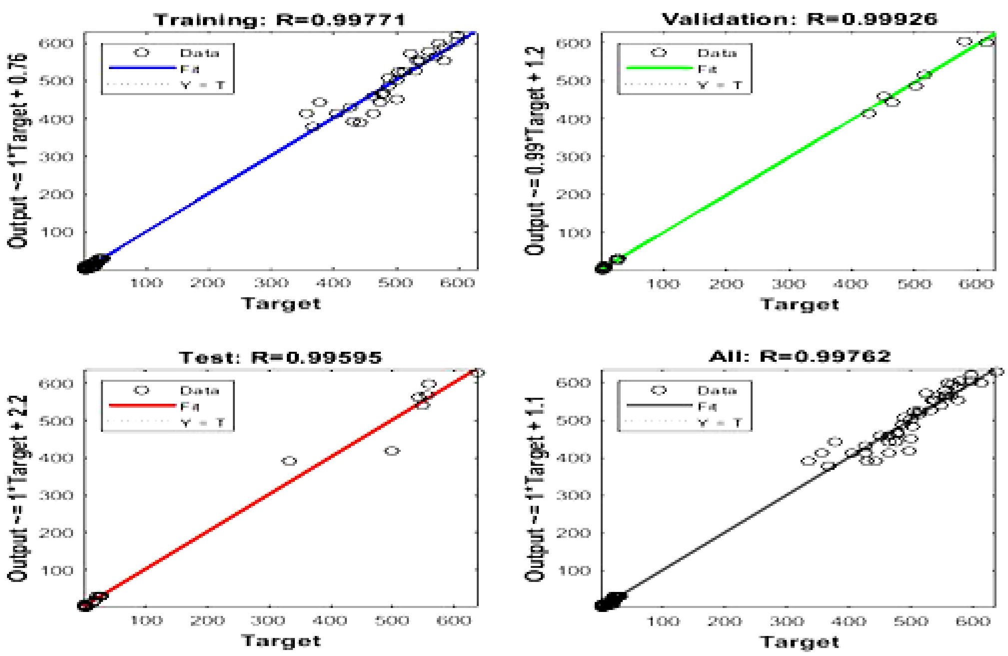


**Figure 4.19:** Overall correlation coefficient of the developed network

## 4.4 Utilization of Oxygenated Blends at Optimized Flow Rate

The combustion, performance, and emission parameters of the acetylene dual fuel engine are investigated with a constant flow rate of acetylene at 4 LPM. In contrast, different blend ratio (5% -20%) of diesel/n-butanol blends and diesel/Di-ethyl ether blends are used as pilot fuel. The results are analyzed and compared with the standard diesel fuel for the entire loading range. The energy shared by acetylene in a dual fuel engine is shown in Fig. 4.20. It is seen that acetylene energy contribution at lower load has a higher weightage compared to full load, which is attributed to lower combustion temperature of the engine cylinder at lower load and also due to the mixing of acetylene with residual gases present inside the engine cylinder from the previous cycle.

Similar outcomes have been found in the various literature (Lakshmanan and Nagarajan 2010a,2010b; Behera, Murugan, and Nagarajan 2014). It is noticed from Fig 4.14, that 48% and 46% of energy is shared by acetylene at optimum load for BDEE10 and BBU10 blends. The maximum substitution may be restricted due to the knocking tendency at higher loads.

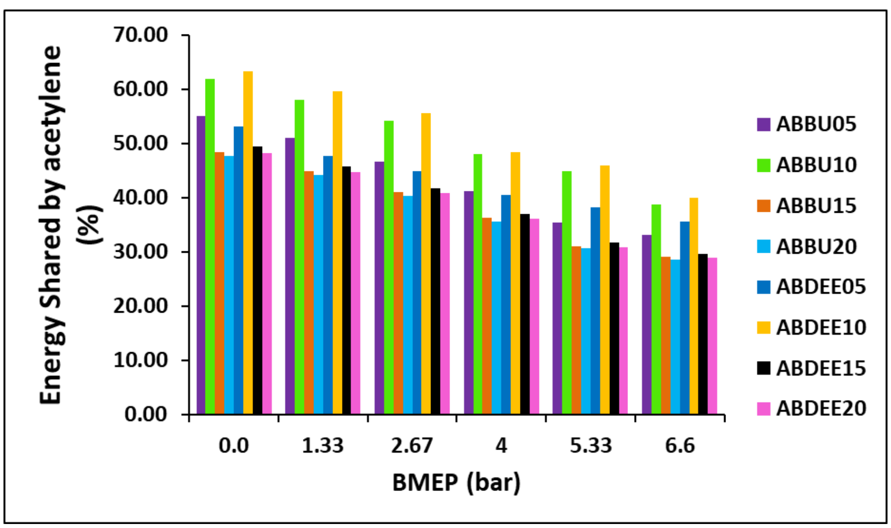


**Figure 4.20:** Energy shared by acetylene (%)

### 4.4.1 Combustion Analysis

In-cylinder pressure, heat release rate, and ignition delay are evaluated for different blends as an ignition source in acetylene DFE.

#### 4.4.1.1 Heat Release Rate

The obtained result shows the standard pressure deviation considered for 100 cycles inside the engine cylinder. Figure 4.21. illustrates the variation of heat release rate versus crank angle. The peak heat release rate for neat diesel is 56.05 J/$^{o}$CA whereas, when acetylene is inducted in dual fuel mode with different pilot fuels, the value of HRR 60.42 J/$^{o}$CA for BBU05 blend as pilot fuel, 61.62 J/$^{o}$CA for BBU10 blend as pilot fuel, 55.32 J/$^{o}$CA for BBU15 mixture as pilot fuel 53.41 J/$^{o}$CA for BBU20 blend as pilot fuel. Furthermore, 63.67 J/$^{o}$CA for BDEE05 blend as pilot fuel, 66.26 J/$^{o}$CA for BDEE10 blend as pilot fuel, 61.47 J/$^{o}$CA for BDEE15 blend as pilot fuel 58.70 J/$^{o}$CA for BDEE20 blend as pilot fuel. The increase in HRR is due to the dominant premixed combustion phase of acetylene as well as the higher calorific value of acetylene in comparison to neat diesel. Few engine experts have also stated that increased accumulation of oxygenated blends causes higher HRR.

The maximum value of HRR for dual fuel engine is observed when the 10% oxygenated diesel blend is utilized as pilot fuel in both the cases for diesel/n-butanol blends as well as diesel/Di-ethyl ether blends. The reason may be due to lower viscosity, density, and higher volatility causing higher HRR up to 10 % blend. Furthermore, when the blend is increased to 15% and 20 % respectively, a reduction in HRR is observed in comparison to the 10% blend owing to the higher latent heat of vaporization causing colling effect eventually reducing the combustion temperature of the charge. Similar causes for the reduction in HRR are mentioned by many engine experts (Mahla et al., 2012; Behera, Murugan, and Nagarajan 2014) while utilizing acetylene in dual fuel mode. With the addition of Di-ethyl ether reduction of ignition delay is observed which increases the combustion temperature and thus the maximum quantity of acetylene can be burnt resulting in higher HRR.

Furthermore, the autoignition temperature of the DEE is lesser than diesel hence earlier combustion is observed in the case of DEE/diesel blends. However, up to 10% DEE, HRR is relatively higher because of excellent mixing and higher volatility. Moreover, when the blend proportion is increased to 15% and 20% the lower viscosity causes cavitation and further deteriorates the injection process which yields a slight decline in HRR at higher DEE percentage and reduction in thermal efficiency. The peak cylinder pressure also follows the same trend.

Moreover, the maximum pressure is observed in the case of BDEE10 blend. However, with increasing DEE% from 5 to 10 % oxygen content also increases which improves overall combustion of dual fuel engine however further increasing DEE % from 10 to 20 % combustion is inefficient because of the higher latent heat of vapourization despite higher oxygen content. The maximum HRR occurs $2^o$ to $5^o$ earlier in the case of DEE/diesel blends compared to baseline diesel may be due to the reduced combustion duration. The earlier HRR is mainly due to higher cetane number, lower autoignition temperature, and reduced ignition delay period of blends (Lakshmanan and Nagarajan 2010a,2010b;).

Similarly, for n-butanol acetylene dual fuel engine, HRR increases due to the advancement of energy released in the cycle caused by acetylene and shorter combustion duration caused by oxygenated blends. The peak HRR for n-butanol /diesel blends is slightly lower than DEE/diesel blends at every proportion. The reason may be due to the lower heating value of n-butanol diesel blends as well as lower cetane index which causes maximum pressure reduction and maximum occurrence of HRR is delayed in comparison to DEE/diesel blends. However, due to improved atomization caused by lower viscosity, lower density, and higher latent heat of vaporization of n-butanol diesel blends results in higher HRR in comparison to single fuel mode. The oxygen concentration of n-butanol/diesel blends is higher than that of DEE/diesel blends which offset the low cetane number.

While increasing the n-butanol percentage from 5 to 10 % there is an increment in HRR which signifies better atomization whereas further increasing percentage of n-butanol from 10 to 20 % HRR is reduced due to lower cetane rating and higher latent heat of vaporization causing ignition a bit retarded. At this instant lower pressure and deterioration of performance are also observed which is explained below. However, NOx emissions are reduced while utilizing a higher percentage of both oxygenated blends might be due to the colling effect and reduction in the combustion temperature. A similar observation has been observed during the pressure versus crank angle graph. Hence, the presence of n-butanol and Di-ethyl ether in acetylene fuelled CI engine shows a dominant premixed combustion stage with a slight reduction in a diffusion flame.

Hence due to this, BTE also improves. Similar findings are mentioned in various literature (Behera, Murugan, and Nagarajan 2014; Srivastava et al., 2017). It can be further noticed that with increasing load, HRR increases for all experimented fuels owing to better combustion characteristics at medium to high loads. This can be further quantified as an increase in load, cylinder pressure of the engine increases significantly, resulting in improved combustion temperature, eventually increasing HRR of all the experimented fuels.

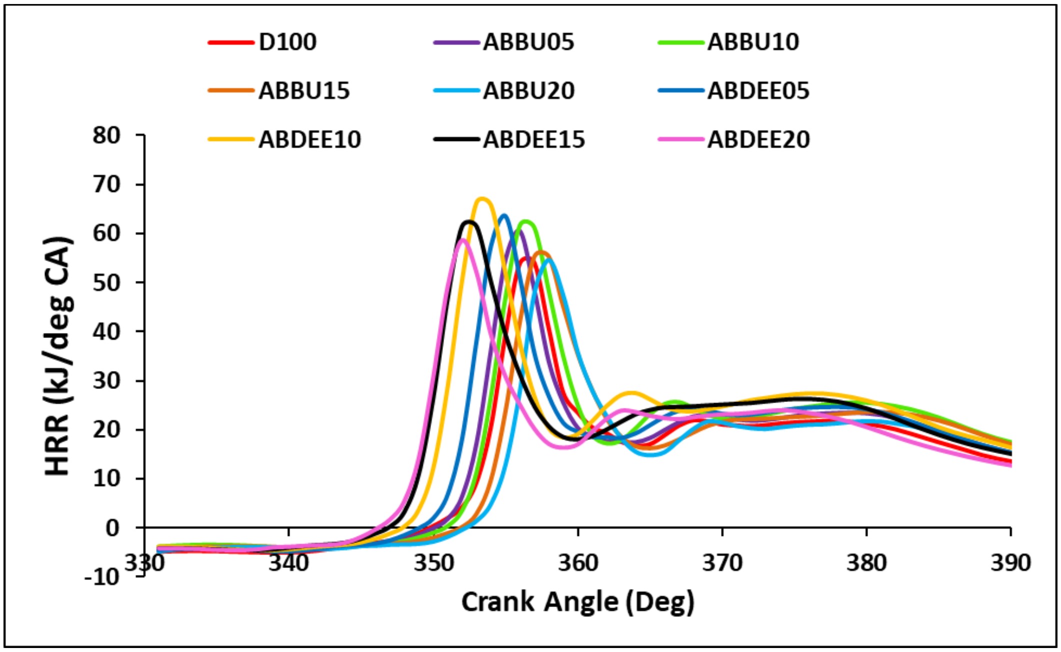


**Figure 4.21:** Variation of HRR with Crank Angle

#### 4.4.1.2 In-cylinder Pressure

The deviation of peak cylinder pressure with changing crank angle is shown in Figure 4.22. For a regular diesel engine, the peak pressure is 69 bar, at full load, whereas while inducting acetylene at a constant flow rate of 4 LPM for different pilot fuels, the peak pressure is 74 bar for BBU05, 76 bar for BBU10, 73 bar for BBU15 and 72 bar for BBU20 diesel/n-butanol blends.

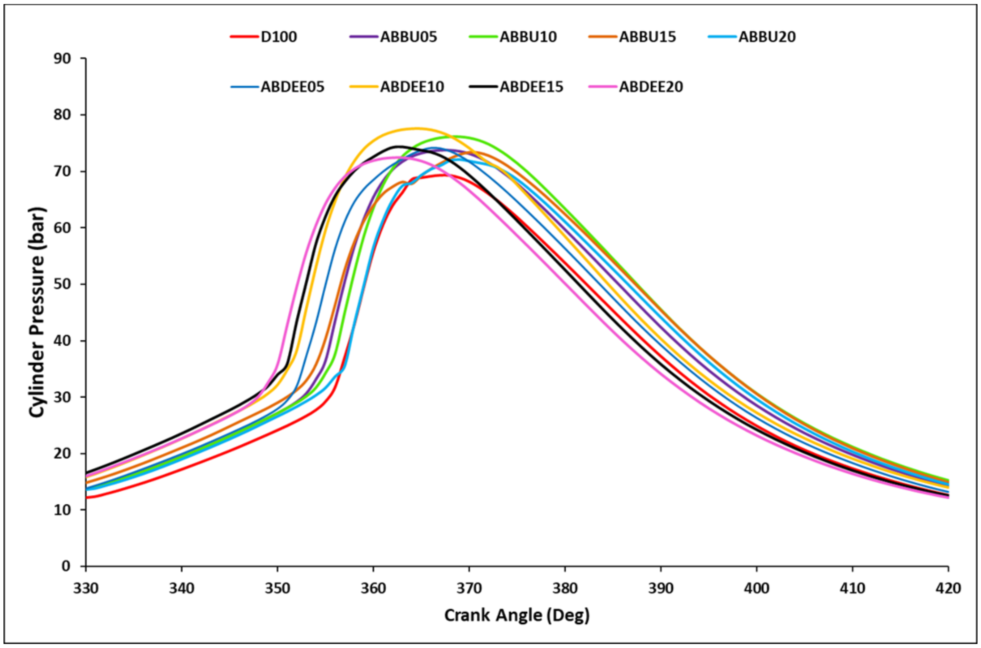


**Figure 4.22:** Variation of Cylinder Pressure with Crank Angle

Moreover, the peak pressure is 75 bar for acetylene-diesel DFE for BDEE05, 78 bar for BDEE10, 74 bar for BDEE15and 72 bar for BDEE20 blend acetylene fuelled CI engine using diesel/Di-ethyl ether blends as a plot fuel. The maximum rise of pressure is observed when BBU10 and BDEE10 blends are utilized as pilot fuel. Due to the increased amount of oxygenated fuels, complete combustion of gaseous fuel mixture is possible, which further elevates the combustion temperature leading to a high rise in pressure peaks. The peak cylinder pressure for acetylene DFE with n-butanol/diesel blends is observed at 7$^o$ to 9$^o$ CA ATDC whereas for DEE/diesel blends peak pressure is observed at 4 to 6$^o$ CA ATDC. The reason has already been explained in the HRR section. Since the higher cetane number and lower autoignition temperature are the reason for the earlier combustion of DEE/diesel blends compared to n-butanol/diesel blends. The progression in peak pressure during acetylene dual fuel engine is possibly due to instantaneous and earlier energy released during the combustion of acetylene in comparison to neat diesel.

#### 4.4.1.3 Ignition Delay

It can be noticed from Figure 4.23. the delay period shortens with a rise in load. It signifies that as the load is increased gradually, combustion duration is reduced which increases the combustion temperature compared to partial loads. Initially, at a lower load, the combustion duration is higher relative to peak loads.

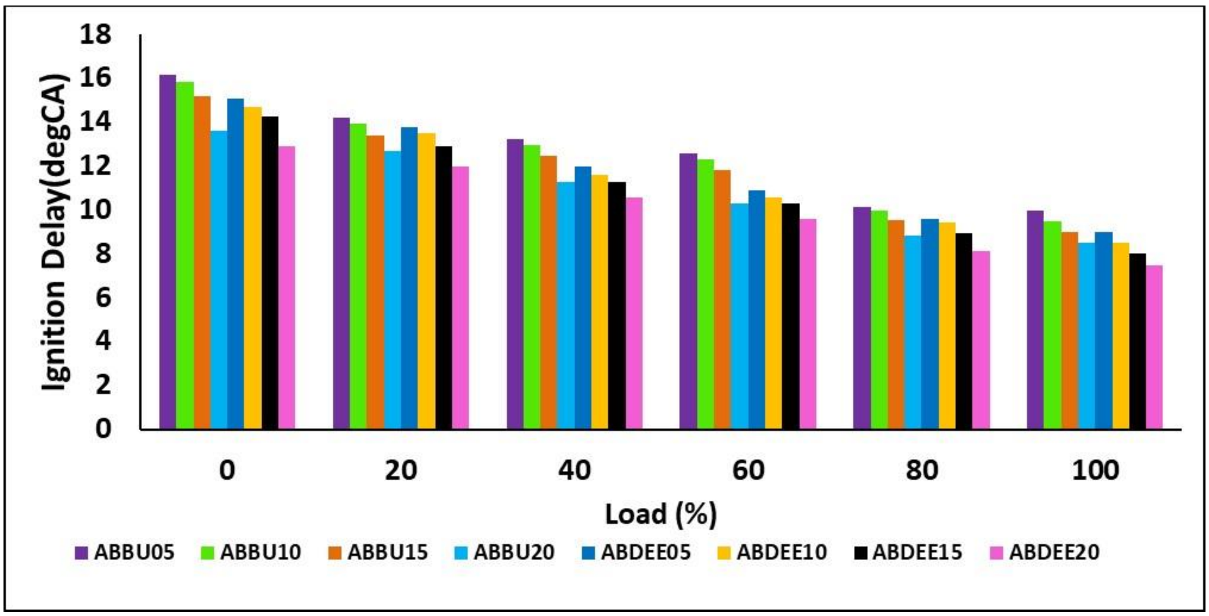


**Figure 4.23:** Variation of Ignition Delay with Load (%)

The ignition delay at 80% load is 10.15°, 9.95°, 9.55°, and 8.82° for dual fuel engine with BBU05, BBU10, BBU15, and BBU20 blends, respectively. However, with a supplement of BDDE05, BDEE10, BDEE15, and BDEE20 blends, the delay period is 9.57°, 9.45°, 8.95°, and 8.12° at full load. The addition of DEE blends reduces the delay period in comparison to n-butanol blends at full loading conditions. The reason has been explained above as DEE blends have higher cetane number as well as lower latent heat of vaporization causing the reduction in ignition delay as compared to n-butanol diesel blends.

It is further observed that with the increase in DEE percentage delay period is reduced in contrast while the increasing n-butanol percentage delay period gets prolonged because of the lower cetane index as well as higher latent heat of vaporization. Overall delay period is reduced at higher loads with the supplement of oxygenated fuels because of improved combustion and increase in combustion temperature causing reduction of physical delay at higher load. Hence both blends up to 10 % can be utilized as ignition improver for increasing the performance of dual fuel engine.

### 4.4.2 Performance Analysis

Performance parameters are measured at a constant flow rate of acetylene of 4 LPM under diverse loading conditions with different blend fraction of diesel/n-butanol and diesel/Di-ethyl ether blends as pilot fuel. Finally, the results are compared to neat diesel.

#### 4.4.2.1 Brake Thermal Efficiency

Figure 4.24. demonstrates the deviation of brake thermal efficiency (BTE) with a change in brake mean effective pressure (BMEP) for different blend ratios of diesel/n-butanol, diesel/DEE, and neat diesel while inducting acetylene at a constant flow rate of 4 LPM in a modified CI engine. It is observed that BTE for acetylene fuelled modified diesel with oxygenated pilot fuel is higher than neat diesel owing to complete combustion and higher energy released. Some researchers have highlighted the improvement in physicochemical properties by utilizing oxygenated blends further improving the performance of the gaseous fuel engine. DEE and n-butanol diesel blends have lower viscosity, lower density, lower compressibility, and higher

volatility which improves atomization as well as ignition characteristics (No et al., 2016). Thus, for the BBU05 blend acetylene engine, maximum BTE is observed as 30.71%; meanwhile, when the n-butanol percentage is increased to 10% (BBU10), maximum BTE is 31.80%, similarly for BBU15 and BBU20 blends, maximum BTE is found to be 29.87% and 29.72% respectively at higher loading conditions.

The thermal efficiency is increased by 0.4% and 3.95% for BBU05 and BBU10 blend as a pilot fuel in dual fuel engine whereas thermal efficiency is decreased by 2.35% and 2.84% for BBU15 and BBU20 blend. Similar trends have been observed by many engine experts while utilizing n-butanol/diesel blends (Gu et al., 2011; Malaviya et al., 2012 and Rakopoulos et al., 2010). Moreover, while utilizing diesel/DEE blends as pilot fuel maximum BTE for BDEE05, BDEE10, BDEE15, and BDEE20 blends are 31.13%, 32.25%, 30.10%, and 29.63%, respectively. Furthermore, it is observed that the BTE of DEE/diesel blends as plot fuel has 1.76% and 5.42% higher BTE than neat diesel for BDEE05 and BDEE10 blends. The reason might be DEE/diesel blends has higher oxygen concentration, high cetane number, and higher volatility which is responsible for the complete oxidation of gaseous molecules.

Similar results have been reported in the literature by many researchers (Rakopoulos et al., 2013). Further increasing the DEE% from 10 to 20% inferior performance has been observed which may be due to higher latent heat of vaporization causing a cooling effect of the charge as well as the reduction in combustion temperature (Rakopoulos et al., 2016). However, NOx emissions are reduced at a higher percentage of oxygenated blends (Karabektas et al., 2014).B10 blends in both types of blends (diesel/n-butanol and diesel/DEE) is higher than that of neat diesel.

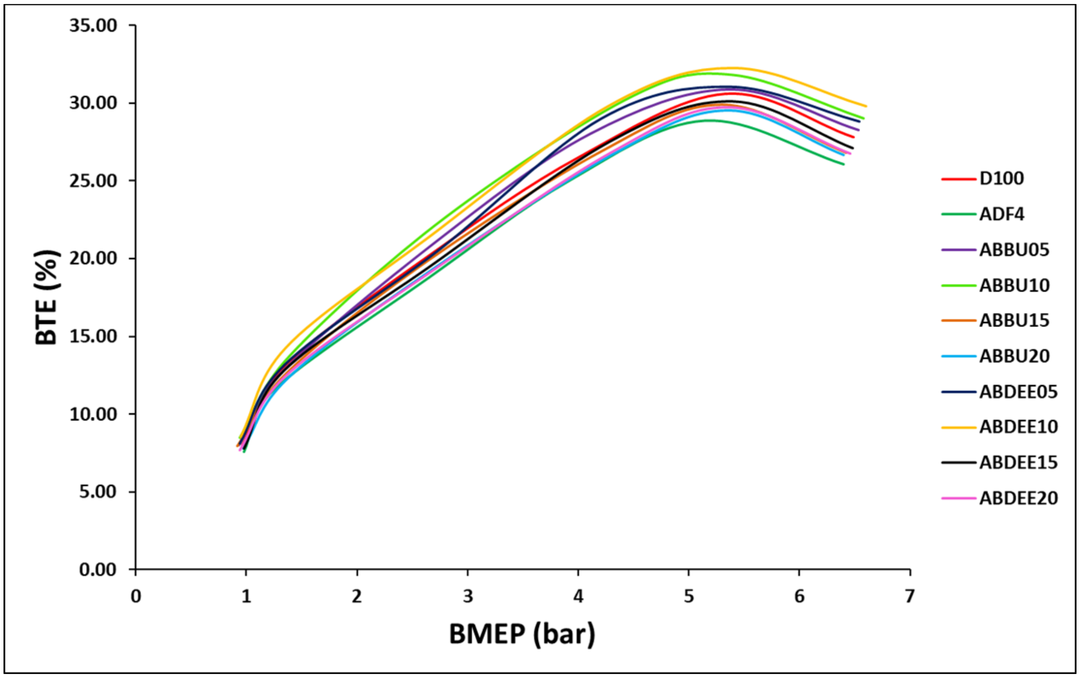


**Figure 4.24:** Variation of BTE with BMEP

In general, it can be inferred that when oxygenated blends are utilized as pilot fuels, efficient and stable combustion takes place, maybe due to improved pysico-chemical properties of the pilot blends, leading to better utilization of gaseous fuel mixture. It is also attributed to complete combustion during dual fuel mode caused due to a substantial rise in pressure as well as combustion temperature that converts the higher amount of fuel-air mixture into useful work. Meanwhile, it is also observed that when the proportion of oxygenated blends is further increased from 10 to 20 %, the performance deteriorated marginally compared to neat diesel. The higher dominance of the latent heat of vapourization (LHV) over the cetane index of the pilot fuel blends produces a cooling effect. Finally, it results in the reduction of combustion temperature, causing incomplete combustion (Behera, Murugan, and Nagarajan,2014).

It is also observed that with the increase in load (up to 80 % load), BTE improves for all tested fuels, which also resembles while increasing load the gaseous-fuel mixture formation is closer to stoichiometric and homogenous in nature that causes complete combustion of gaseous fuel mixture, may be initiated by the higher flame speed of gaseous fuel eventually leading to higher heat release rate which can be seen during combustion analysis (Srivastava et al., 2017).

#### 4.4.2.2 Brake Specific Energy Consumption

Brake-Specific energy consumption (BSEC) is the product of brake-specific fuel consumption (BSFC) and the amount of heat supplied. It is inversely proportional to the brake thermal efficiency (BTE). Figure 4.25. depicts the variation of BSEC with the change in BMEP. When the acetylene is inducted, BSEC is observed to be higher at lower loading conditions compared to neat diesel. Still, when the load is increased gradually, BSEC reduces for acetylene dual-fuelled engine. This may be due to improved combustion characteristics at the higher loads.

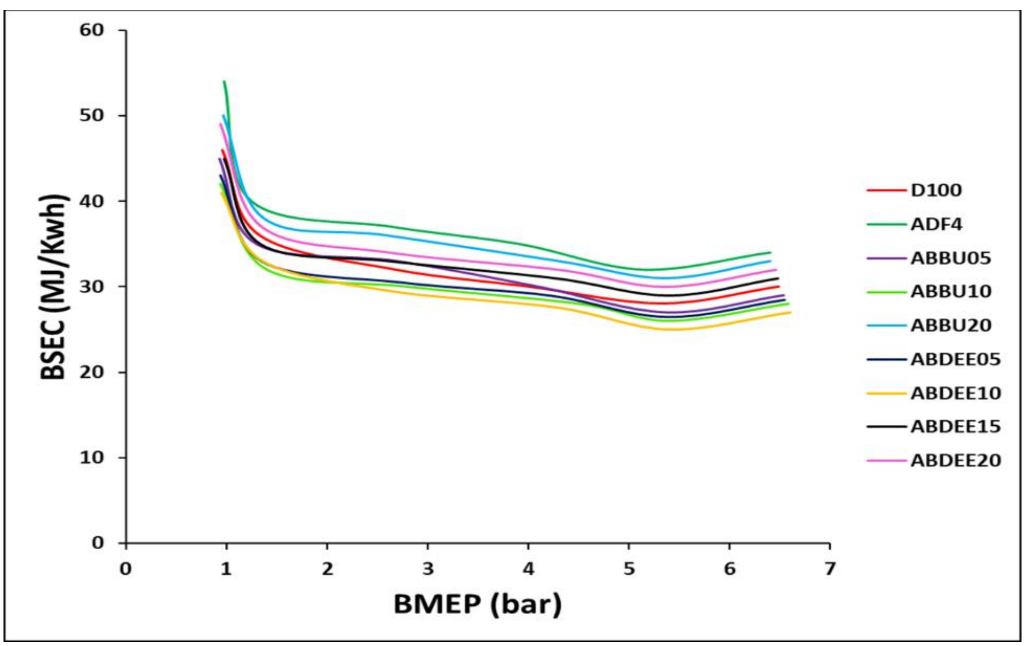


**Figure 4.25:** Variation of BSEC with BMEP

As the load increases, BSEC reduces considerably; this can be justified as at medium to high loading condition, the leaner gaseous-air mixture is realized; hence fuel consumption is reduced significantly at peak loads (Karim, 1983). BSEC at full load is observed 26 MJ/kWh for neat diesel, 35 MJ/kWh for acetylene-diesel dual fuel engine while inducting acetylene at a fixed flow rate of 4 LPM, 27 MJ/kWh for an acetylene-BBU05 blend, 25 MJ/kWh for acetylene-BBU10 blend, 29 MJ/kWh for acetylene- BBU15 blend and 36 MJ/kWh for the acetylene-BBU20 blend.

Similarly, BSEC for the acetylene-BDEE05 blend is 26 MJ/kWh, for acetylene- BDEE10 blend 24 MJ/kWh, for the acetylene-BDEE15 blend, is 32 MJ/kWh and for the acetylene-BDEE20 mixture is 36 MJ/kWh. The lowest brake-specific energy consumption is found when 10% of oxygenated blends are added to neat diesel as an ignition source. In contrast, the brake specific energy consumption increased in the case of 20% oxygenated fuels. Similar trends have been observed by many engine scientists (Mallikarjun, 2015). The reasons have been already explained during BTE, i.e., improved physico-chemical characteristics of oxygenated blends are responsible for ameliorated performance. Overall, it can be concluded that 5 to 10 % oxygenated fuels can be blended with neat diesel for proper utilization of acetylene and optimize the performance of acetylene fuelled diesel engine. It might be due to the efficient combustion of gaseous fuel caused by enough oxygen supplied by diesel/n-butanol and diesel/DEE blends.

#### 4.4.2.3 Exhaust Gas Temperature

Exhaust gas temperature measure the temperature of the exhaust of the CI engine. It is also a key factor in evaluating combustion behavior inside the engine cylinder. Figure 4.26. demonstrates the exhaust gas temperature (EGT) deviation with the change in brake mean effective pressure (BMEP). It can be seen from Figure 4.26 with increasing load EGT increases for all the tested fuels may be due to increased load, the higher rate of pressure rise takes place, which promotes an increase in combustion temperature, causing the maximum amount of energy released during high loads which can also be seen during the study of combustion parameters.

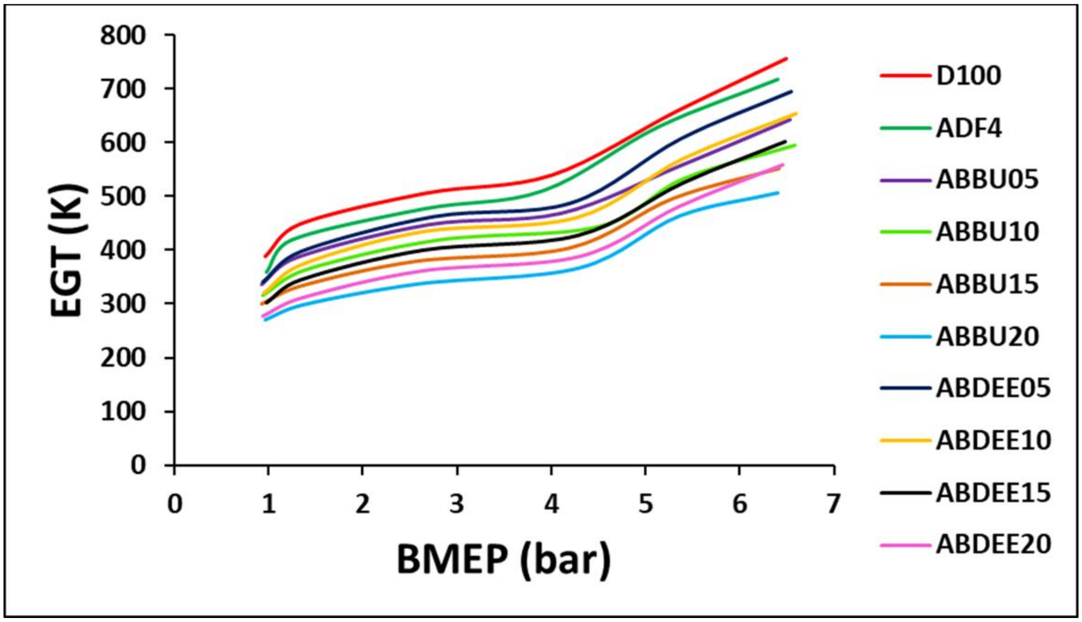


**Figure 4.26:** Variation of Exhaust Gas Temperature with BMEP

The EGT decreases with the induction of acetylene compared to standard diesel. It might be due to acetylene's higher flame speed, causing higher HRR, eventually increasing heat transfer losses from the cylinder wall to the surroundings. This result is also verified by the cylinder pressure graph mentioned above during combustion analysis. EGT is reduced drastically when diesel/n-butanol and diesel/DEE blends are used as pilot fuel. It proves that due to the addition of oxygenated blends at a higher proportion (15% and 20%) combustion temperature reduces significantly because of the relatively higher LHV of oxygenated blends than petroleum diesel. Furthermore, it produces a cooling effect. The same result is corroborated during the emissions study, i.e., nitrogen ($NO_x$) reduced drastically with a higher proportion of oxygenated blends because of slightly reduced peak pressure and temperature of the engine cylinder (Mahla et al., 2012).

It is observed that for diesel/n-butanol blends, the EGT increases up to 643 K, 596 K, 551 K, and 507 K for BBU05 blend, BBU10 blend, BBU15 blend, and BBU20 blend, respectively. Similarly, the maximum EGT is found to be 695 K for acetylene-BDEE05 blend, 655 K for acetylene-BDEE10 blend, 602 K for acetylene-BDEE15 combination, and 558 K for acetylene-BDEE20 blend. It is also observed that with the increasing proportion of oxygenated blends in dual fuel engine, EGT reduces significantly may be due to a higher latent heat of vaporization of the pilot fuel, causing cooling effect reducing the combustion temperature of the charge (Sudheesh and Mallikarjuna, 2012).

Overall, it can be concluded that EGT for acetylene fuelled engine differs from conventional diesel engines mainly due to variation in combustion characteristics, which depends on the quality of the fuel-air mixture, engine speed, ignition delay, nature of loading, latent heat of vaporization, and heating value of the fuel (Yusuf et al., 2010). Excessive EGT may lead to crack in the cylinder head and often cause wear in a piston, and thus it is undesirable. The utilization of gaseous fuel is advantageous in dual fuel engine (Paul et al., 2015).

### 4.4.3 Emission Analysis

The emissions analysis for acetylene fuelled CI engine while injecting different pilot fuels are compared with neat diesel.

#### 4.4.3.1 Hydrocarbons

The major reasons for hydrocarbons (HC) emissions are in-appropriate mixing of fuel-air mixture, variation in air-flow rate, fluctuating speed, and ignition timing. Some of the unburnt particles are trapped in the chamber crevice, resulting in incomplete combustion, further causing a higher unburnt hydrocarbon. Other causes of higher HC emissions are quenching the flame near the combustion chamber's walls and engine cylinder crevices. The deviation of hydrocarbon (HC) emissions versus brake mean effective pressure (BMEP) is illustrated in Figure 4.27.

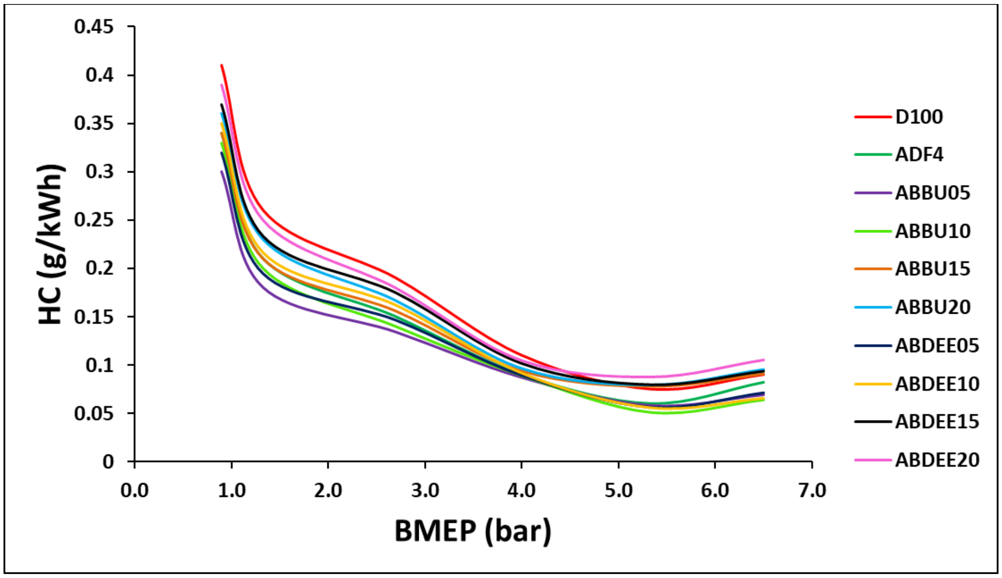


**Figure 4.27:** Variation of HC emissions with BMEP

When the engine is operated on neat diesel, HC emissions rise due to heterogeneous fuel-air mixture formation due to an excess amount of gaseous fuel being inducted at lower loads. In contrast, when the load increases gradually, the gas-air mixture tends to be more homogenous, and a lean variety allows enough oxygen to take part in the combustion process, which eventually enhances the engine's performance at higher loads. Moreover, when oxygenated blends are injected as pilot fuel, the level of HC emissions rises marginally for a higher percentage of blend proportion (Lakshmanan and Nagarajan (2011a).

It is observed that HC emissions are reduced to 0.069 g/kWh for the BBU05 blend. For BBU10, blend HC is decreased to 0.064 g/kWh while increasing blend percentage to BBU15 and BBU20 HC emission increases slightly compared to neat diesel, i.e., 0.091 g/kWh 0.096 g/kWh, respectively, at full load. The increased rate of HC emissions with higher blend fraction attributes to the combined effects of low cetane index and high evaporation heat of n-butanol/diesel blends. The lower cetane index of the blends lengthens the ignition delay period, permitting sufficient time for pilot fuel to evaporate. Meanwhile higher rate of vapourization causes increased HC emissions. Similarly, when diesel/DEE blends are injected as pilot fuels, for BDEE05 blends, it reduces to 0.072 g/kWh, and for the BDEE10 blend, it is further reduced to 0.066 g/kWh relative to baseline diesel. However, while utilizing the BDEE15 and BDEE20 blend, HC emissions increases to 0.094 g/kWh and 0.105 g/kWh. Moreover, the trend shows that the cooling effect may be dominated at a higher blend fraction of oxygenated fuels; hence, gaseous fuel mixture cannot ignite utterly due to lower combustion temperature, and, consequently, un-burnt particles are expelled to the atmosphere (Mahla et al., 2012).

This peculiar characteristic maybe also due to over mixing and prolonged ignition delay of the gaseous-fuel mixture. The lowest HC emissions are observed for the BBU10 blend (diesel/n-butanol), i.e., 29 % lesser than petroleum diesel, whereas for the BDEE10 blend (diesel/DEE), i.e., 27 % lesser than baseline diesel.

#### 4.4.3.2 Carbon Monoxide

The deviation of Carbon monoxide (CO) with the change in BMEP is depicted in Figure 4.28. Most scientists have quoted the major reasons CO formations are incomplete combustion caused due to lack of sufficient oxygen responsible for proper combustion. In all the cases, CO emissions are higher at no load condition, whereas when the load is increased gradually, CO formation reduces drastically; this may be due to lean mixture formation at higher loads.

The acetylene fuelled CI engine emits lesser CO emissions than petroleum diesel; this may be due to a lesser number of C: H ratio present in acetylene gas, higher diffusivity, and higher flame speed of gaseous fuel causing proper combustion. At full load, CO reduces to 0.35 g/kWh during acetylene induction while injecting diesel as pilot fuel. Furthermore, while injecting diesel/n-butanol blend, as a pilot fuel in dual fuel engine, CO declines to 0.27 g/kWh for BBU05 blend, 0.22 g/kWh for BBU10 blend, 0.19 g/kWh for BBU15 blend, and 0.17 g/kWh for BBU20 blend. Similarly, while injecting diesel/DEE blend, as a pilot fuel in dual fuel engine, CO decreases to 0.28 g/kWh for BDEE05 blend, 0.23 g/kWh for BDEE10 blend, 0.20 g/kWh for BDEE15 blend, and 0.18 g/kWh for BDEE20 blend. Hence, it can be seen from Figure 4.28 the lowest CO emissions are observed while utilizing a 20% blend as a pilot fuel for both cases for diesel/n-butanol blends as well as diesel/DEE blends at full load in DFE compared to regular diesel. The excess oxygen present in the BBU05 blend is 12.90% higher than BDEE05 blend. Moreover, BBU10 blend has 12.29% higher oxygen (%) than BDEE10 blend. Furthermore, BBU15 blend has 11.30% higher oxygen percentage than BDEE15 blend and BBU20 blend has 10.76% higher oxygen (%) than BDEE20 blend. Hence n-butanol diesel blends showed slightly lesser CO emissions in comparison to DEE/diesel blends.

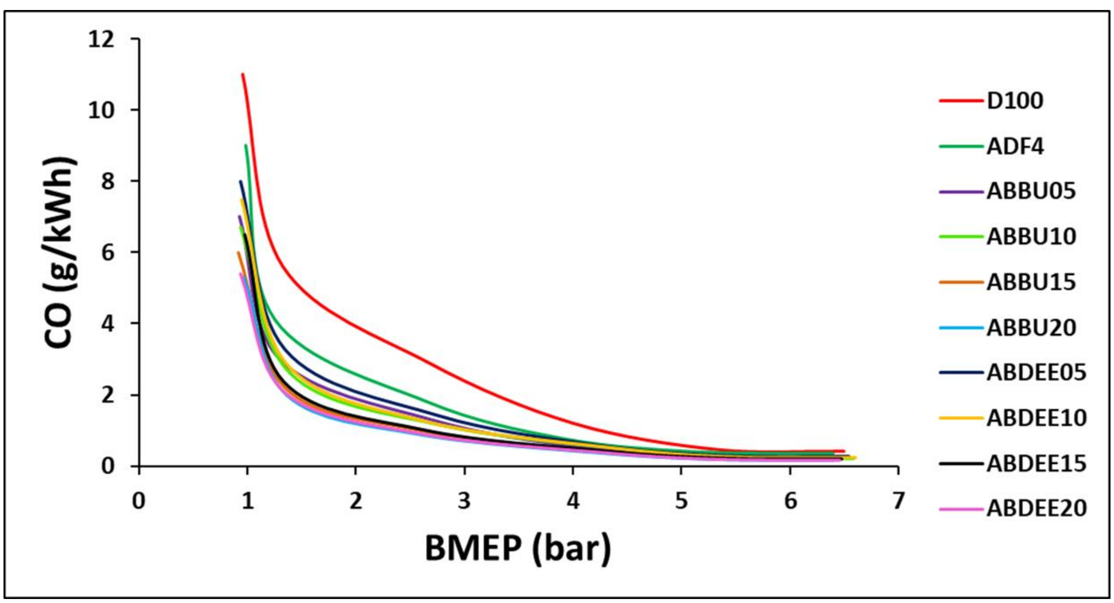


**Figure 4.28:** Variation of CO emissions with BMEP

Moreover, with the addition of oxygenated fuel, combustion improves gradually due to sufficient oxygen available, which is responsible for burning the gaseous fuel-air mixture completely; simultaneously, significant improvement in the performance is also observed, which is explained above during performance analysis. Hence, the dual fuel engine has an advantage over HCCI and other available combustion techniques. CO emissions are 37 % and 34% lesser than that of neat diesel while injecting BBU10 and BDEE10 blends in dual fuel engine in comparison to neat diesel at optimum load (80%).

#### 4.4.3.3 Oxides of Nitrogen

Oxides of Nitrogen ($NO_x$) emissions are caused due to higher combustion temperature, an increase of residence time, and when an enormous quantity of oxygen takes part in the combustion process (Behera et al., 2014). Moreover, $NO_x$ formation also depends on in-cylinder pressure and the equivalence ratio (Behera et al., 2014). Figure 4.29. depicts the deviation of $NO_x$ with a change in brake mean effective pressure (BMEP). Acetylene fuelled diesel engine with diesel as ignition source shows higher NOx emissions than neat diesel due to the rapid rate of heat release caused by instantaneous combustion during acetylene induction (Lakshmanan and Nagarajan, 2011b).

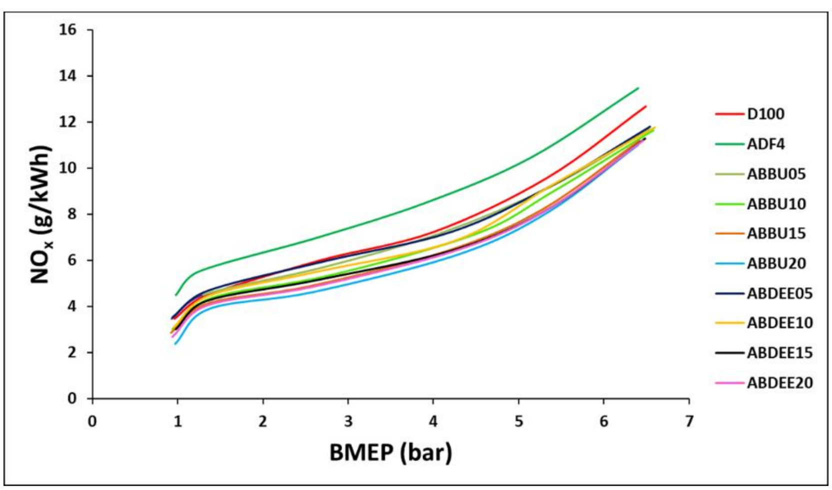


**Figure 4.29:** Variation of $NO_x$ emissions with BMEP

$NO_x$ emissions for neat diesel at full load are 12.67 g/kWh, whereas when it is operated on dual fuel mode, $NO_x$ increased to 14.76 g/kWh at full load while diesel as the pilot source. However, $NO_x$ is reduced drastically while using oxygenated fuels as pilot fuel during acetylene fuelled CI engine. It is observed that while injecting diesel/n-butanol blends as pilot fuel, the values of $NO_x$ for BBU05, BBU10, BBU15, and BBU20 blends are 10.87 g/kWh, 9.16 g/kWh, 8.13 g/kWh and 6.97 g/kWh respectively at full load.

Similarly, while injecting diesel/DEE blends as pilot fuel, observed $NO_x$ for BDEE05, BDEE10, BDEE15, and BDEE20 blends are 11.49 g/kWh, 9.87 g/kWh, 8.45 g/kWh, and 7.34 g/kWh, respectively at full load. $NO_x$ emissions reduced to 27 % and 45 % for BBU10 and BBU20 blends respectively in the case of diesel/n-butanol used as pilot fuel, whereas $NO_x$ emissions reduced to 22 % and 42 % for BDEE10 and BDEE20 blends respectively in case of diesel/DEE used as pilot fuels. The lowest $NO_x$ emissions are observed for 20% blends in both types of oxygenated fuels. That attributes to the relatively low combustion temperature of an engine cylinder is attained during the 20% blend due to the higher latent heat of vapourization (LHV) of both oxygenated fuels, which causes reduction in $NO_x$ for the entire range of loads. Similar results have been observed by many engine scientists (Choudhary, Nayyar, and Dasgupta, 2018).

#### 4.4.3.4 Smoke Opacity

Figure 4.30. depicts smoke opacity deviation with the change in the brake mean effective pressure (BMEP). Maximum smoke emissions are observed during neat diesel cases; subsequently, when acetylene is inducted, the rate of smoke emissions reduces drastically because of enhanced combustion characteristics caused by gaseous fuel mixture and due to less number of carbon particles present in gaseous fuel mixture in comparison to neat diesel. It is found that smoke emissions decrease considerably for acetylene fuelled CI engine, i.e., 25.67 % lesser than baseline

diesel at full load. Moreover, when oxygenated blends with diesel are injected as pilot fuel, the smoke level is reduced at a substantial rate owing to better mixing characteristics of pilot fuel resulting in complete combustion of gaseous fuels. Overall, smoke opacity is lowest for 10% blends in both cases, i.e., 43.58 % lesser in the case of BDEE10 blended pilot fuel and 39.18% lesser in BBU10 blended pilot fuel compared to regular diesel at maximum load. It might be because of the lower carbon atoms present in both oxygenated fuels as well as primary fuels than petroleum diesel.

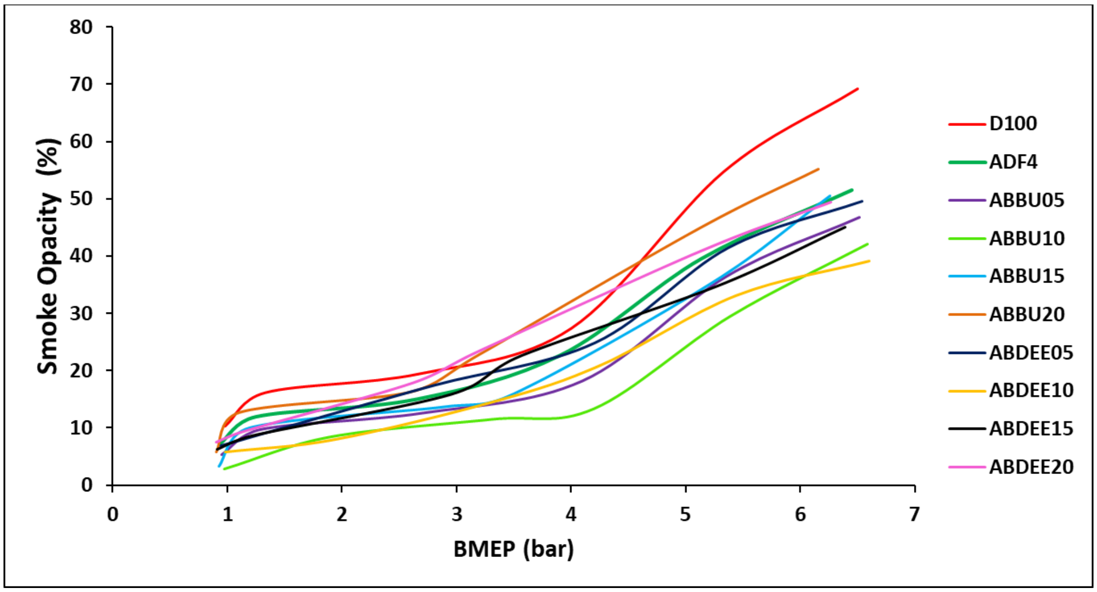


**Figure 4.30:** Variation of smoke opacity with the change in BMEP

The summary of the experimented results at high load is mentioned in table 4.5 whereas a comparative assessment of the present work with previous literature is given in table 4.6.

**Table 4.5:** Summary of experimental results

| Parameters | ABB U05 | ABB U10 | ABBU 15 | ABBU 20 | ABDEE 05 | ABDEE 10 | ABDEE 15 | ABDEE 20 |
|---|---|---|---|---|---|---|---|---|
| BTE (%) | 30.71 | 31.80 | 29.86 | 29.75 | 31.13 | 32.25 | 30.10 | 29.63 |
| Cylinder Pressure(bar) | 74 | 77 | 73 | 72 | 75 | 78 | 74 | 73 |
| HRR (kJ/$^o$CA | 60.48 | 61.62 | 55.34 | 54.61 | 63.67 | 66.26 | 61.47 | 58.69 |
| EGT (K) | 643 | 596 | 551 | 507 | 695 | 655 | 602 | 558 |
| HC(g/kWh) | 0.069 | 0.064 | 0.091 | 0.096 | 0.072 | 0.066 | 0.094 | 0.10 |
| CO(g/kWh) | 0.27 | 0.22 | 0.19 | 0.17 | 0.28 | 0.23 | 0.19 | 0.18 |
| NOx (g/kWh) | 10.87 | 9.16 | 8.13 | 6.97 | 11.49 | 9.87 | 8.45 | 7.34 |
| Smoke (%) | 46.8 | 42.12 | 50.54 | 55.16 | 49.59 | 39.08 | 45.13 | 49.43 |

**Table 4.6:** Comparative study of experimental results (Oxygenated blend)

| **Parameters for comparison** | Results in previous literature (Lakshmanan and Nagarajan, 2011a; Mahla et al., 2012; Brusca et al.,2014; Behera et al., 2014; Saha et al., 2016) | Results in this study |
|---|---|---|
| **BTE** | Increases*<br>Increases/Decreases** | Increases*<br>Increases up to 10% oxygenated blend |
| **BSEC** | Decreases*<br>Increases** | Decreases*<br>Decreases at 10% oxygenated blend |
| **EGT** | Increases*<br>Decreases** | Increases*<br>Decreases** |
| **HC** | Increases*<br>Decreases** | Increases *<br>Decreases at 10% oxygenated blend |
| **CO** | Increases*<br>Decreases** | Increases*<br>Decreases at 10% oxygenated blend |
| **NOx** | Increases*<br>Increases ** | Increases*<br>Decreases at 20% oxygenated blend |
| **Smoke** | Increases*<br>Decreases * | Increases*<br>Decreases at 10% oxygenated blend |

* (concerns to increase in load); ** (increase in oxygenated fuel)

CHAPTER 5

# CONCLUSIONS AND FUTURE SCOPE

## Overview

*In the present work, experiments are carried out on modified diesel engine to run on dual fuel mode. Acetylene, which is very little explored in the literature is utilized as a primary fuel in dual fuel engine. Furthermore, the combustion, performance, and emission characteristics are studied of acetylene dual fuel engine with pilot diesel and the results are compared with the neat diesel. The current study also covers the predictive analysis of dual fuel engine by using ANN techniques. Moreover, DEE/diesel and n-butanol/diesel blends are utilized as an ignition source for improving the combustion characteristics of an acetylene-fuelled engine. The significant conclusions are presented in this chapter.*

### 5.1 Conclusions

- ❖ The diesel engine is successfully operated on acetylene and liquid diesel with slight modification.
- ❖ The dual fuel engine performed satisfactorily up to 4 LPM however when the mass flow rate of gaseous fuel was further increased performance deteriorated. Moreover, the engine at a higher flow rate was prone to noise and vibration.
- ❖ The optimized flow rate of acetylene based on the experimentation as well as predicted

results are found to be 4 LPM.

- ❖ ABDEE10 and ABBU10 show the highest brake thermal efficiency, i.e., 1.66 % and 1.21 % than neat diesel at 4 LPM acetylene induction because of improved combustion characteristics caused by oxygenated fuels.
- ❖ NOx emissions which are the major challenges of diesel engine are reduced by 22 % and 27 % in dual fuel engine while utilizing ABDEE10 and ABBU10 blend, respectively, at full load relative to neat diesel.
- ❖ CO emissions are reduced substantially by 34 % and 37 % compared to neat diesel while utilizing acetylene under dual fuel mode with ABDEE10 and ABBU10 blends.
- ❖ The smoke opacity (%) decreases significantly by 42 % and 38 % while utilizing ABDEE10 and ABBU10 blends respectively under dual fuel mode at full load compared to that neat diesel operation. Hence, it is advantageous to operate the conventional engine on a dual fuel mode.
- ❖ HC emissions are reduced by 27 % and 29 % for ABDEE10 and ABBU10 blends respectively at full load relative to baseline diesel.

In general, utilizing oxygenated blend in acetylene fuelled engine the improvement in the engine performance is observed. Moreover, substantial reduction of harmful pollutants is also found under dual fuel mode of operation. Hence, the present research solves the purpose of a clean and efficient combustion mechanism in the customized diesel engine. By adopting this technology engine experts could utilize alternative gaseous fuels as well as liquid fuels to mitigate environmental hazards and fossil fuels crisis.

## 5.2 Future Scope

- A well-to-wheel analysis in dual fuel operations may be performed for commercial application and social viewpoint.
- Utilizing gaseous fuel in a variable compression ratio (VCR) engine can be investigated to get more precise results.
- Computational fluid dynamics (CFD) modeling for dual fuel engine can be carried out in the future to validate experimental results.
- An optical diagnostic tool may be used in the future to improve the combustion characteristics of a dual fuel engine.

# APPENDIX

## Conversion of Emission results in to (g/kWh)

The following equations are used to quantify the emissions in g/kWh relative to brake power.

$$NOx\left(\frac{g}{kWh}\right)=\left[\frac{(m_a+m_D+m_{acetylene})x\ NOx\ (ppm)x\ 32.4}{m_{acetylene}x1000xBP}\right] \quad (A1)$$

$$HC\left(\frac{g}{kWh}\right)=\left[\frac{(m_a+m_D+m_{acetylene})x\ HC\ (ppm)x\ 13}{m_{acetylene}x1000xBP}\right] \quad (A2)$$

$$CO\left(\frac{g}{kWh}\right)=\left[\frac{(m_a+m_D+m_{acetylene})x\ CO\ (\%)x\ 28}{m_{acetylene}x1000xBP}\right] \quad (A3)$$

## Cost Analysis of Acetylene Fuelled engine

Several alternative fuels are available, but they may not be cost-efficient; hence they are restricted to laboratory research and may not be commercially viable. Fuel cost is a significant issue from a customer standpoint than any other factors such as availability, production techniques, storage, transportation facilities, and equivalent energy. The fuel economy of a customized diesel engine fuelled with acetylene under dual fuel mode is calculated and compared with standard diesel, as given below. It is found that the operational cost for running unit power in one hour for the acetylene dual fuel engine is 1.26 times of a standard diesel engine. It is anticipated that with the advent of new technology, the production cost of acetylene may reduce in the coming years, which will further decrease the operational cost of the engine.

**Fuel cost analysis for neat diesel fuel operation at full load**

Cost of the Diesel fuel for

1 litre (i.e., 836 kg/m3) = Rs 71.82

Cost of 1 kg of diesel fuel = Rs 85.90

Diesel fuel consumed per hour = 1.42 kg

Brake Specific Fuel Consumption = 1.42/5.5

= 0.258 kg/kWh

Cost for one unit of power Produced = 0.258×85.90

= Rs 22.16

**Fuel cost analysis for acetylene diesel dual fuel operation cost diesel**

Diesel consumed per hour in dual

fuel mode = 0.86 kg/h

Acetylene consumed hour in dual fuel operation

= 0.264 kg/h

1 kg cost of acetylene = Rs 300.00

Total fuel cost in dual fuel mode for one unit

= cost of diesel + cost of acetylene

= 0.86 x 85.90 + 0.264 x 300

= Rs 153.07

Dual fuel cost per kWh = 153.07 /5.5

= Rs 27.83/ kWh

The cost of dual fuel mode is higher by = 27.83/22.16

= 1.26 times

The cost of acetylene diesel dual fuel operation is 26% higher than that of diesel fuel for producing unit power output per hour.

**Specifications of Instruments used in the experimentation**

| **Instrument** | **Details** |
|---|---|
| Pressure transducer | Make PCB Piezotronics, Model S111A22, Range 5000 psi, Diaphragm stainless steel type, and hermetic sealed |
| Crank Angle encoder | Make Kubler, Model 8.KIS40.1361.0360Clamping/Synchro flange, 6x12.5mm shaft, IP64Logic level: RS422; Supply= 5VDCSquare wave O/P: A, A', B, B', 0, 0'Incr/turn: 360 PPR, Termination: 2m long axial cable |
| Temperature sensor | Make Radix Type K, Ungrounded, Sheath Dia.6mmX110mmL, SS316, Connection 1/4"BSP (M) adjustable compression fitting |
| Temperature transmitter | Make ABUSTEK, Model : Fr Block, Input : Thermocouple (K), Range : 0 To 1200ºC, Output : 4- 20 mA, Power supply : 24 V DC, Dimension : 44 X 25 MM, , Pre-calibrated to 1200 Deg C |
| Data acquisition device | NI USB-6210 Bus Powered M Series Multifunction DAQ Device, NI DAQmx driver Software |
| Gas flowmeter | Make: Japsin Industrial Instrumentation; Maximum: 12 LPM |
| Pressure Regulator | Make: ESAB, Model: IDA 4B, Double-stage pressure regulator, Maximum Pressure: 15 bar |
| Flame Arrester | Make: ESAB, Model: PROTEX RA, Maximum Operating Pressure: 15 bar |

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# LIST OF PUBLICATIONS

This thesis is based on the following publications, which has been highlighted with bulleted points. The papers are appended in the order listed below:

- “Performance Analysis of Diesel Cycle under Efficient Power Density Condition with Variable Specific Heat of Working Fluid”, **Roshan Raman**, Naveen Kumar: J. Non-Equilib. Thermodyn. 2019; aop, DOI:10.1515/jnet-2019-0020 **(SCI indexed)**
- “The utilization of n-butanol/diesel blends in Acetylene Dual Fuel Engine”, **Roshan Raman**, Naveen Kumar: Energy Reports, 2019; DOI: 10.1016/j.egyr.2019.08.005 **(SCI indexed)**
- “Experimental investigation to analyze the effect of induction length of diesel-acetylene dual fuel engine”, **Roshan Raman**, Naveen Kumar: Energy Sources, Part A: Recovery, Utilization, and Environmental Effects, 2019, DOI:10.1080/15567036.2019.1663314**(SCI indexed)**
- “Experimental studies to evaluate the combustion, performance and emission characteristics of acetylene fuelled CI engine”, **Roshan Raman**, Naveen Kumar: International Journal of Ambient Energy, (2020): DOI: 10.1080/01430750.2019.1709896 **(ESCI indexed)**
- “Performance and emission characteristics of twin cylinder diesel engine fueled with mahua biodiesel and DEE”, **Roshan Raman,** Naveen Kumar, Transportation Engineering, Volume 2,2020, 100024, ISSN 2666 **(Elsevier)**