PERFORMANCE EMISSION AND COMBUSTION ANALYSIS OF LOW TEMPERATURE COMBUSTION USING BIODIESEL ON C I ENGINE

Thesis

Submitted in partial fulfillment of the requirements for the Degree of

DOCTOR OF PHILOSOPHY

by

PARASHURAM BEDAR

DEPARTMENT OF MECHANICAL ENGINEERING

NATIONAL INSTITUTE OF TECHNOLOGY KARNATAKA,

SURATHKAL, MANGALORE-575025

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DECLARATION

I hereby declare that the Research Thesis entitled **"PERFORMANCE EMISSION AND COMBUSTION ANALYSIS OF LOW TEMPERATURE COMBUSTION USING BIODIESEL ON C I ENGINE"** which is being submitted to the **National Institute of Technology Karnataka, Surathkal** in partial fulfillment of the requirements for the award of the Degree of **Doctor of Philosophy** in **Mechanical Engineering** is a *bonafide report of the research work carried out by me*. The material contained in this Research Thesis has not been submitted to any other Universities or Institutes for the award of any degree.

Register Number: **121205ME12F02**

Name of the Research Scholar: **PARASHURAM BEDAR**

Signature of the Research Scholar:

Department of Mechanical Engineering

Place: NITK-Surathkal

Date:

CERTIFICATE

This is to certify that the Research Thesis entitled **"PERFORMANCE EMISSION AND COMBUSTION ANALYSIS OF LOW TEMPERATURE COMBUSTION USING BIODIESEL ON C I ENGINE"** submitted by **Mr. PARASHURAM BEDAR (Register Number: 121205ME12F02)** as the record of the research work carried out by him, *is accepted as the Research Thesis submission* in partial fulfillment of the requirements for the award of the Degree of **Doctor of Philosophy.**

Dr. Kumar G N
Research Guide
Date:

Chairman-DRPC

Date:

ಸಮರ್ಪಣೆ

ಜನ್ಮ ನೀಡಿ, ಜೀವನ ಕಲಿಸಿಕೊಟ್ಟ ಪೂಜ್ಯ ತಂದೆ – ತಾಯಿಯವರಿಗೆ ಹಾಗೂ ವಿದ್ಯಾದಾನ ಮಾಡಿ, ಬದುಕು ರೂಪಿಸಿಕೊಟ್ಟ ಎಲ್ಲಾ ಗುರುವೃಂದದವರಿಗೆ.

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ABSTRACT

The present investigation is carried to analyse the performance emissions and combustion studies of CRDI engine fuelled with 10%, 20% and 30% of Jatropha curcas fuel blend along with the application of 10%, 20% and 30% EGR rates for zero, 25%, 50%, 75% and 100% load conditions. All above test conditions were carried out for three different fuel injection timings (10° CA, 14° CA and 18°CA BTDC) and fuel injection pressure (FIP) of 600,800 and 1000 bar with single injection mode.

The parameters like brake thermal efficiency (BTE), brake specific energy consumption (BSEC), emission characteristics such as smoke opacity, oxides of nitrogen (NO_x), Hydrocarbon (HC) and carbon mono-oxide (CO) and combustion characteristics like pressure vs crank angle, net heat release rate (NHRR) were measured and analysed.

The results shows improvement in BTE with the combined effect of biodiesel blends along with EGR and also decrease of smoke opacity, HC and CO, NO_x emissions for 1000 bar FIP at 18°CA BTDC. In summary, it is optimized that engine running with biodiesel blend JB20 with 15% EGR rate culminates into NO_x reductions without affecting engine efficiency and other emissions like smoke opacity, hydrocarbon and carbon mono-oxide for single injection mode.

Experimental analysis has been continued with multiple injection modes for conventional as well under LTC mode for optimized values at JB20 blend for 1000 bar FIP with 15% EGR rate at 75% load, as obtained from single injection mode. To ensure better thermal efficiency and stable operation of the engine, the pilot injection timings of 20°CA, 30°CA and 40°CA BTDC was chosen with 10°CA BTDC as main injection timing for 5%, 10% and 15% pilot quantity. From analysis it was found that simultaneous reduction of NO_x and smoke emissions can be achieved using the combination of pilot injection timing and pilot quantity with improvement in BTE.

CONTENTS

LIST OF FIGURES

List of Tables

NOMENCLATURE

Symbol	**Expansion**
BTE	Brake thermal efficiency
BSEC	Brake specific energy consumption
Bxx	Contains Biodiesel blend by xx % of volume
°CA or CAD	Crank angles degrees
CI/ SI	Compression ignition/ Spark ignition
CO	Carbon monoxide
CO_2	Carbon dioxide
ECU	Electronic control unit
EGR	Exhaust gas recirculation
HC	Hydrocarbon emissions
NHRR	Net heat release rate
CMHR	cumulative heat release rate
H_2O	Water
ICE	Internal combustion engine
IMEP/BMEP	Indicated/brake mean effective pressure
LTC	Low temperature combustion
NOx	Oxides of nitrogen
O_2	Oxygen
RPM	rotations per minute

SOC	Start of combustion
SOI	Start of injection
BTDC	Before top dead centre
ATDC	After top dead centre

CHAPTER 1

INTRODUCTION

1.1 General Background

Energy is a measure of the prosperity of a country, Economy is the wheel of development and natural resources are the driving force for the better economy. Industrialization and motorization in a developing country like India is highly dependent on a limited and finite reserve of crude oil resources. It is forecasted from many literature/research that, the deposition of crude oil resources may end up in coming decades (Agarwal A K 2009) and a large void has existed between crude oil consumption and production. The Currently available resources are not enough to reduce the gap between energy consumption and production in the field as shown in Figure 1.1(Energy Information Administration 2017). If this current trend continues and no attempts are made to fill this void, it may lead to the *energy crisis*. In the view to overcome this problem, India imports major portion of crude oil to satisfy increasing demand and supply. The crude oil price during the year 2002-03 was US$ 26.65 per barrel(bbl.) and it raised $111.89 for the year 2011-12 and India have imported 180 MMT crude oil worth around Rs.9 lakh crore for the financial year ending 2014-15 (Ministry of Petroleum, Govt. of India). Continuation of a lot of oil imports may result in foreign exchange crisis. Hence there is an urgent need to address this energy security problem and look for some alternative sources of fuel which have huge potential to make country self-sustainable in this area.

Compared to traditional gasoline engines, diesel-powered engines are widely used in transportation and power generating sectors due to their high thermal efficiency, durability and low fuel consumption (Labecki 2012 and Benajes 2012). But diesel engines churn out harmful and hazardous emissions like particulate matter (PM) and nitrogen oxides (NO_x). India has around more than 150 million vehicles on road (Ministry of RoadTransport, Govt. of India) and is the biggest consumers of petroleum products which are illustrated in Figure 1.2 (Sunil Kumar 2012). Diesel vehicles are vastly used in urban areas for applications such as railway, road, agriculture (tractor, pump sets) and captive generation. In the recent past, on-going vehicle population explosion has accounted an environmental pollution than any other sector like electricity generation sector and placed great strain on the country's environment, causing serious air pollution problems especially in metro cities.

With increasing energy demands, expected depletion of fossil fuels and their ever increasing prices along with degradation of the environment, it has become necessary to search for alternative fuels, which promise a harmonious correlation with sustainable development, energy conservation, efficiency and environmental preservation. With these uphill tasks and challenges, the search resulted in bio-origin renewable fuels which can be produced from resources available locally within the region such has vegetable oil, biodiesel or alcohols that may provide feasible solutions to present crisis (2015).

Biofuels production can also provide new income and employment opportunities in rural areas, as its use is developing favourably worldwide. Many developed and developing countries are pinning their hopes on biofuels and have launched the programme with a view to using it as a fuel as a result biofuel is gaining acceptance and a market share against mineral diesel. Biodiesel is defined by ASTM as *"a fuel comprised of mono-alkyl esters of long-chain fatty acids derived from vegetable oils or animal fats, designated B100"* (Nabi 2009).

In the present context, a renewable alternative fuel i.e. ***Biodiesel*** usage can be considered in conventional diesel engines. Experts believe that this can be done with minimum modifications in the engine or in the fuel supply system. Biodiesel has comparable energy density and cetane number with diesel, which virtually decreases all regulated emissions and urgent need, is to develop advanced technologies and concepts in this area to optimize the various engine parameters that will result in cleaner, more efficient diesel engines (Agarwal A K 2009).

Biodiesel can be blended in any proportion with mineral diesel to create a biodiesel blend or can be used in its pure form and use of biodiesel in conventional diesel engines results in a decrease of particulate matter (PM) and carbon monoxide (CO), but a slight increase of oxides of nitrogen (NO_x).The thermal efficiency of an engine operating on biodiesel is generally better than that diesel operation (Agarwal A K 2007).The worried increasing level of NO_x can be answered by using Low temperature combustion (LTC) mode that offers a promising solution to reduce the formation of NO_x and soot simultaneously without affecting the performance.

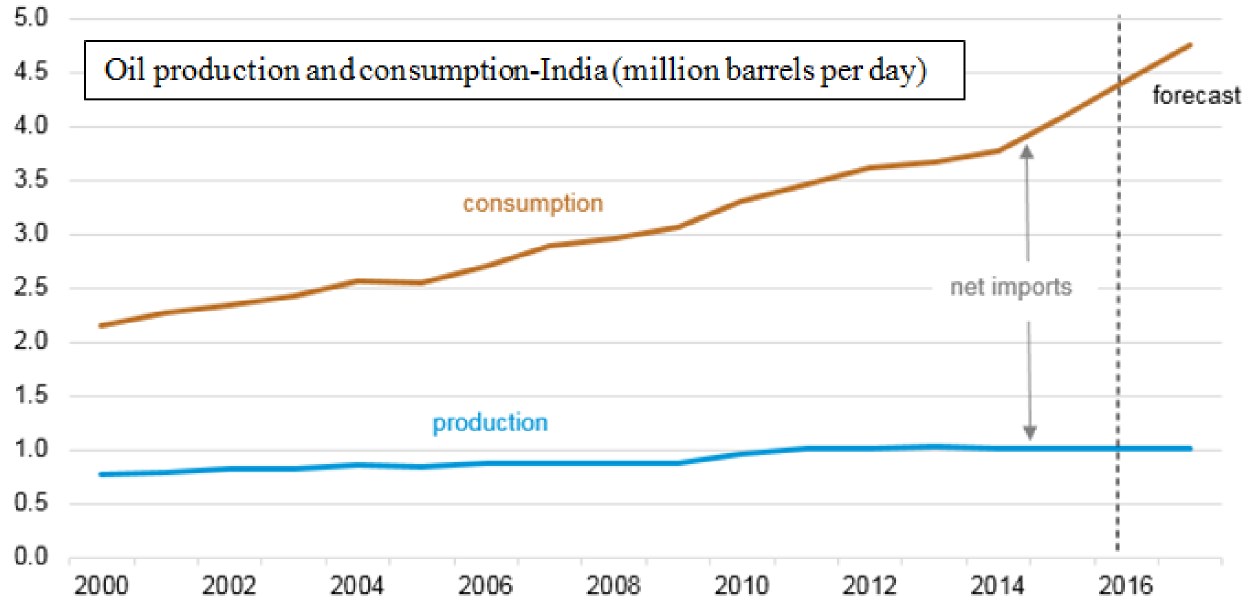


Fig.1.1 crude oil production and consumption with respect to Indian context

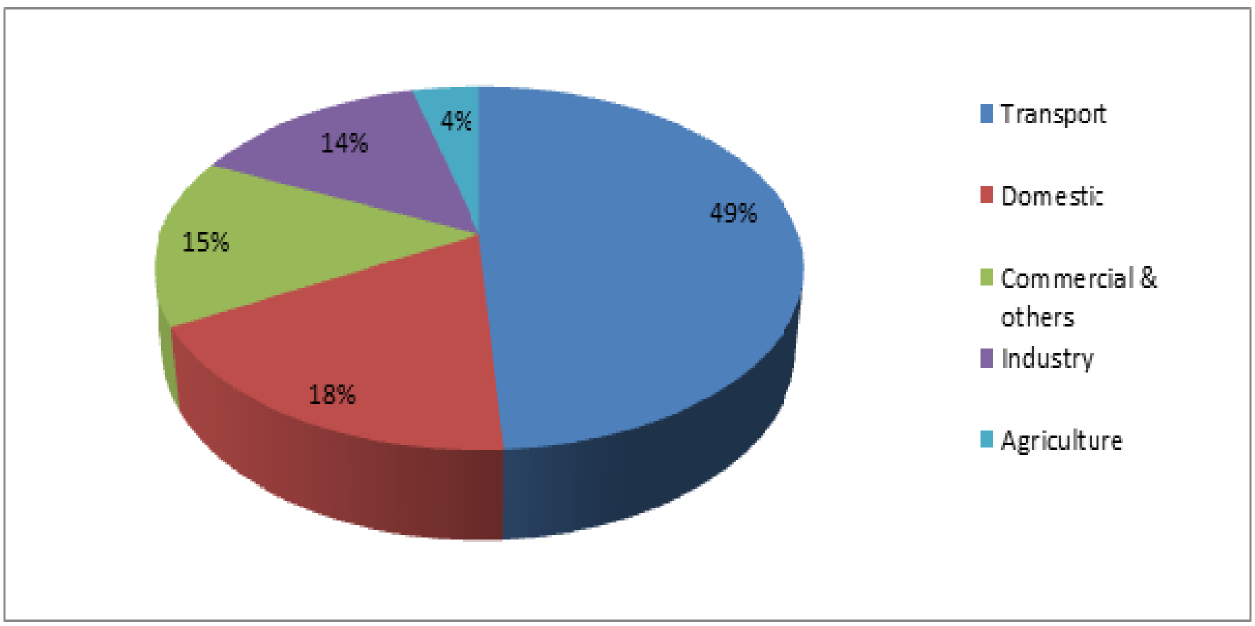


Fig.1.2 Illustration of sector wise utilization of petroleum products

1.2 Emission formation, sources and norms

Deterioration of natural air quality by the intrusion any unwanted material into our atmosphere that will have a deleterious effect on life upon our planet can be termed as *air pollution.* The air pollution is caused by several sources like urbanization and rapid industrialization in many different forms which include automobile emissions and smoke from industry. The principal sources of pollutants formed from internal combustion engines during combustion of fuel are carbon monoxide (CO), unburned hydrocarbons (UBHC), particulate matters (PM) and oxides of nitrogen (NO_x) which is a mixture of NO and NO_2. Other engine emissions include aldehydes such as formaldehyde and acetaldehyde primarily from the alcohol fuelled engines, benzene and polyaromatic hydrocarbons (PAH).

1.2.1 Carbon monoxide

Carbon monoxide occurs during the combustion process, it is a product of incomplete combustion due to an insufficient amount of oxygen in the air-fuel mixture or inadequate time in the cycle for complete combustion. It is also formed in the exhaust if the oxidation of CO to CO_2 is not complete.

1.2.2 Hydrocarbons

Hydrocarbons, also known as organic emissions are produced as a result of incomplete combustion of any hydrocarbon fuel. The concentration of UBHC is usually specified in terms of total hydrocarbons concentration expressed in parts per million atoms. These Unburned HC emissions present in engine exhaust originate because of several processes like flame quenching in the combustion chamber when fuel gets away in narrow passages and incomplete oxidation of fuel that gets trapped in the oil film.

1.2.3 Nitrogen Oxides (NO_x)

Nitric oxide (NO) and nitrogen oxide (NO_2) combined together is called oxides of nitrogen. High temperature prevailing inside combustion chamber assists in combining the species of nitrogen and oxygen together. Hence high mean gas temperatures and oxygen availability are the two important causes for the creation of NO_x. Kinetics of NO_x

generation is governed by Zeldovich mechanism, and its materialization is vastly dependent on temperature and availability of oxygen (Heywood 1988 and A K Agarwal 2007). In C I engine combustion, three probable sources of NO_x formation are

(i)Thermal NO_x, *(ii) Fuel* NO_x, *(iii) Prompt* NO_x

The first type of source i.e. Thermal NO_x is generated at very high combustion temperatures, commonly above 2000°C and this is due to oxidation process of the diatomic nitrogen present in the air. Thermal NO_x is the principal source associated to another type in I C engines and is largely reliant on the temperature of combustion and the residence time at that temperature. The second type of source, fuel NO_x is generated when the nitrogen arising from fuel structure combine together with the oxygen in the air. Prompt NO_x is formed during earlier phase of combustion due to chemical reaction of atmospheric nitrogen with radicals in the air (Kent 2012 and 2017)

1.2.4 Soot and particulate matter emissions

Visible smoke in the exhaust gases generally consists of Particulate matter (PM), particulates in general are any matter that can be collected by filtering the exhaust, expect water. They can be further classified as

- Solid carbon material or soot
- Hydrocarbons which get condensed and their partial oxidation products

Combustion of rich mixture produces soot which is a carbonaceous particulate matter. The appearance of black smoke emissions in the exhaust indicates a high concentration of soot (Pundir 2012).

Particulate emissions from the diesel engines are a major health concern and are more difficult to control. Soot emissions have been associated with respiratory problems and are thought to be carcinogenic in nature. The particle size is important as the particles smaller than 2.5 μ can reach lungs along with the inhaled air and cause health problems. The particles smaller than 2.5 μ constitute more than 90 percentage mass of the total particulate matter in the diesel exhaust (Agarwal A K 2007).

1.2.5 Emission standards

To control air pollutants from internal combustion engines (S I and C I engines), Government of India has introduced the emissions standards with the name Bharat stage emission standards (BSES) in the year 1991 (Pundir 2012). BSES sets the standards and the timeline for implementation, the typical Bharat stage emission norms for light commercial vehicles is shown in Figure 1.3 **(**Vance Wagner 2013).

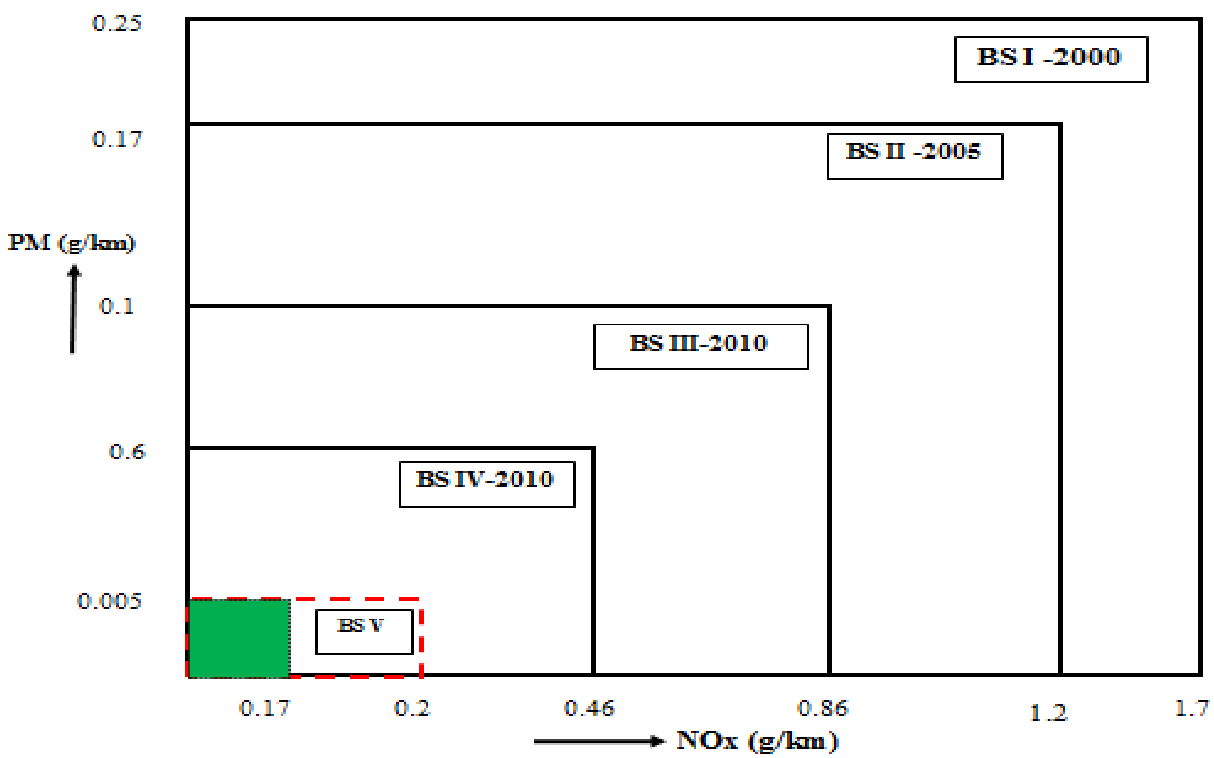


Fig.1.3 various emission standards for light commercial vehicles in India

With the intention to prevent emissions polluting the atmosphere, the United States of America (USA) has implemented the vehicle emission standards in the year 1965 for the first time and then enacted legislation in 1968 to implement nationwide vehicle emission regulations. In Europe, vehicle emission standards were implemented during the year 1970. Progressively the emission standards have become more stringent and have driven the development of advanced engine designs and emission control technology. Now, nearly all the countries all over the world have enforced vehicle emission regulations of varying severity following largely either the US or the European regulations, In India, emission norms were implemented by following European standards and Bharat stage standards are equivalent European standards (Pundir 2012).

1.2.5.1 Regulated and unregulated pollutants

Fossil fuels Combustion leads to the emission of various pollutants that can be further categorized as regulated and non-regulated pollutants. The pollutants on which limits have been imposed by certain environmental legislations like USEPA, EURO and BSES norms are categorized as regulated pollutants (such as NO_x, CO, HC, particulate matter (PM)) whereas the others are characterized under non-regulated pollutants which include formaldehyde, benzene, toluene, xylene (BTX), aldehydes, SO_2, CO_2, methane etc. (Agarwal A K 2007).

These pollutants (regulated as well as unregulated) have to several short- and long-term health effects on humans. CO, nitrogen oxides, PM, (primarily regulated pollutants) formaldehyde etc., generally cause short term effects while poly-aromatic hydrocarbons (PAH’s), BTX, formaldehyde, (primarily unregulated pollutants) etc. are responsible for long term effects. CO is lethal in a large dosage as it aggravates heart disorders and affects the central nervous system. It forms carboxyhemoglobin when in contact with blood which impairs is oxygen-carrying capacity. Nitrogen oxides cause irritation in the respiratory region. Hydrocarbons cause coughing drowsiness and eye irritation (Agawam A K 2007).

1.3 Emission Control Strategies

CI engine’s exhaust emission can be controlled by using in-cylinder and also with after treatment devices, here is a brief overview of NO_x and PM controlling strategies are shown in figure 1.4 and 1.5

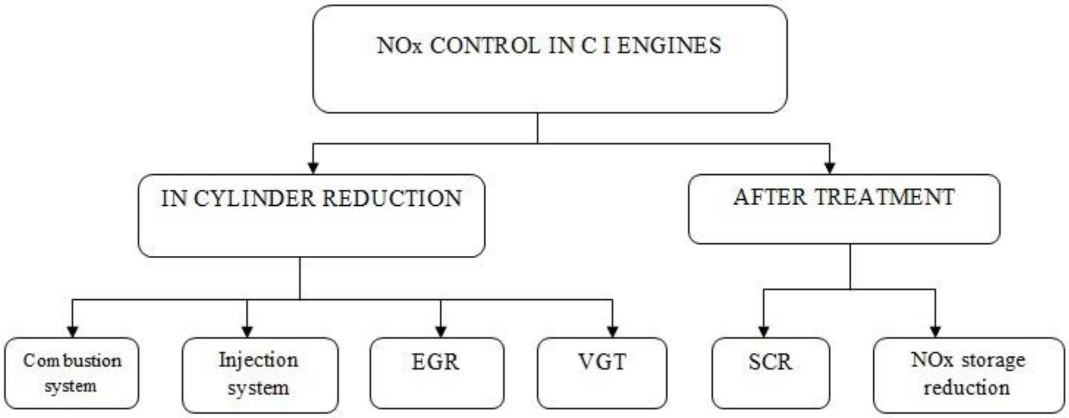


Figure 1.4various technologies used for NO_x reduction in C I engines

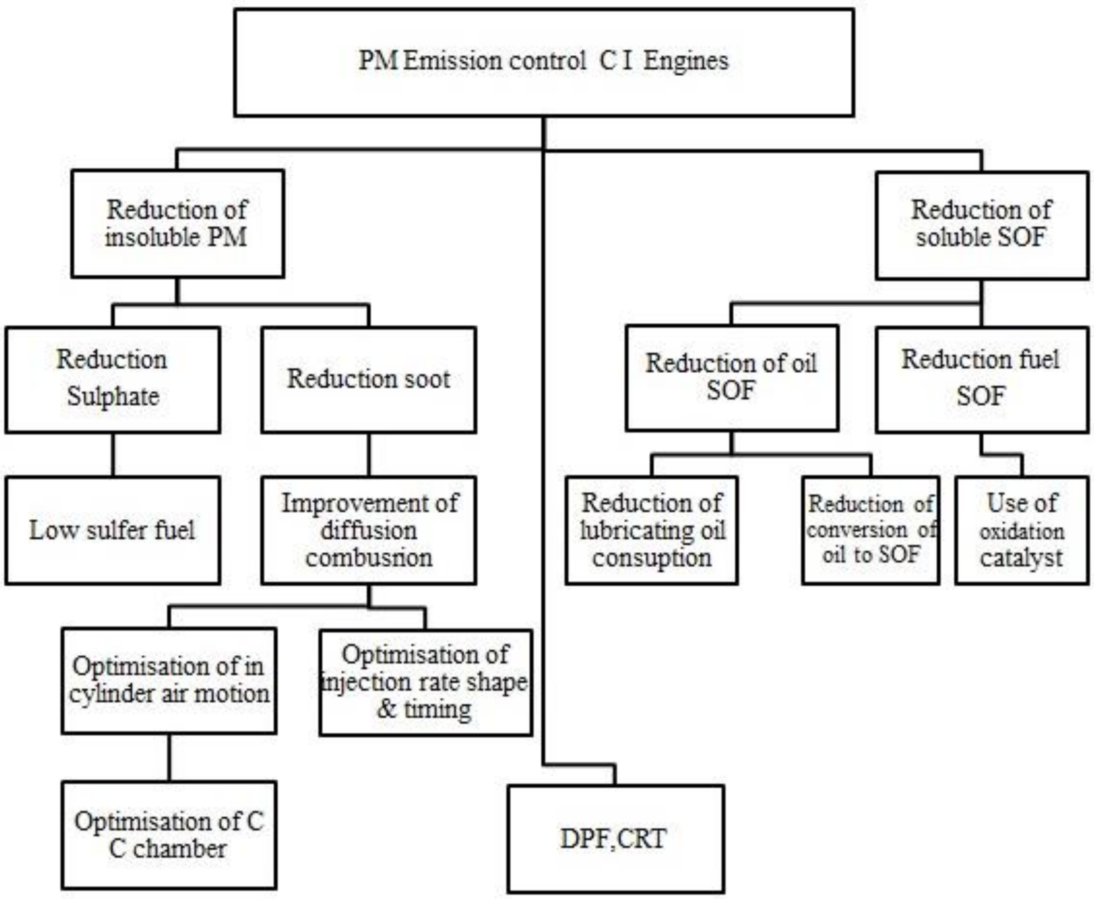


Figure 1.5various technologies used for PM reduction in C I engines

1.3.1 Diesel After Treatment Systems

The in-cylinder methods reduce the pollution to some extent and remaining can be reduced by after-treatment devices (ATD). ATD mitigate the exhaust pollutants and these techniques have concentrated on reducing emissions such as PM, NO_x, HC and CO.

1.3.1.1 Diesel particulate filter (DPF)

Carbonaceous component of PM released from vehicular exhausts is eliminated by the DPF device.

1.3.1.2 Diesel oxidation catalysts (DOC)

The THC and CO emissions from the exhaust can be removed by DOC through catalyzed oxidations. Exhaust gases react with oxygen on the catalyst bed to produce CO_2 and water. In fact, by simple oxidation of diesel exhaust by a standard DOC can reduce PM up to 30%, hydrocarbon based soluble organic fraction (SOF) by around 70%, and THC and CO by over 80% content.

1.3.1.3 Lean NO_x trap (LNT)

The working principle of an LNT includes storage of NO_x under lean burning conditions and reducing it when the engine runs rich, using substrates that are coated with precious metals such as Ba and Pt. During fuel-rich operation CO and THC are used as reductants to convert NO_x into N_2, CO_2 and water.

1.3.1.4 Selective catalytic reduction (SCR)

Selective catalytic reduction (SCR) systems use a homogeneously extruded substrate which is coated with a catalyst and a reducing reagent to convert NO_x into molecular nitrogen and oxygen in the exhaust line of mobile source applications, an aqueous urea (AdBlue) solution is the most preferred reductant.

1.4 Low temperature combustion (LTC)

Presumably, futuristic emission norms would be stricter hence diesel vehicle manufacturers have to switch to more innovative methods which radically reduce the particulate matter (PM) and NO_x emissions. Among the new technologies, in related to in-cylinder strategies are emerging to be a top contender even though there is enormous scope for after-treatment devices.

The modern research on in-cylinder strategies for total emissions controlling and improving the efficiency is mainly based on tactics to reduce in-cylinder combustion temperatures, such a combustion concept is known as low temperature combustion (LTC). In LTC approach, the combustion temperatures are reduced by dilution effect of the in-cylinder combustible charge, by creating fuel-lean mixtures using excess charge or with varying flow rates (low to high levels) of Exhaust Gas Recirculation's (EGR) (Musculus 2013). The goal with low temperature combustion (LTC) engine is to achieve high levels of fuel efficiency without producing harmful emissions. Oxides of nitrogen (NO_x) and particulate matter (PM) are the two main regulated pollutants in diesel combustion.

Figure 1.6 shows the relationship between flame temperature and equivalence ratio for emission formation in C I engines (Valentino 2012). The fuel and air first react in a rich mixture, leading to soot formation, and then this rich mixture burns out in a high-temperature diffusion flame at the jet periphery, leading to NO_x formation (John Dec 2009). NO_x formation and PM is minimized with lower flame temperature and lean-combustion operation respectively. LTC engines burn cool and lean enough (low equivalence ratio) to stay away from the high soot and NO_x formation zones, yet giving high thermal efficiency: they ideally function without a throttle and have high compression ratios (Xingcai Lu 2011). If the temperature during combustion is below 1650 K then both NO_x and soot formation zones are completely avoided regardless of the equivalence ratio. This New combustion concept called low temperature combustion (LTC) is formed on a similar basis to HCCI combustion and was proposed by Sandia National Laboratories, USA (John E.Dec 2009). From then Numerous LTC engine concepts are being investigated at other universities, laboratories, and agencies across the world.

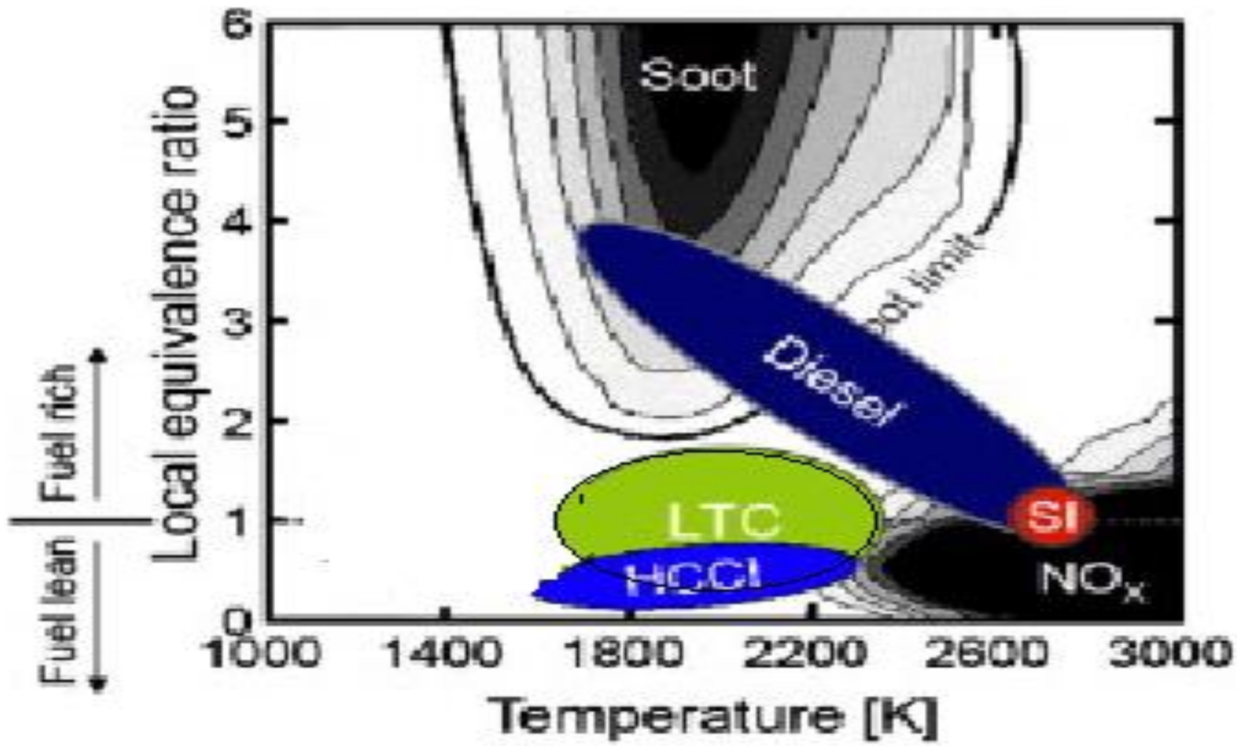


Fig.1.6NO$_x$ and soot emission zone for various types of diesel combustion

Various LTC Strategies are:

- Homogeneous Charge Compression Ignition (HCCI)
- Premixed Charge Compression Ignition (PCCI)
- Reactivity-Controlled Compression Ignition (RCCI)
- Exhaust Gas Recirculation (EGR)
- Early injections
- High injections and boost pressures

1.4.1 Homogeneous Charge Compression Ignition (HCCI)

Combining of SI and CI engines combustion by evading their drawbacks is principally named as HCCI combustion. A conventional spark-ignition (SI) engine charge is a homogenous fuel/air mixture which mixes in the intake port and then undergoes induction compression. Here air-fuel mixture is ignited by with the aid of spark discharge and a load of the engine is controlled by adjusting the throttle. It features tremendously low soot emission but also has lower thermal efficiency due to low compression ratio (Musculus 2013).

In distinction to this, a conventional compression-ignition (CI) engine uses heterogeneous fuel/air mixture. CI engine injects diesel fuel rapidly and directly into the combustion chamber before top dead center (BTDC), after ignition delay the mixture is self-ignited consequently there occurs is less pumping loss. In CI engine ratio generates higher thermal efficiency due to higher compression ratio, the drawbacks to this process is that it exhausts out more amounts of NO_x and PM emissions. The alternative to these combustion methods is a new conception called as HCCI combustion which governs the in-cylinder combustion pressure, temperature and charge mixture will be homogeneous which is auto-ignited at multiple points and reacts homogeneously to burn volumetrically as it is compressed by the piston during upward movement.

Heat release reaction is disseminated all over the combustion chamber without local high-temperature zones or rich-fuel zones. Consequently, a uniform mixture and average low temperature bound the generation of PM and NO_x emissions. HCCI combustion has been a well-thought-out promising solution (Musculus 2013) and investigation for usage of a variety of fuels such as diesel, ethanol, natural gas and others are underway (John 2009).

Over the last two decades, numerous terms have been ascribed to this novel combustion process, including HCCI (Homogeneous-Charge Compression-Ignition), CIHC (Compression-Ignited Homogeneous Charge), PCCI (Premixed-Charge Compression-Ignition), UNIBUS (Uniform Bulky Combustion System), PREDIC (PREmixed lean Diesel Combustion), CAI (Controlled Auto-Ignition), PCI (premixed compression ignition), MK (Modulated Kinetics), and OKP (Optimized Kinetic Process) in four-stroke engines and ARC (Active Radical Combustion), ATAC (Active Thermo-Atmospheric Combustion), TS (Toyota-Soken) for conventional two-stroke engines (Xingcai Lu 2011).

1.4.2 Premixed Charge Compression Ignition (PCCI)

Premixed Charge Compression Ignition (PCCI) concept implies a non-homogeneous air-fuel mixture before combustion and evades soot emissions by injecting fuel early into the combustion chamber. Many obstacles have to be overcome before commercialization of PCCI engine diesel combustion like control of combustion phasing as a result of low volatility, high ignitability of diesel fuel and charge preparation. In this strategy, fuel can be introduced through port fuel injection by late direct injection or early direct injection into the cylinder. Port fuel injection and early direct injection may lead to more amounts emissions in the form of CO and HC as a consequence of incomplete fuel vaporization and fuel spray impingement on the cylinder walls (Xingcai Lu 2011).

1.4.3 Reactivity-Controlled Compression Ignition (RCCI)

The recent advancement in-cylinder combustion techniques like HCCI PCCI and RCCI aims to gain more control over combustion phasing by using multiple fuels of differing reactivity in a CI engine. Multiple injections of these several fuels at scheduled intervals deliver control over the reactivity of the charge during combustion for optimal combustion duration and magnitude. In RCCI combustion method, a low reactivity fuel will be injected early into the engine cycle and which can mix homogeneously with the air. Then a higher reactivity fuel is injected into the cylinder. This approach causes combustion to occur at different times and at different rates by creating pockets of differing air/fuel ratios and reactivity's, RCCI engines are run by multiple injection systems. Permutations of this approach have been explained with all combinations of gasoline, diesel, and natural gas as well as several biofuels and boutique fuels (Kokjohn 2011).

1.4.4 Exhaust Gas Recirculation (EGR)

External exhaust gas recirculation (EGR) is the most preferred in-cylinder technique in current direct injection (DI) compression ignition engines for controlling NO_x emissions. Lower peak combustion flame temperature achieved through this technique is a result of several effects mentioned below.

The dilution or oxygen displacement effect: oxygen mass fraction is reduced because of the replacement of fraction of an oxygen by recirculated exhaust gases during fresh intake air charge using inert gases and causes a spatial broadening of the flame resulting in a reduction of local flame temperature.

Thermal effect: specific heats of gases involving in combustion process play a pivotal role, charge entering engine cylinder is now a mixture of recirculated gases and fresh air which mainly contains CO_2 and H_2O species which have greater specific heat than that of intake air, hence average specific heat capacity of a charge increases.

The chemical effect: Dissociation of CO_2 and H_2O takes place due to an endothermic chemical in combustion zone which will result in a decrease of combustion temperature (Maiboom 2008).

In EGR technique, exhaust gases are reintroduced into the engine intake air, diluting the fuel-air mixture in the combustion chamber and increasing its heat capacity, this reduces the temperature rise during combustion. NO_x formation is suppressed in combustion environments with EGR because peak temperatures are kept below 3000^0C, the temperature above which NO_x is readily formed.

Diesel engines are generally lean burn systems and operate at much leaner (excess air) fuel- air ratios than spark-ignition engines. Hence diesel engines must use much higher proportions of re-circulated exhaust gases in the intake air, especially at lower loads when the fuel-air mixtures contain the largest amount of excess air. To control both NO_x and PM emissions accurately, the amount of recirculated exhaust gas and air entering the engine must be controlled precisely under all operating conditions at different load conditions.

For conventional naturally aspirated Diesel engines the implementation of EGR is straight forward because the exhaust tailpipe backpressure is normally higher than the intake pressure. When a flow passage is devised between the exhaust and the Intake manifolds and regulated with a throttling valve exhaust gas recirculation is established (Zheng.M 2004). The pressure differences generally are sufficient to drive the EGR flow of the desired amount, except during idling whilst a partial throttling in the tailpipe itself can be activated to produce the desired differential pressure. If the exhaust gas is recycled to the intake directly, the operation is called hot EGR. If an EGR cooler is applied to condition the recycled exhaust it is called cooled EGR and is shown in Figure 1.7.But most modern diesel engines, however, are commonly turbocharged, and the implementation of EGR is, therefore, more difficult. Depending on applications a low-pressure loop EGR and high-pressure loop EGR are used (Zheng.M 2004).

Most current diesel engines operate with cooled EGR systems. These systems use a heat exchanger to reduce exhaust gas temperatures before introduction into the intake system, resulting in higher EGR rates and lower combustion temperatures, which results in even lower NO_x formation. Despite their additional cost cooled EGR systems are becoming increasingly common owing to more stringent emissions regulations.

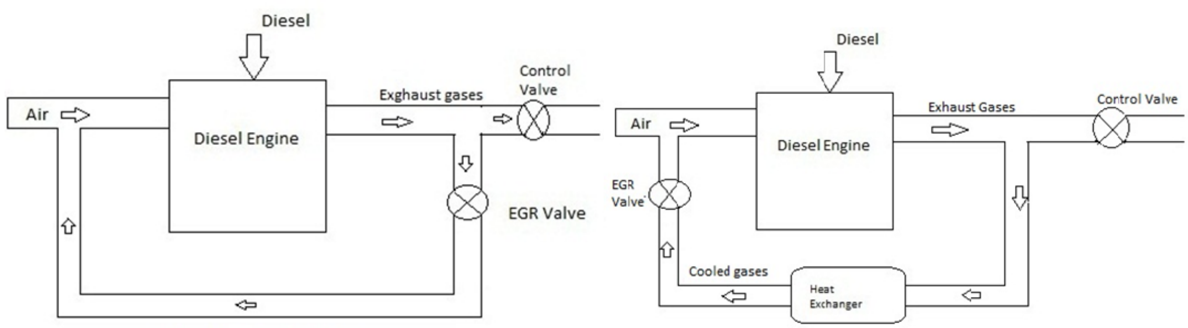


Fig. 1.7 Hot and cooled EGR system

1.5 Common Rail Direct Injection (CRDI) Fuel Injection System

The engine's power, torque, emissions, noise level and specific fuel consumption is determined by the way fuel is injected into the cylinder. The other critical factors on which overall performance of the engine is dependent on fuel's pressure entering the cylinder, the shape and the number of the injections. With the significant improvement in fuel injection and control technologies, mechanical fuel injection systems are being replaced by electronic fuel injection systems.

Optimization of fuel injection parameters is a crucial and powerful tool available for control of NO_x emissions from such electronic fuel injected compression ignition engines e.g. common rail direct injection (CRDI) systems. CRDI means direct injection of the fuel into the cylinders via a single, common line called the common rail which is connected to all the fuel injectors. Common rail is regulated by the electronic engine management, and control the injection timing as well as the amount of fuel injected into each cylinder as a function of the cylinder's actual need, the typical CRDI system is shown in Figure 1.8.

Essentially a CRDI System is designed to provide the usage of moderately higher injection pressures at required fuel pressure and consistently offer flexible and precise control of fuel injection timings to the electronically controlled injectors through a shared fuel reservoir (Dhar 2012). For large-scale efficient utilization of biodiesel in existing CI engines, its effect on performance, emissions and combustion characteristics of modern CRDI engines in a wide range of control parameters needs to be experimentally investigated and optimized.

various injection parameters, including injection timing, injection pressure and injection strategies (multi-injection), can be adjusted very easily and flexibly with common rail injection system (Huang 2015). Multiple injections divide the total quantity of fuel into two or more injections per combustion event. A pilot injection is also usually defined as an injection where 15% or less of the total mass of fuel is injected in the first injection (Mobasheri 2012).

The advantages of CRDI systems compared to conventional direct injection system are

- Required rate of fuel pressure available on demand
- Flexible injection pressure independent of engine speed.
- Higher injection pressures and finer atomization of fuel
- Multiple Injections per cylinder are possible
- precise injection timing is obtained, pilot and post injection increase the combustion quality
- Small, consistent injection quantities for pilot and post injections sequences.
- The flexible timing of beginning and end of injection.
- more power and torque than the normal direct injection engine
- it also offers superior pick up, lower levels of noise and vibration

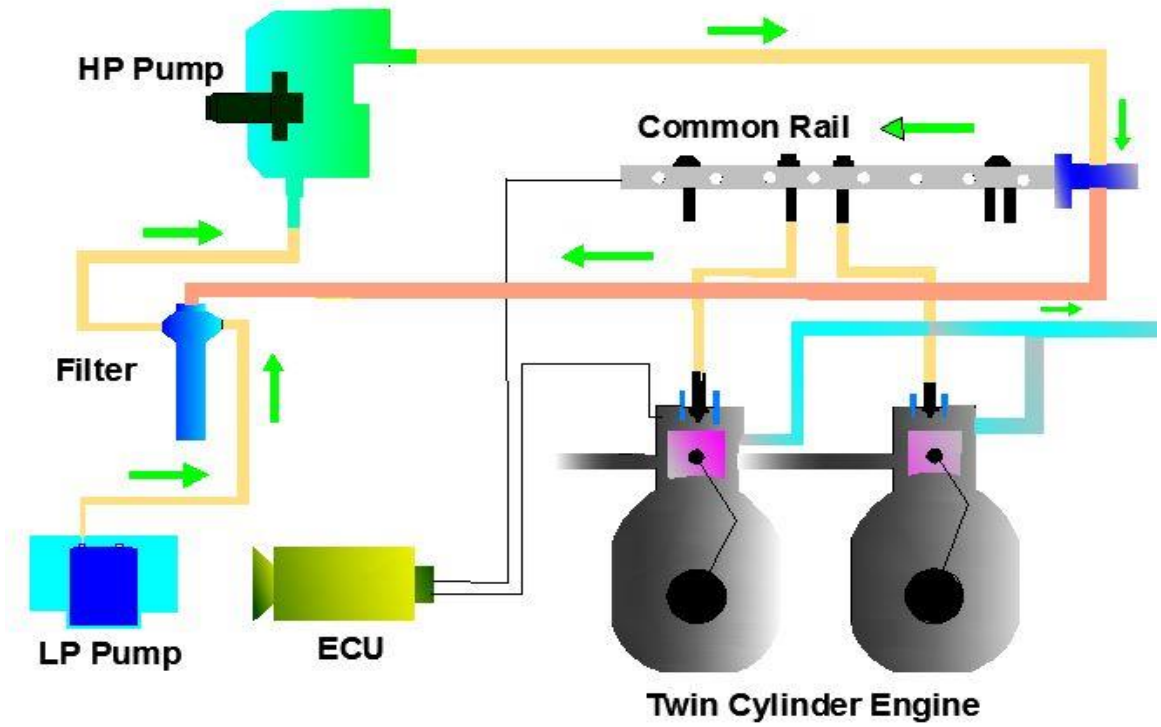


Fig.1.8 Typical CRDI system

Benefits of CRDI systems

- Reduction of overall exhaust emissions
- Reduction of particulates emissions
- Reduction of noise emissions
- Improved fuel efficiency
- Higher performance
- Lower fuel consumption

Disadvantages of CRDI systems are:

The key disadvantages of the CRDI engine are

- It is costlier than the conventional engine.
- Includes high degree of engine maintenance and high-cost spare parts

Applications of CRDI systems

The most common applications of common rail engines are marine and locomotive applications also in the present day they are widely used in a variety of vehicle models

1.6 *Jatropha curcas* biodiesel as a fuel

Apparently, biodiesel is destined to make a substantial contribution to the future energy demands of the domestic and industrial economy. Among the edible and non-edible oil as sources of feedstock for biodiesel production, non-edible oils such as *Jatropha curcas* and Karanja oil reviewed as a promising alternative to diesel fuel in India. However edible vegetable oils such as sunflower, rapeseed, soy and palm oil, extraction and refining process costs are far expensive to be used as fuel and these are also used as food source crops in India.

Therefore, production of biodiesel from non-edible oils is an effective way to overcome all the associated problems with edible oils. However, the potential of converting non-edible oil into biodiesel must be well examined, because physical and chemical properties of biodiesel produced from any feedstock must comply with the norms of ASTM specifications for biodiesel fuels.

Jatropha curcas is an oilseed produced from small tree or large shrub, up to 5–7 meter tall, belonging to the Euphorbiaceae family, the genus name Jatropha derives from the Greek *jatrós* (doctor), *trophé* (food), which implies medicinal uses and curcas is the common name for physic nut in Malabar, India.

It is a drought-resistant plant capable of surviving in abandoned and fallowed agricultural lands. With respect to botanical features, the height of the tree can reach up to 3-8 meter, but can also attain between 8-10 meters. With regard to ecological requirements, Jatropha grows almost anywhere, that includes saline soils, gravelly and sandy soil, crevices of rocks. The leaves shed during winter and mulched around the base of the plant. The organic matter from shed leaves boosts earthworm activity in the soil around the root zone of the plants, which improves the fertility of the soil.

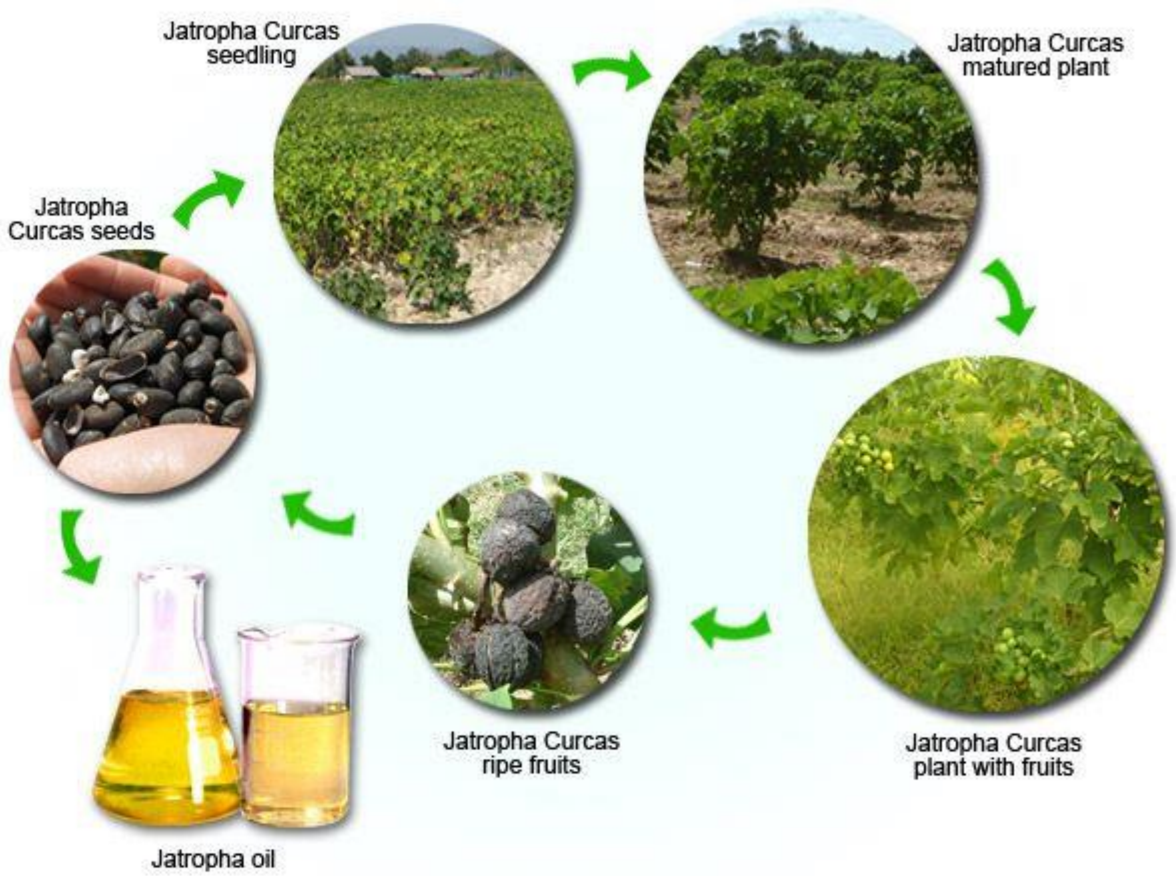


Fig.1.9 Jatropha crop cycle

Jatropha is a tropical plant that thrives in most of the climatic zones with rainfall of 250–1200 mm. It is found in many parts of the country, can survive with minimum inputs and is easy to propagate. The ideal plant density per hectare is 2500 and it produces seeds after 12 months and reaches its maximum productivity by 5 years and can live up to 30–50 years. The production rate of seed is around 3.5 tons per hectare (seed production ranges from about 0.4 tons/hectare in the first year to over 5tons/hectare after 3 years). Depending on the variety, the decorticated seed of Jatropha contains approximately 45–60% of the oil. Jatropha curcas or physic nut contains approximately 170 known species. A typical Jatropha crop cycle is shown in Figure 1.9 (Shirke biofuels 2017).

Regarding climate, Jatropha curcas is found in the tropics and subtropical regions, it favours heat, although it does grow well even in lower temperature and can survive a light frost. The requirement of water is enormously low and in the case of drought, it can stand for extended periods by shedding most of its leaves to reduce transpiration loss. Jatropha is also suitable for preventing soil erosion and shifting of sand dunes.

Cultivation, as discussed in the previous section, is uncomplicated, when production is concerned it completes germination in 9 days and with proper farming *Jatropha curcas* start yielding from 9-12 months' time, and best yields are obtained only after 2-3 years' time. Average life span of a plant is 60 years while producing up to 50 years.

Jatropha curcas seeds with correct management, soil and plant nutrition along with adequate moisture seeds can achieve the yields 4.5, 9, 13.5 and 15 tons respectively over 4 years planted 3000 trees per hectare.

The main advantages of biodiesel include (i) Renewable, sustainable and are environmentally friendly (ii) The fuel-bound oxygen helps proper combustion and affects combustion temperature and emission formation. (iii) Higher cetane number, lower sulphur content, and lower aromatic content (iv) Increased lubricity decreases wear of fuel injection system. The use of biofuels will also balance the climatic changes due to reduced greenhouse gas emissions.

The disadvantage is due to high viscosity and low volatility of vegetable fuels which affects fuel atomization and spray pattern leading several problems like incomplete combustion, rigorous carbon deposition and injector choking. The methods that can reduce the viscosity are blending with diesel, emulsification, pyrolysis, and trans-esterification. Trans-esterification is the frequently used commercial procedure to create clean and environmentally friendly fuel (Forson 2004 and Agarwal 2007). However, this adds extra cost to processing as the trans-esterification reactions involve chemicals and heat inputs (Dovebiotech 2017).

1.7 ORGANIZATION OF THE THESIS

This thesis has been presented with the detailed experimental investigation carried out on performance, emission and combustion characteristics of CRDI engine for *Jatropha curcas* blends with conventional and Low temperature combustion mode, thesis comprises of following chapters.

Chapter 1: Introduction: - This chapter gives a brief introduction about present scenario related to energy and fuel crisis and environmental degradation. This chapter consists of of sources of, norms, formation and their controlling strategies including various approaches of LTC. It includes about advantages, utilization of biodiesel and its blends as an alternative fuel in CI engines. It also consists of different fuel injection techniques and about working principle of CRDI engine with its benefits.

Chapter 2: Literature Review: - Briefly discusses the comprehensive literature relating to the research in biodiesel usage, LTC. It also comprised of variation of engine performance, emission and combustion characteristics for different fuel injection pressure, fuel injection timing, EGR rate and with multiple injections is also mentioned.

Chapter 3: Objectives: - Based on the literature survey and research gap, the objectives of the present work and methodology are framed in this chapter.

Chapter 4: - Experimental setup and research methodology: This chapter describes the development of experimental test rig and the explanation of instruments used during experiments. The details of the CRDI engine set up, various parameter measurement techniques, scheme, procedure and methodology of experiments with a brief introduction about combustion parameters are given in this chapter. The uncertainty and errors involved in the experimentations are also analyzed.

Chapter 5: Results and Discussion: - This chapter deals with discussion of the results obtained from the experimental investigation. It consists of results obtained for engine performance, emission and combustion characteristics with neat diesel in compared to Jatropha biodiesel blends and also outlines application of various EGR rates for different fuel injection pressure, fuel injection timing with various load condition. The detailed discussion when engine operated with high-pressure fuel injection for multiple injection modes with and without LTC mode is also included in this chapter.

Chapter 6: Conclusions and scope for future work: - This chapter makes the concluding remarks of the present investigation and scopes for future work have been listed.

CHAPTER 2

LITERATURE REVIEW

Limited availability of crude oil reserves, day by increasing prices of fuel, environmental degradation by automobiles and most stringent norms for them have given new dimension in vehicle technology up gradation and many research activities over world are going in this field with two main objectives. First one for alternative fuel that may substitute conventional one, secondly on the processes that reduces the exhaust emission as well improves performance. With these concerns and objectives, an exhaustive literature survey has been carried out in the area considering biodiesel as an alternative fuel with single and multiple injection mode. Literature review continued for analysis of CI engines performance, emission and combustion characteristics for different fuel injection pressure (conventional and high pressure), and fuel injection timings with conventional as well low tempearature combustion mode and summarized.

2.1 Performance emission and combustion analysis of a CI engine with biodiesel as alternative fuel

Agarwal A K (2007) summarized about world energy scenario and published that, the world is facing two major crises one is the fast depletion petroleum reserves and another surrounding air contamination. Extraction and continuous consumption of fixed available amount fuel reserves have accounted for reduction in underground stored carbon resources. Hence exploration of alternative fuels which are renewable, energy efficient and environmental friendly is suitable in the present context. He further states that the fuels of biological origin can provide a best solution to this worldwide fuel crisis. Conventional fuel-driven automobiles are the major contributors of greenhouse gases (GHG) emission and timely assessment on case-to-case basis required for the benefit of interest and specific applications. Bio-origin renewable fuels have the potential of replacing mineral-based fuels when transportation sector is considered and it is noted to be one of the major consumer of fossil fuels and major contributor to environmental pollution.

Thermal efficiency of an engine operating on biodiesel is generally better than diesel powered engines and results show that there is 2% increase in peak thermal efficiency at B20 biodiesel blend. Since biodiesel is a oxygen rich fuel results in less particulate formation and emission. Smoke opacity(direct measure of smoke and soot), hydrocarbon, CO emissions are much lower, whereas slight increase in NO_x emission is reported in case of biodiesel usage compared to diesel fuelled engine.

Sunil Kumar et al. (2012) illustrated with the present scenario of increasing petroleum products usage, crude oil import dependency may increase as high as 92% by the year 2020. A two-phase National Mission on Biodiesel was proposed by Planning Commission in 2003 with the following objectives:

(a) Phase-I: A Demonstration Project supposed to be implemented by the year 2006–2007.

(b) Phase-II - Production of large scale Biodiesel

Main objective was to develop a self-sustainable cycle reaching up to retailing of biodiesel, targeting 20% blending of petrodiesel by 2012. They proposed Jatropha plantations on 4,00,000 hectares of waste land by 2007 but total mission fail to take up till date for several reasons.

Estimated cost analysis from the available data, it is evident that cost of jatropha biodiesel is approximately Rs.46/liter, while in the past biodiesel was sold between Rs.30–32 per liter approximately.

Community involvement

In a view of developing country rural background which house a large population, several mass scale employment schemes have been evolved as a part of social security. The scheme aims at providing employment at proposed Jatropha plantations which comprise 62% of the expenditure on plantation in the form of direct wages to unskilled labour. Employment schemes offer 311 person days (first three years) for One hectare of plantation during the implementation of the project and followed by 40 man days per year on a long-term basis. Thus any bio-fuel promotion policy can be considered as the mass employment scheme, also social schemes like MNREGA can be integrated with it.

Sunil Kumar et al. (2012), Interprets that among several alternatives to conventional fuels, Jatropha biodiesel looks a favourable option for India. Jatropha curcas plant is a drought-resistant, perennial plant living up to 50 years and has the capability to grow almost anywhere, even on gravelly, sandy and saline soils. It needs little irrigation and can be grown in all types of soils along with arid land where it offers the additional benefit of erosion control thus making Jatropha a more sustainable choice than other vegetable oils. Advantages of Jatropha are, short gestation period, pest resistant, produces non-edible oil and the byproducts of biodiesel are also quite useful as biofertilizer and glycerin.

The seed harvest period of Jatropha does not fall in the rainy season in June–July when most agricultural activities takes place. This makes it possible for people to generate additional income through the year. India's vast wasteland area cover is best suited for Jatropha cultivation, it can generate large volume of biodiesel feedstock, provided production cost becomes compatible with the price of conventional fuel. According to Indian scenario sustainability parameters when critically analyzed certain issues has been identified and it can be regrouped in four major categories: technological, environmental, economical and social.

Technological issues

For successful adoption and promotion of Jatropha curcas as a biodiesel, the most critical issue is it’s physical and chemical properties, the fuel property comparison is listed in Table 2.1. Desirable properties which favours the use of Jatropha biodiesel as engine fuel are such as higher cetane number, low sulfur content, built-in oxygen and higher flash point. Biodiesel’s particulate reducing effect could be attributed to its lower aromatic and short-chain paraffin HC and higher oxygen content. Study suggests that carbon deposition on the cylinder head, piston top, piston ring grooves, and injector of Jatropha biodiesel-fueled engine are substantially lower compared to the CI engine.

Table 2.1 Properties of diesel and Jatropha biodiesel

Fuel properties	Diesel	Jatropha biodiesel
Density (kg/m^3)	830-842	850-880
Calorific value (MJ/kg)	42-45	38-40
Kinematic viscosity at 40 ^{0}C	~3	~5
Flash point	70-80 ^{0}C	180-200 ^{0}C
Pour point	~7	~4
Cetane number	45 to 52	52–55
carbon residue	~0.10%	~0.01%
Ash content	~0.01%	~0.014%
Sulfur content	~0.25%	<0.001%
Carbon	84-86%	75-80%
Oxygen	~1.2%	~10-12%
Hydrogen	~12.8%	~11.9%

Environmental issues

Emission reduction technique which includes Exhaust gas recirculation can reduce NO_x emissions up to 50% and smoke emissions by 15%. The emission of aromatic and polyaromatic compounds, as well as their toxic and mutagenic effect, has been generally reported to be less with biodiesel. Experimental evidences suggest that carcinogenic potential of exhaust components emitted in terms of metals diminishes in biodiesel fuelled exhaust compared to CI engine exhaust.

Economical issues

A carefull consideration is requird while adopting of any new clean fuel technology/biofuel with issues related to economics such production cost, cost benefit examination as equated to convetional petrodiesel with the expected employment generation and available for long-term along with other viable possibilities. search for alternative fuel mainly depend on instabilities in the cost and difference between demand supply rather than any other issue

Social issues

Non- government nodal agencies should be set up to stimulate farming of biodiesel crops mainly Jatropha, this could help in achieving Energy-based economic development concepts in India favorable biofuel policies. Best intensive steps have been taken by government of Chhattisgarh by buy back policy where government purchases one liter jatropha oil at the rate 18 rupees and one kilogram seeds at the rate of 4.50 rupees. Also fallow land belonging to Government should leased out through private entrepreneur for sole purpose with Jatropha plantations. Andhra Pradesh State Government planned cultivation of Jatropha in 15 lakh acres in succeeding years through subsidy at the rate of 90%. Improved Jatropha seeds have been developed to bear the have oil contents nearly 1.5 times of normal seeds by NOVOD (National Oilseed and Vegetable Oil Development Board). However, due to shortage of supply, primarily, these improved Jatropha seeds would be provided merely to Agricultural Universities for multiplication and development.

National Action Plan on Climate Change (NAPCC) has a goal of establishment of forest cover of 6 Mha over degraded land by 2017 and further developing forest cover from 23% to 33% of India's territory. Indian Railway in general and Southern Railway in particular started using biodiesel as an alternate fuel to conventional fuel. Some private sectors have also started working in the field of biodiesel on commercial lines. State sponsored social welfare schemes may be interlinked and successful co-operative dairy business model can be considered for jatroha biodiesel production and supply.

In his conluding remarks states that, research on biodiesel in India is still at infant stage, there is a dire need to adopt severe plans on the technological development for its production, utilization of by products and evaluation in various types of engine with respect to power output, emissions and even malfunctioning etc.

Marina kousoulidou et al. (2010), the work conducted to analyze the influence of biodiesel on a current technology common-rail light-duty diesel engine. The combustion characteristics were analyzed by measuring the in-cylinder pressure and HRR during combustion with use of conventional diesel and a 10% blend of palm oil origin biodiesel (PME) and rapeseed oil (RME). Regulated harmful emissions and fuel economy were measured over steady state tests. There is slight increase in BSFC with use of biodiesel compared to conventional diesel, may be because of non-optimized ignition timing. NO_x, PM emissions totally dependent on local conditions, speed and load capacity.

Ramesh et al. (2009), have tried adopted different methodology in his study to enhance the performance of a Jatropha oil in with water-cooled direct injection single cylinder. Jatropha methyl ester was also blended with methanol and orange oil in different proportions and tested. Experiments were conducted with neat Jatropha oil has shown slightly low thermal efficiency and high exhaust emissions. But overall engine characteristics were considerably improved with the methyl ester of Jatropha oil. Exhaust emissions were also reduced considerably compared to Jatropha oil. Results concluded good performance at part loads and appreciable performance output at higher loads with Methyl ester of Jatropha oil.

An et al. (2013), Studied the performance, combustion and emission characteristics of biodiesel derived from waste cooking oil using a Euro IV four-stroke turbocharged with common rail fuel injection system diesel engine. The effect of biodiesel blend (10%, 20%, 50%, and 100%), engine speed and engine load on the engine performance and emissions were investigated.

The experimental outcome showed that the engine operation with low speed and partial load condition notably affected the combustion and exhaust emission formation processes.

B10 biodiesel evaluation shows, BSFC is same as diesel but some improved performance was observed under some engine operating conditions and marginal increase in BSFC observed with respect to biodiesel @ B50. Huge increase in BSFC was found by 42% and 35% when engine operated with 25% load at 800, 1200 rpm respectively. High thermal efficiency of biofuel was found at 50% and 100% loads due to the enhanced combustion resulted from oxygenated biodiesel, however, the contrasting trend was observed at 25% load due to its higher viscosity.

Palash et al. (2014), An experimental investigation was carried on a four-cylinder CI engine to estimate the performance and emission characteristics of Jatropha biodiesel blends (5%, 10%, 15% and 20%) also with addition of N,N'-diphenyl-1,4-phenylenediamine (DPPD) antioxidant for various speed. The results suggests antioxidant additive could reduce NO_x significantly with a slight difference in terms of engine power and Brake Specific Fuel Consumption (BSFC), including CO and HC emissions. However, when compared to diesel combustion, there is no appreciable change in the HC and CO emissions with the addition of the DPPD additive. By the addition of 0.15% (m) DPPD additive in B5,B10,B15 and B20 Jatropha blends , NO_x emissions reduced by approximately 8%, 3.5%, 14% and 17% respectively along with exhaust gas temperature compared to biodiesel blends without the additive under the full throttle condition.

Chauhan et al. (2012), studies suggested the target to achieve clean fuel in terms of harmful emissions and an a alternative to conventional fuel, biodiesel addition to diesel or pure biodiesel usage is best possible solution for near future. This paper main objective is to study performance, emission and combustion characteristics of biodiesel derived from non-edible Jatropha oil in a dual fuel diesel engine and comparing with baseline results of diesel fuel. brake specific energy consumption (BSFC) was increased with lower brake thermal efficiency with the usage of Jatropha methyl ester (JME) and its blends compared to with neat diesel.
Conversely,exhusts emissions CO, CO_2, HC and smoke were found to be lower with for JME fuel, with higher NO_x emissions than diesel. The concluded results from the experiments recommend the use of non-edible oil like Jatropha could be a good alternative to diesel fuel in CI engine in the near future as far as decentralized energy production is concerned. Overall performance results and emission stats suggests that Jatropha as a fuel blend could be used in a conventional diesel engine without any modification.

Aydin et al. (2010), in this study, ethanol is used as an supplement in an unmodified diesel engine, to understand the possible use of higher percentages of biodiesel. The fuel mixture is abbreviated as B20 when blend ratio is as 20% biodiesel and 80% diesel fuel and 80% biodiesel and 20% ethanol, called here as BE20. The experimental results reveal that the performance of CI engine is improved with also lower emissions by the use of BE20 especially in comparison to B20. The fuel properties of BE20 were quite similar to diesel fuel in its main characteristics. The addition of DME to diesel fuel changes the physicochemical properties of the blends. By using ethanol properties such as low calorific value, density, kinematic viscosity and aromatics fractions of the blends decreased. Simultaneously, H/C ratio and oxygen content of the blends are enhanced, which has some favorable effects on the ignition and combustion of the blends.

Ong et al. (2014), evaluated the engine performance and emissions for Jatropha curcas, Ceibapentandra and Calophylluminophyllum biodiesel on diesel engine. His experiments conducted with 10%, 20%, 30% and 50% biodiesel blend percentage at full throttle load condition. The engine performance study concluded that blend trials studied are suitable alternative for use in diesel engines. Among the trials 10% biodiesel blend selected as best engine performance in terms of engine torque, engine power, fuel consumption and brake thermal efficiency. Biodiesel blends have also shown a significant reduction in exhaust emissions such as CO, $CO_{2,}$ and soot, whereas a slight increase in NO_x emissions.

2.2 Effect EGR on performance emission and combustion analysis of a CI engine

Zheng et al. (2008) conducted experiments on high load engine operating conditions using biodiesel fuels derived from Soy, Canola & yellow grease and compared Engine performance and emission characteristics with that of results obtained from ultra-low sulphur diesel fuel. Observation reported in the paper suggests that the biodiesel fuel effuse higher emissions of nitrogen oxides (NO_x), compared to diesel. Biodiesel-fuelled engines were generally lower in soot, carbon monoxide and un-burnt hydrocarbon emissions. Exhaust gas recirculation (EGR) was then installed to commence low temperature combustion (LTC) mode from low to medium load conditions. An throttling valve has been fitted in the intake to improvise EGR utility by increasing the differential pressure between the intake and exhaust. With the prolonged ignition delay by 50% NO_x as well as soot was reduced simultaneously from the case with 0% EGR at low load conditions. The tests rig used was naturally-aspirated, 4 stroke, single cylinder Direct Injection diesel engine that was coupled to a DC motoring dynamometer with injector Nozzle opening pressure of 204 bars.

Theoretical EGR estimation

$$EGR\ Ratio = \frac{M_c}{M_a + M_f + M_c} \approx \frac{Inatke\ CO_2}{Exhaust\ CO_2}$$

Observation suggests, the accurate quantity of EGR can be assessed by computing carbon dioxide (CO_2) in the Intake and Exhaust

Where M_a = Mass flow rate of the fresh air

M_c = Mass flow rate of recycled gases

M_f = Mass flow rate of the fuel

Under steady state conditions, EGR could also be assessed from following equation

$$\text{EGR Ratio} = 1 - \frac{\text{MAF}_{\text{new}}}{\text{MAF}_{\text{initial}}}$$

Where MAF_{new} = Intake mass air flow with EGR

$MAF_{initial}$ = Intake mass air flow without EGR

Tormas et al. (2010), carried out an experimental investigation on a modified single cylinder engine fuelled alternately with diesel/biodiesel fuel to investigate the potential of high-speed direct injection (HSDI) diesel engines while using biodiesel as an alternative fuel. His works involves use of different combustion enhancement techniques such as EGR when operated at various engine loads, including high EGR rates anticipating future diesel engines to operate it on low tempearature combustion (LTC) region. In the study, it was found that 100% biodiesel and EGR leads minor improvement in indicative efficiency and overall reduction in NO_x, UHC, CO and other PM and exhaust gas temperature

Francisco et al. (2012), Concludes that the rise in NO_x emissions observed while operating with biodiesel is due to the presence of oxygen in biodiesel and according to the Zeldovich mechanism NO_x formation also depends on oxygen content. Experimental and simulation studies were carried out in a single cylinder HCCI engine using mineral diesel oil blends compatible with EN590 and colza biodiesel compatible with EN14214. Fuel injection was fixed at 10° before TDC, known as modulated kinetics (MK) which is a combination of high swirl level and EGR rates. Another fuel injection strategy as advanced injection timing about 50°–45° before top dead center with biodiesel fuel was analyzed. Injection pressure increased to maximum up to ≈650 bar, this change improves mixture homogeneity. All test fuels exhibited increased HC and CO emissions with EGR rate.

Zheng et al. (2004), illustrates that Diesel engines have elevated thermal efficiency because of their higher compression ratio and lower equivalence ratio. The high temperatures required to achieve auto-ignition is produced by using higher compression ratios. NO_x production rate is a strong function of fueling rate and load level of the engine. Incorporating EGR dilutes the O_2 concentration in the combustion chamber; as a result, NO_x production is considerably lowered. However, in diffusion controlled Diesel combustion, Oxygen deficiency for fuel-rich pockets is a major problem during the later stages of combustion, especially while operating on higher loads. This effect gets further aggravated by using EGR and increases the difficulties to burn smoke-free.

He used the equation to calculate mass flow rate EGR i.e

$$EGR\ ratio = \frac{Intake\ CO_2 concentration}{Exhaust\ CO_2\ concentration}$$

Hoekman et al. (2012) reviewed the effect of biodiesel on NO_x emissions and three types NO_x formation mechanisms

Thermal NO_x

It is the predominant contributor to total NO_x formation and is directly proportional to the in-cylinder temperature. The flame temperature for biodiesels is higher in comparison to conventional diesel. A greater percentage of exhaust gas re-circulation is needed while working with biodiesel.

Prompt NO_x

Also known as “fenimoreNO_x” is widespread under fuel rich conditions in the presence of abundant hydrocarbon fragments that react with N_2.

Fuel NO_x

Fuel species containing nitrogen get oxidized to NO_x in the progression of combustion inside a CI engine. Biodiesel commonly reduces HC, CO and PM, but a slight raise in NO_x is typically observed and considered as a setback that requires mitigation. The various mitigation ways discovered by investigators can be roughly categorized in two groups:

(1) Modifications to engine and (2) Modifications to fuel

Author concluded with mentioning about most effective parameters considered important in engine modification include high pressure injection, injection timing, radiative heat transfer, adiabatic flame temperature, ignition delay and EGR.

Qi et al. (2011), tested Ford Lion 4-stroke 6-cylinder direct injection engine, predict that feasible engine modifications parameters may be optimized for engine emissions and combustion characteristics, from their tests following important observations were recorded. BSFC faintly increased by retarding the main injection timing and increasing the EGR rate under various injection timings accompanied by reduced speed and power output. NO_x and combustion temperatures always decrease by retarding the main injection and the increment of EGR rate even by 5%, under different main injection timings, high NO_x reduction efficiency up to 50% is observed with slight increase of BSFC about less than 2%.Further, soot emissions are also reduced at low loads when retarding the main injection timing. It is also seen that HRR and soot emission are marginally greater than before for increased EGR rate at higher loads of the engine.

Labecki et al. (2012) outlines the use of biodiesel in engine is determined by the cultivation region and crop type. Karanja and Jatropha oil are used in India. In his experimental study using multi-cylinder, turbocharged engine, the effects of various operating engine conditions like fuel injection pressure, fuel injection timing and EGR on the performance , combustion and emission of rapeseed oil and its diesel blends have been studied. The smoke number, emissions of gaseous compounds such as CO and THC, NO_x are reduced with minor modifications in combustion parameters.

Musculus et al. (2013), analyzed that Low in-cylinder temperatures are attained by employing EGR, Ignition closely coupled with the fuel injection event, while the vital role is still played by chemical kinetics. He also reviewed that the emissions norms can be reached by making use of in-cylinder approaches like increasing injection pressure of fuel, increasing intake boost, changing fuel injection timing, lowering intake temperatures, using low to moderate levels of exhaust gas recirculation (EGR) and improving combustion chamber design without exhaust gas after treatment systems which are economically a burden to end users.

John (2009), summarizes and outlines that sophisticated compression-ignition (CI) engines can simultaneously produce high efficiencies as well as less NO_x & particulate (PM) emissions. Also, efficiencies are as good as diesel conventional engines, the mixture charge here will be highly diluted or premixed partially with use of low to high-level EGR or operating with lean mixture in the development of these advanced C I engines, Low tempearature combustion (LTC) approach is also one popular method to be used. Hence to obtain dilute LTC, the approach depends on EGR rates and injection timing which is characteristically shifted 10–15^o CA prior or later compared to usual diesel combustion such that temperatures are lesser delaying ignition and thus providing more premixing time.

Saravanan et al. (2013), carried out the experiments on a air cooled single cylinder engine using crude rice bran oil methyl ester (CRBME) biodiesel with 23.4°bTDC standard injection timing, he modified the combustion progression by delaying fuel injection timing in combination with recirculation of exhaust gases, fuel injection pressure and drawn following conclusions from his experiments

- Cylinder pressure dropped down with extended delay period.
- MHRR was retarded as well as increased.
- CO emissions dropped drastically along with minor increase in smoke density.
- UBHC emission and brake thermal efficiency increased significantly, CRBME blend NOx emission are reduced with a minute increment in smoke density on account of this modification process

Percentage EGR calculated using Equation

$$EGR\% = \frac{volume\ of\ air\ without\ EGR - volume\ of\ air\ with\ EGR}{volume\ of\ air\ without\ EGR}$$

Imtenan et al. (2014), stated that simultaneous reduction in diesel emissions including soot and oxides of nitrogen is the key research trend. Even though various technologies have been set up to cut down emissions, in-cylinder reduction techniques such as low tempearature combustion (LTC) will persist to be the most vital in modern diesel engines research. In addition, rising prices and the depletion of diesel fuel have pioneered a growing interest. For this reason, it is estimated that future CI engines will operate on pure biodiesel/biodiesel blends. Being a noteworthy technology to mitigate emissions, emission characteristics of both diesel and biodiesel LTC requires a critical study. Many differences in physical and chemical properties require a critical investigation for diesel and biodiesel emissions from the perspective of LTC attaining strategies.

Chen et al. (2014), studied the combustion and exhaust emission characteristics of using n-butanol/diesel ratio (40% butanol) at 1400 rpm in a heavy-duty diesel engine. In addition, the effect of EGR also was experimentally assessed and compared with pure diesel fuel (Bu00). It was observed that Bu40 has longer ignition delay, greater in-cylinder pressure and quicker burning rate as compared to Bu00. Bu40 has elevated NO_x as the high-temperature combustion region is wider, lesser soot as a result of locally lower equivalence ratio distribution and higher CO because of lesser gas temperature in the later parts of the expansion process. For Bu40, with EGR no noticeable influence on soot is observed but NO_x emissions dramatically reduced. Meanwhile, till the EGR threshold value is reached there are no noteworthy changes in HC, CO emissions and indicated thermal efficiency (ITE).

HC and CO emissions increase severely, and ITE decreases distinctly when EGR rate surpasses the threshold level. The Bu40 threshold appears at a lower level EGR rate when compared with Bu00. Conclusively, merging high butanol/diesel blend ratio with moderate EGR rate has the capability to accomplish very low NO_x and soot emissions at the same time while retaining the high thermal efficiency level.

2.3 Effect fuel injection timing (IT) on performance emission and combustion analysis of a CI engine

Peng et al. (2012), outlines that biodiesel fueling and the modification in fuel injection approach have a substantial impact on CI engine performance and emission characteristics. Biodiesel application will lessen carbon monoxide, particulate matter (PM) and unburnt hydrocarbon emission with increasing oxides of nitrogen (NO_x). The in-cylinder techniques like modification in fuel injection approaches comprising ascending the FIP, advancing the start of fuel injection too can lower particulate matter emissions however nitrogen oxides emissions increases. It is resumed from different research articles that the source of the biofuel NO_x outcome is from a many of coupled mechanisms that is fuel injection pressure, injection timing, combustion phasing, chemical kinetics and premixed-burn fraction five categories grouped together. His anticipated idea was validated by carrying out investigations on Ford DI diesel engine with compression ratio is 17.2. The engine was provided with ECU controlled common rail fuel injection system (1800 bar) and two adjustable geometry turbochargers. From investigations, it is concluded that optimization of an injection strategy by variation of injection pressure and timing in CI direct injection is an efficient manner to prevent exhaust particulate matter and NO_x emissions. SOI is in the ranged from 9^0 BTDC to 3^0 BTDC.

The trade off between PM-NO_x with biodiesel can be effectively countered with lowering fuel injection pressure by keeping SOI constant, with these injection strategy biodiesel NO_x is avoided while preserving a comparable PM or even lower one than mineral diesel. With respect to performance, there is no remarkable change in brake thermal efficiency for biodiesel fueling or modification of fuel injection pressure and is reduced with retardation. Fraction of premixed combustion is higher with Retarding the start of fuel injection after top dead center but at a fixed SOI, apparent net heat release rate is higher and start of combustion (SOC) is little earlier with increase of fuel injection pressure due to the better mixing of air and fuel.

Benajes et al. (2012), Experimented and analyzed the diesel low tempearature combustion (LTC) for early injection timing (-24°aTDC to -33°aTDC) and was able to achieve low NO_x (<35ppm) and lesser PM through use of EGR. An intake oxygen concentration of 12.1% proved to be low enough to reduce the maximum adiabatic flame temperature below 2200 K and the NO_x emissions below 35 ppm. Single cylinder, direct injection, four-stroke type of diesel engine with fuel injection pressure of 1450 bar was used and found that as the injection timing advances from -24°aTDC to -33°aTDC there is an increased PM, HC and CO emissions.

Pandian et al (2011), investigated the effect of injection system parameters such as injection pressure, injection timing and nozzle tip protrusion on the performance and emission characteristics of a twin cylinder water cooled naturally aspirated CIDI engine with biodiesel derived from Pongamia seeds and drawn following conclusions from his study

Advancing the injection timing to 30° before TDC will increase NO_x emissions mean time other emisins like smoke, HC and CO were reduced. Higher NO_x emissions were also observed by increasing the fuel injection but at same condition increase of brake thermal efficiency with lesser BSEC, CO, smoke and HC noticed at all injection timings.

Ganapathy et al. (2011), illustrated experimentally that engine performance and emission characteristics with Jatropha biodiesel largely depend on engine operating parameters and fuel injection timing. Here advancement of fuel injection timing by 5 CAD from original settings increases the BTE and NO_x with lower CO,HC, smoke and BSFC. With respect combustion parametes peak heta release rate, peak cylinder when engine operated for Jatropha curcas blends. Conversely, retardation of injection timing by 5 degree CA results in ascend of CO,smoke, HC and BSFC with lower combustion pressure,NHRR, BTE and NO_x.

HC, CO,BTE and smoke were lower with for blends of Jatropha compared reference operating fuel diesel, for fixed injection timing. However torque, load, speed and peak pressure NO_x and BSFC higher with than reference operating fuel diesel compared to blends of Jatropha. 340 CAD (20°BTDC) is chosen to be best IT for Jatropha biodiesel operation with maximum peak pressure, HRR, BTE and minimum CO,HC, BSFC and smoke. Hence a suitable injection strategy can lead to substantial benefits in terms of emissions, combustion and performance of a CI engine operated with Jatropha biodiesel.

Sayin et al. (2009), remarked that ignition delay is the time between start of injection and start of combustion vary as the fuel injection timing and state air into which fuel is injected. Ignition delay will be longer due to lower initial air and pressure and will be shorter due to higher air pressure and temperature, when injection starts later (when piston closer to TDC). So timing of injection paly major role in engine performance and exhaust emissions.

Engine exhausts emission like HC nad CO greatly vary and increase if EGR rate surpasses the threshold limit with decrease of thermal efficiency. Simultaneous reduction of soot and NO_x to very low level is possible if engie operated by combined effect of EGR nas high butanol blend ratio without affecting efficiency of the engine.

2.4 Effect fuel injection pressure (FIP) on performance emission and combustion analysis of a CI engine

Kannan et al. (2011), studied the effect of biodiesel on engine operating problems like unfavorable pumping, choking, spray characteristics and piston ring staking on a diesel engine. Author also used alcohol along with biodiesel which improves the availability of oxygen and mixing properties in the combustion chamber. For experimental study researcher used blend of 30% of waste cooking palm oil, 10% ethanol and remaining 60% diesel and named as diestrol. Authors varied injection timing and injection pressure to study the performance, combustion emission characteristics. Results revealed that increase in injection pressure to 24MPa and injection timing of 25.5° BTDC, 31.3% of maximum BTE is obtained. Results also showed that 33%, 6.3%, 4.3% and 27.3% reduction in carbon monoxide (CO), carbon dioxide (CO_2), nitric oxide (NO) and smoke emission respectively was obtained when compared to diesel. But there is a slight increase in unburnt hydrocarbon (UHC) was observed. In-cylinder pressure and net heat release rate were increased for diestrol fuel. Lesser ignition delay of 12.7° CA was obtained for diestrol fuel.

Mohanan et al. (2014) used cardanol as an alternative fuel for diesel engine. In their research varied the injection pressure (higher) for a B20 blend along with 10% of methanol to study performance, combustion and emission characteristics of diesel engine. Results showed that greater reduction in CO, HC, and smoke emissions but increase in the oxides of nitrogen (NO_x) emission for 220 bar injection pressure for B20M10 blend when compared with diesel. Also, brake thermal efficiency was maximum at 220 bar for B20M10 blend. Authors were optimized injection pressure of 220 bar for B20M10 blend based on performance and emission characteristics.

Puhan et al. (2009) were used high linolenic linseed oil methyl ester as an alternative fuel to investigate performance and emission characteristics experimentally. Authors varied the injection pressure from 180 bar to 240 bar in steps of 20 bar at constant speed. Results revealed that 240 bar of injection pressure is optimum for linseed methyl ester oil. With this fuel at optimized injection pressure, great reduction in the CO, UBHC and smoke emissions were obtained without much change in thermal efficiency. But NO_x emissions found to be increase with this higher injection pressure for linseed methyl ester oil. The combustion analysis shows that the ignition delay can be reduced at higher injection pressures compared to diesel. The combustion duration was almost same at all the injection pressures. Also in cylinder pressure is maximum for the optimized injection pressure.

Gumus et al.(2012) varied the injection pressure (18, 20, 22, and 24MPa) and engine load (12.5, 25, 37.5, and 50 kPa) condition to study the emission and performance characteristics of diesel engine at constant speed. Different biodiesel blends were used for the experimental analysis. From the experimental research it can be seen that BSFC, carbon dioxide (CO_2), nitrogen oxides (NO_x) and oxygen (O2) emission increased, smoke opacity, unburned hydrocarbon (UHC) and carbon monoxide (CO) emissions decreased due to the fuel properties and combustion characteristics of biodiesel for the standard injection pressure. As the injection pressure increased decrease in BSFC of high percentage biodiesel–diesel blends (such as B20, B50, and B100), smoke opacity, the emissions of CO, UHC was obtained. But emissions like CO_2, O_2 and NO_x were increased. Finally, author concluded that higher biodiesel blends will give the better results in higher injection pressure.

Bhanapurmath et al. (2009) conducted the research in Honge oil, its ester, and ester blends with diesel (B10, B20, B40, B80, and B100) to study the performance, combustion and emission characteristics of the diesel engine. Authors were conducted the experiment in single cylinder diesel engine at constant speed of 1500 rpm. In the experiment, different higher injection pressures (205, 220, 240, 260, and 280 bars) and injection timings (19°, 23°, and 27° BTDC) were used. From the experiment results, it was found that 19° BTDC resulted in better performance characteristics for Honge oil and Honge oil methyl ester. Injection pressure of 260 bar resulted in improved combustion characteristics. Authors concluded that the B20 blend showed promising results in combustion and performance characteristics.

Agarwal et al. (2015) conducted the experimental research in CRDI engine to analyze the performance, combustion and emission characteristics. Authors varied injection timing and injection pressure for Karanja biodiesel (KOME) blends at a constant engine speed. Experimental results revealed that brake thermal efficiency of KOME blends was slightly higher than neat diesel with increase in fuel injection pressures. Lower biodiesel blends showed lower BSCO and BSHC emissions in comparison to neat diesel, however, BSHC and BSCO emissions were found to be higher for some operating conditions for KOME50. BS NO_x emissions of KOME20 and KOME10 were higher than mineral diesel for all FIPs however they were almost identical to mineral diesel for KOME50. Maximum cylinder pressure increased with increasing fuel injection pressure at fixed SOI timing for all test fuels and SOC advanced for lower biodiesel blends in comparison to mineral diesel. From the experimental results, authors concluded that utilization of up to 10% Karanja biodiesel blends in a CRDI engine will improves thermal efficiency with reducing emissions, without any significant hardware changes or ECU recalibration.

2.5 Effect multiple injection on performance emission and combustion analysis of a CI engine

Huang et al. (2015), have analyzed experiment study the effect of multiple injection for different pilot injection timing and pilot injection mass for 25% exhaust gas recirculation condition. The combustion and emission characteristics were evaluated for the common rain diesel engine for pure diesel, B20 and B30 butanol blends. Outcome of multiple injection shows the peak value of heat release rate decreases with pre-injection fuel for advancing the pilot injection timing and the in-cylinder pressure peak value reduces with the rise of maximum pressure rise rate (MPRR), while NO_x and soot emissions reduce.

Heat release greatly deend on pilot injection fuel mass, for pre-injected fuel the peak value of heat release ascends and but descends for the main injection with increase of in-cylinder pressure peak value and NO_x emissions increase. Soot emissions improved with blending of diesel and n-butanol ratio. The increase of n-butanol ratio causes increase in MPRR and advancing crank position for 50% cumulative heat release as well as NO_x and soot emissions decrease when pilot injection is adopted.

Dhar et al. (2015), concluded from his experimental studies with biodiesel blends that latest emission norms can be met in modern CRDI diesel engines with higher thermal efficiency by the application of pilot and post injections. Karanja biodiesel blend of B10, B20 and B50 in diesel were used for examination of a single cylinder engine for various multiple injection. Higher fuel injection pressure of 500 and 1000 bar are used for different main and pilot injection timings were tested. From the results obtained it was observed that Brake specific fuel consumption (BSFC) is higher with concentration blends of Karanja in the diesel. Performance of engine with respect to BTE was marginally higher for biodiesel blends in relation to conventional diesel fuel. Lower emissions in carbon monoxide with hydrocarbon were noticed for lower blends of Karanja. Oxides of nitrogen were more for blend concentration at B10 and B20 and in respect of combustion analysis, combustion duration higher for B50 in comparison to diesel and other blends.

Suh et al.(2011), The experimental analysis was conducted for better understanding of combustion stability low compression ratio single cylinder CI engine. It is noticed that double pilot injection improved performance and engine operated in stable condition. With respect to emissions reduction in HC, soot and NO_x emissions observed with hgher value of CO. for same condition peak value in pressure achieved almost the same level of single injection combustion even though its peak hear release is reduced by approximately 50% compared to single injection combustion.

Huang et al. (2016), made comparative study using four-cylinder diesel engine to study the effect of multiple injection strategy on emissions combustion of a with EGR ratio for neat diesel, gasoline, n-butanol and gasoline/n-butanol blends with diesel for various concentartions were were investigated

It is showed that soot is reduced dramatically with increase of brake specific fuel consumption (BSFC) and for multiple injection strategy involving different pilot injection timing and pilot fule mass ratio. maximum pressure rise rate can be decreased with proper multiple injection strategy by increasing pilot injection ratio with retarded pilot injection timings where fuel is injected near to TDC. Engine emissions like hydrocarbon, soot and carbon monoxide decreased increased and NO_x emissions decreased with larger quantity of pilot ratio. BSFC can be reduced remarkably with smaller pilot-main interval . Continuing, with less pilot injection quantity and longer interval between main and pilot injection wil lower soot emissions. However, the blending of of gasoline or n-butanol with mineral diesel can significantly offset the deterioration of soot emissions, induced by the adoption of pilot injection strategy. Moreover, the variations in soot emissions from the combustion of all the blended fuels are less sensitive to variations in pilot injection ratios and pilot-main intervals compared to diesel. For all the studied fuels, effects of pilot injection ratios on NO_x emissions are less obvious than the effects induced by the pilot-main interval. A decrease in pilot-main interval can further reduce the NO_x emissions. For all the studied fuels, decrease in pilot injection ratios or pilot-main intervals can reduce the emissions of THC and CO, whereas the difference among four fuels is quite small. However, the pilot-main interval shows a more remarkable effect on the THC emissions than the pilot interval ratio.

Park et al.(2015), experimental analysis is carried out using optical diesel engine for multiple injection strategy to find out soot distritions and flame temperature for all biodiesel blends and diesel fuels. The intensive of this work is establish correlation between NO_x formation and flame temperature with both numerical and experimental methods. Work continued with spary distribution in regard to combustion processes of all the test fuels. Though physical and chemical properties of the blends lead to drawbacks in spray development but multiple injection allows biodiesel fuel to improve the atomization process and lower engine exhaust emissions. Following impartant conclusions are highlighted

(a) shorter interval between main and pilot injection increases the combustion quality with maintaining same or better fuel economy. The pilot heat release rate caused due to pilot injection promotes the main spray development for biodiesel blends as a homogeneity increased for combustion.

(b) flame temperature in the combustion chamber is lowered because of it reduces the locally fuel rich zones as air-fuel mixture process improced. Totally, with multiple injection strategy flame temperature distribution is better in the entire combustion chamber, hence NO_x formation chances are lowered. The advanced pilot injections with 40°CA BTDC conditions are results in lower NO_x concentrations due to improved homogeneous combustion in the cylinder.

(c) Biodiesel and its blends spary formed high soot concentration due to comparatively fuel rich zones caused by poor atomization. Conversely, high combustion temperature and oxygen of biodiesel accelerate the oxidation process in the combustion chamber.

Thus, soot emission in the exhaust are reduced in comparision with neat diesel fuel combustion. Additionally, homogeneous combustion initiated by the pilot injection also suppresses soot emission formation.

Roh et al.(2016), In this work, the effects of DME-biodiesel blend, biodiesel-diesel blend and conventional diesel fuel on the combustion and emissions characteristics of a compression ignition engine according to pilot injection were studied. On the basis of experimental results, the conclusions of this study are summarized as follows:

a) The combustion characteristics showed a higher pressure for DME80B20 fuel compared to B80D20 and conventional diesel fuel, while the DME80B20 exhibits a lower peak in pilot injection with the same fuel quantity due to DME's low LHV.
b) The maximum pressure of the pilot injection mode is significantly lower than that of the single injection mode without pilot injection. Based on the results of mean pressure, the pilot injection mode indicates a lower mean effective pressure than that of single injection without pilot injection.
c) The HC emission for the pilot injection mode shows lower emission than that of single injection without pilot injection. As the injection timing is advanced, the concentration of HC emission is similar for all three types of fuel due to the shorter ignition delay and excellent evaporation characteristics.
d) In a pilot injection cycle, the NO_x emissions of DME-biodiesel blended fuel (DME80B20) are higher than those of diesel and diesel biodiesel blended fuel (B20D80), regardless of injection timings. It showed that NO_x emissions of pilot injection mode resulted in the significant increase when compared to the single injection case.
e) In the case of DME80B20 fuel, the soot emission is near zero for both single and pilot injection, while diesel and B20D80 fuel in pilot injection mode exhibit a higher distribution of soot emission than that of DME80B20.

Yao et al. (2010), Article deal with experimental investigation carried out with n-butanol-diesel blends on a heavy duty DI diesel engine. Influence of n-butanol and its blends with diesel was analyzed with the help of engine having facility of multiple injection of a fuel. Emission and performance characteristics were analyzed at a constant speed and load of engine. NO_x emission was fixed at 2.0 g/kWh with Exhaust gas recirculation rates were and blends of B0, B05, B10 and B15 are used.

The experimental outcomes confirm that emission characteristics like CO and soot can be drastically improved with supplement of n-butanol without affecting it BSFC at a NO_x

emission. The similar outcome in engine characteristics of was noticed with pilot and post injection for n-butanol blends as well with neat diesel. Soot is reduced with early pilot injection but reverse trend is observed with CO emission but soot and CO can be lowered with post injection. Further reduction in soot is observed with higher amount of blends injected at advanced timings and lowest soot emission can be achieved with triple-injection approach for higher amount of n-butanol fraction used in this analysis.

Park et al.(2011), The experimental analysis of engine in multiple-injection modes exhibits that engine exhaust emissions characteristics in terms of HC, CO and soot are lower with short interval between pilot and main of injection but reverse trend is noticed for NO_x emissions. From the results at 30° BTDC fuel spray combustion reaction was still not activated and resulting in a sudden increase of soot, CO, HC emissions. Contrary to this, the large number of particles decreased drastically for multiple-injection modes when compared relatively with conventional single-injection combustion in a CI engine.

2.6 Summary of the Literature Review

Some of the most notable features from the literature are as follows:

- Fast depleting fossil fuel reserves have alarmed to find alternate renewable fuels which could effectively replace conventional and be available at lesser cost.
- Diesel fuelled engines have been proven to be an efficient solution to the hunt for alternate fuels.
- Biodiesel is produced from renewable sources. They act as a very good oxygenates and have sufficient heat content to replace diesel effectively.
- Biodiesel usage can be considered in standard diesel engines with minimum or no engine/fuel system modifications and biodiesel has comparable energy density, cetane number also very low sulfur content.
- Biodiesel virtually decreases all regulated emissions and urgency need is to develop advanced technologies and concepts in this area to optimize the various engine parameters that will result in cleaner, more efficient diesel engines

- Biodiesel can be blended in any proportion with mineral diesel to create a biodiesel blend or can be used in its pure form and use of biodiesel in conventional diesel engines results in decrease of particulate matter (PM) and carbon monoxide (CO), but slight increase of oxides of nitrogen (NO_x).
- Proper optimization of injection pressure, injection timing and EGR ratio for each blend ratio will yield better efficiency of combustion.
- LTC has a significant role in the efforts on the mitigation of increased NO_x and other emissions among numerous technologies in conventional compression ignition engines and a very few works related to usage of EGR in CRDI engine for different injection strategies
- From the perspective of this particular objective, the experimental study of combined effects of Jatropha curcas biodiesel blends by applying different EGR rates on a variable speed CRDI engine for combustion, performance and exhaust emissions in comparison with diesel fuel operation is investigated and reported here.

2.7 Research Gap

The following gaps were found from literature survey

- The analysis of performance, emission and combustion characteristics Jatropha biodiesel blends for various engine loads under low tempearature combustion (LTC) with single injection mode.
- Effect of high pressure fuel injection for various injection timing, EGR ratio, Loads on performance, emission and combustion characteristics of CRDI engine in single and multiple injection mode.
- Effect of pilot injection timings, pilot fuel quantity under low tempearature combustion (LTC) with Jatropha curcas blends for performance, emission and combustion characteristics of CRDI engine.

CHAPTER 3

OBJECTIVES OF THE INVESTIGATION

3.1 Motivation for present investigations

From the point of view of long term energy security, it is necessary to develop new alternative fuels for diesel engines with properties comparable to petroleum-based fuels. Biodiesel has attracted increasing attention within the alternative fuels due to its fuel properties in reducing emission. Fuel Injection pressure and timing are the most important operating parameter that plays a very pivotal role in engine performance and emission compared to other parameters and present engine operating parameters are standardized for fossil diesel only, for any other fuel optimization of the operating parameters have to be carried out with the view of specific fuel properties.

LTC has a significant role in the efforts on the mitigation of increased NOx and other emissions among numerous technologies in conventional compression ignition engines and a very few works related to usage of EGR in CRDI engine for different injection strategies are reported. From the perspective of this particular objective, the experimental study of combined effects of Jatropha curcas biodiesel blends by applying different EGR rates on a variable speed CRDI engine for combustion, performance and exhaust emissions in comparison with diesel fuel operation is investigated and reported here.

3.2 Objectives of the Investigation

The objectives of the experimental investigation are as listed below:

1. To study the performance emission and combustion characteristics of Jatropha curcas biodiesel blended fuel in a CRDI engine
2. To investigate the effect of low temperature combustion on a CRDI engine using biodiesel as a fuel

3. To study the experimental investigation of the effect of high pressure common rail fuel injection on diesel engine for various fuel injection timings.
4. Optimization of EGR rate and biodiesel blend ratio for best performance with low emissions.
5. To study the effect of multiple injections using Jatropha curcas blends in a CRDI engine.

3.3 Scope of investigation

The main aim of the present experimental investigation is to analyze the performance, combustion and emission characteristics of twin CRDI engine by combined effect with simultaneous application of EGR for various Jatropha curcas biodiesel blends with suitable injection timing and fuel injection pressure has been carried out with single injection mode. Engine operating parameters such as IT, FIP, EGR rate and blend ratio are optimizes and further investigation continued with multiple injection strategy for optimized condition. In multiple injection, effects of pilot injection timings and pilot quantity for biodiesel blends under conventional as well as low temperature combustion mode has been studied. The CRDI engine is provided with open ECU for operating engine at high fuel injection pressure with wide range of fuel injection timings.

Experimental results obtained for the baseline diesel fuel were compared with results acquired from various engine operating parameters like high pressure fuel injection and injection timing for all tested loads. Comparative analysis of performance emission and combustion characteristics of engine is done to obtain optimized condition for better performance with reduced lower emissions for all tested loads. The exhaust emissions were measured in real time with exhaust gas analyzer and samples being taken as raw sample i.e., without any exhaust after-treatment devices in between the sampling point and the engine exhaust manifold.

CHAPTER 4

EXPERIMENTAL SET UP AND RESEARCH METHODOLOGY

The aim of this chapter is to describe the components of the research engine test facility, plan of work and methodology applied to achieve the objectives framed under present research work. The discussion also includesengine modifications, experimental parameters, measurement techniques and instrumentation for the course of the research to fulfil the present study of objectives framed.

4.1 Experimental setup

The heart of experimental test setup is twin cylinder four stroke CRDI diesel engine connected to eddy current type dynamometer for loading. The engine has high pressure common rail injection system, with this, it is possible to control the fuel injection quantity and injection timing, including the multiple-injection strategies under high pressure conditions. It is provided with necessary instruments for combustion pressure andcrank-angle measurements.These signals are interfaced to computer through engine indicator for pressure vs crank angle diagrams. Provision is also made for measuring the airflow, fuel flow,temperatures and load. The setup has stand-alone panel box consisting of air box, fuel tank, manometer, fuel measuring unit, transmitters for air and fuel flow measurements, process indicator, load indicator and engine indicator as shown in Figure 4.1. Two independent cooling systems were used to dissipate heat generated by the engine and dynamometer, Rota meters are provided for thepurpose ofmeasurement and control of cooling water flow.The test facility enables study of engine performance for brake power, BMEP, brake thermal efficiency, specific fuel consumption etc., Labview based Engine Performance Analysis software package "Enginesoft" is provided for online performance evaluation.Engine would be run periodically at a standard test condition to ensure consistency of the data and check for errors before the requiredtesting was carried out. Detailed specifications of the engine are tabulated in Table 4.1

The exhaust emission composition was studied from AVL 437S smoke meter and AVL DI Gas 444 gas analyzer for the analysis of exhaust gas which include HC, CO, CO_2 and NO_x. Engine would be run periodically at a standard test condition to ensure consistency of the data and check for errors before the required testing was carried out.

Table 4.1 Engine specifications

Parameter	Details
Engine make and Model	Mahindra, Maxximo
Type	Twin Cylinder, four stroke
Stroke x bore	83 x 84 mm
Power Rating	18.4Kw at 3600 rpm
Compression ratio	18.5
Injection	CRDI with ECU
Speed	3600 rpm max, variable
Torque	55 NM at 2500rpm
Aspiration	Natural aspiration
Cooling system	Water cooled
Dynamometer	eddy current, water cooled, with loading unit
Open ECU	NIRA make

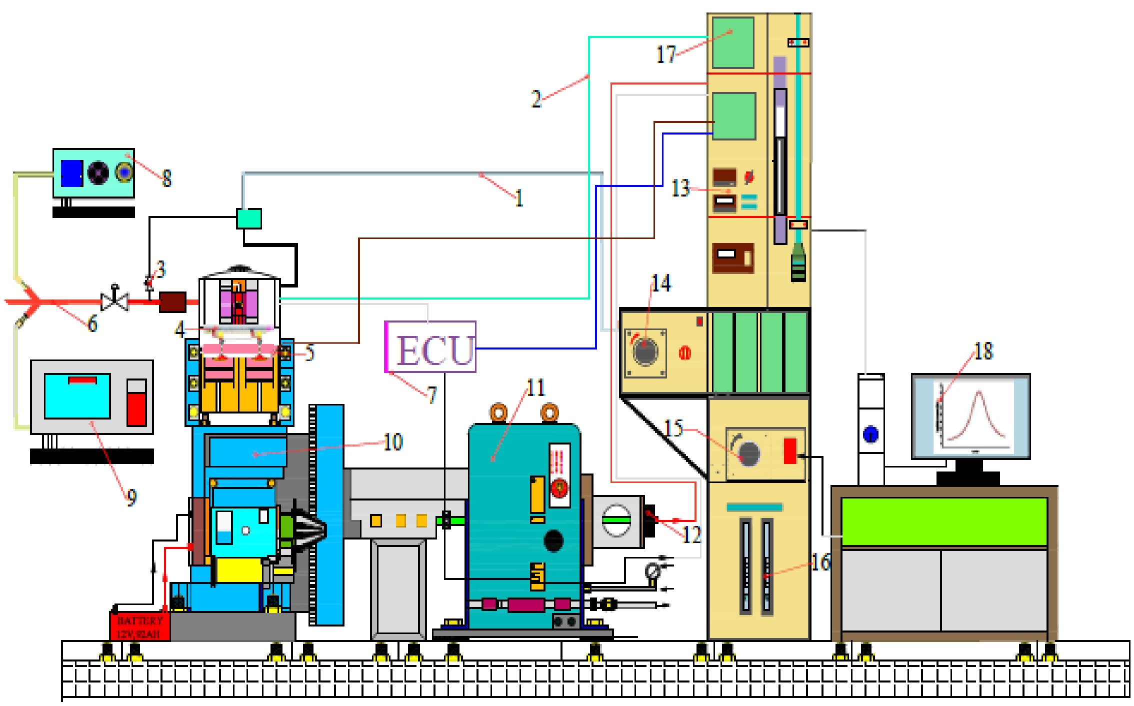


Fig. 4.1 Twin cylinder CRDI engine experimental test rig

(b)

1.Airline 2.Fuel line 3.EGR set up 4.Common Rail 5.Pressure transducer 6.Exhaust line 7.ECU 8.Gas analyzer 9.Smoke meter 10.Engine 11.Dynamometer 12.Encoder 13.Speed and load display unit 14.Throttle control unit 15.Load control unit 16.Rota meters 17. Fuel tank 18. Computer display

Fig. 4.2 (a) and (b), photographic view of experimental facility

4.2 Measurement of parameters and instrumentation

4.2.1 Air inflow measurement

Air inflow measurement is done by the air box method, it is M S fabricated with suitable volume of around 500 times swept volume having orifice meter (diameter 35mm). Air box fitted with suitable manometer isinstalled in the control panel. Pressure difference in the two limbs of the U-tube manometer, one end fixed across the inlet and other at outlet of air box gives flow rate. Air box is also used to dampen out vibrations caused by the cyclic nature of engine.

4.2.2 Fuel supply and measurement system

Fuel measurement is done using electronic engine management which measures fuel consumption online and displays on computer screen, for this purpose fuel tank of capacity 15 liters with glass tube (burette) fuel metering column is fitted and mounted on the control panel. Fuel is drawn from tank through Yokogawa make differential pressure transmitter by low pressure fuel pump located in the fuel supply line assembly then passed to the fuel filter. Impurities and any rust particles are screened out using fuel filter with the help of cartridges containing a filter paper. From filter, fuel enters high pressure pump and becomes pressurized here and enters common tube or rail. Here common rail acts as an accumulator and maintains constant pressure with the help of electronic engine management and distribute the fuel to the injectors at a constant pressure. Here high - speed solenoid valves, regulated by the electronic control unit (ECU) modifies injection pressure and timing precisely as needed and control the amount of fuel injected for each cylinder as a function of the cylinder's actual need based on data obtained from sensors on the cam and crankshafts. Excess over flow fuel is sent back in the fuel return line to low pressure fuel pump and temperature of fuel in overflow return line is higher caused due to high pressure in common rail

Provision is also made to measure fuel consumption offline manually by using a standard burette of 50cc mounted on control panel and fixed to fuel tank. The time taken to for 20cc fuel consumption was recorded accurately using stop watch. The necessary precautions were taken while observing the fuel levels between fuel burette calibrated marks while starting and stopping of the stop watch.

4.2.3 Speed measurement

Engine used for research is a variable speed and speed is sensed and indicated by an inductive pickup sensor in conjunction with a digital rpm indicator, which is a part of the eddy-current dynamometer controlling unit. The dynamometer shaft rotating close to inductive pickup, rotary encoder sends voltage pulse whose frequency is converted to rpm and displayed by digital indicator in the control panel, which is calibrated to indicate the speed directly in number of revolution per minute.Crank angle sensor has a resolution 1 Deg and speed 5500 RPM with TDC pulse.The engine control unit (ECU) is connected to a crank position sensor which also detects the speed of the engine.

4.2.4 Load measurement

Load measurements done using SAJ make water cooled eddy current dynamometer. It consists of a stator on which a number of electromagnets are fitted and a rotor disc driven by the output shaft of the engine. The coil is wound in circumference direction to create to excite the magnetic pole. When a current passes through exciting coil, a magnetic flux is formed around the coil through stators and a rotor. The rotation of rotor produces density difference, then eddy-current goes to stator. The electromagnetic force applies in opposite of the rotational direction by the product of this eddy-current and Vector of magnetic flux, thus loading the engine. These eddy currents are dissipated in producing heat so that this type of dynamometer needs cooling arrangement. Regulating the current in electromagnets controls the load. A moment arm measures the torque with the help of a strain gauge type load cell mounted beneath the dynamometer arm. The analog load cell signal is then converted to the digital signal and it is displayed on the display provided on dynamometer control panel. The dynamometer is loaded with the help of control unit fixed in the control panel. Heat generated by the dynamometer while in running condition is dissipated using water cooling system. Flow of water is measured and controlled using

Eureka make rotameter and flow rate was fixed at 175 litre/hour for cooling purpose. The specification of the dynamometer is given in appendix I.

4.2.5 Cylinder pressure and crank angle measurement

Pressure versus crank angle history was recorded using water cooled piezo-electric pressure transducer and crank angle sensor.

The crank angle was measured with resolution of 1 degree CA intervals by a Kubeler-Germany make crank angle encoder mounted directly on the engine crank shaft. One more additional crank angle signal with fixed resolution is used in engine test bench for the engine management. Precautionary step has taken to precisely center the encoder in order to prevent vibration.TDC location is determined by setting TDC indicator provided at the engine, the crank angle sensor body is slowly rotated till the TDC indicator lamp glows fixed in control panel, at this position it need to clamp the flange screws to fix the crank angle sensor.

A PCB Piezotronics make piezo-electric pressure transducer is used for recording the cylinder pressure for number of consecutive cycles for combustion variability studies. The sensor was mounted in a drilled threaded hole on the engine head and sensor was flash with the cylinder head surface. This type of transducer contains a quartz crystal. One end of the crystal is exposed through diaphragm to the cylinder pressure. As the cylinder pressure increases, the crystal is compressed and generates an electric charge which is proportional to the pressure. Cylinder pressure is sensed by pressure transducer and converted to electrical charge signals and amplified signal is then transmitted to digital computer operating on windows system loaded with LABVIEW based software which is capable of data logging with maximum of 100 consecutive combustion cycles.

The present experimental work with CRDI diesel engine, the signal of cylinder pressure was acquired for every 1 degree CA and the acquisition process covered 100 completed cycles, the average value of these 100 cycles being used as pressure data for calculation of the other combustion parameters. The pressure signal and crank angle encoder signal was fed into a NI based data acquisition card (NI USB-6210, 16bit) linked to the computer. Data acquisition card collects data at the rate of 250 kS/s. In cylinder pressure versus crank angle data was stored in the computer and used to calculate rate of heat release and analyse the combustion characteristics. The software “Enginesoft 9” in the computer draws pressure- crank angle and pressure volume diagram. One hundred consecutive combustion cycles of pressure data were collected and averaged to eliminate cycle-to-cycle variation in each test, then averaged data was analyzed to calculate HRR, mass burn fractions (MBF) and the specification details of pressure transducer are given in appendix II.

4.2.6 Exhaust Gas Recirculation (EGR) system and measurement

Exhaust gas recirculation (EGR) is an effective overall process to control engine emissions with no after treatment process, here fraction of exhaust gases are cooled by using heat exchanger as shown in Figure 4.1. Heat exchanger is directly connected in the exhaust line and then this cooled exhaust gas is sent to intake manifold where it mixes with fresh air and enters the engine cylinder. The rate of exhaust gas being re-circulated is calculated on the concentrations of carbon dioxide (CO_2)in intake and exhaust gas and also based on the volume fraction of air using following formula

$$EGR\% = \frac{(CO_2\%)_{\text{int}}}{(CO_2\%)_{exh}} x100$$

where $(CO_2)_{int}$, $(CO_2)_{ext}$are CO_2 concentration at intake and exhaust of the engine.

Another method is based on the CO_2concentration for determination of EGR rate i.e. (aka)

$$EGR\ rate = \left\{\frac{mass\ of\ air\ without\ EGR - mass\ of\ air\ with\ EGR}{mass\ of\ air\ without\ EGR}\right\}$$

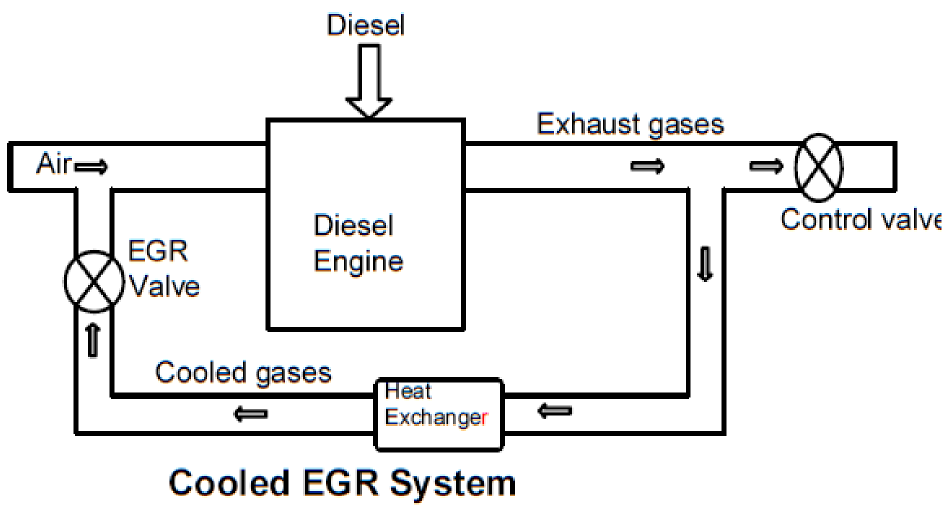


Fig. 4.3 schematic layout of cooled EGR system

4.2.7 Temperature measurement

Digital temperature indicator having six channels fixed on the control panel is used in conjunction with K-type thermocouples which measures the time averaged temperatures at various points including EGT temperature.

4.2.8 Data acquisition system

A National Instruments (NI) based hardware system USB-6210, 16-bit, 250kS/s was used for data acquisitionwith the aid of Labview based “Enginesoft” software from the engine sensors for engine performance and combustion analysis. It consisted of a single processor, three A/D boards, and acontrol area network (CAN) board.

4.2.9 Exhaust Emission measurement

An engine exhausts gas emission analysis is done using AVL 437S smoke meter which has measurement range from 0 to 100% opacity of soot concentration in the exhaust and AVL DI Gas 444 gas analyzer for the analysis of exhaust gas which includes species like carbon monoxide (CO), carbon dioxide (CO_2), oxygen (O_2) in percentage of volume and unburnt hydrocarbons (HC) and oxides of nitrogen (NO_x) in ppm. The five gas analyzerand smoke meter were calibrated as per manufacture manual and necessary precautions are taken to see the proper working of them by regular checkup of all types of filters and cleanliness of probe was maintained. Leakage test and zero adjustments are done regularly. The specifications of the analyzer are given as Appendix III.

4.3 Biodiesel preparation, blending and Testing

Biodiesel is prepared using Jatropha curcas seeds by using trasestrification process as per standards. Blending of biodiesel has been done by volume basis with diesel in which 10% referred as JB10, similarly 20% and 30% blend referred as JB20 and JB30.

4.3.1 Analysis of fuel properties

Prior to conducting the experimental studies, a careful fuel analysis needs to be carried out. The fuel properties can influence the fuel droplet size, the size distribution, spray characteristics, fuel evaporation, temperature, ignition delayand emissions.Hence these various fuel properties such as kinematic viscosity, flash point, fire point, lower calorific value were determined using standard instruments and are listed in Table 4.2.

4.3.2 Tests for flash and fire point

The flash point and fire point tests were conducted to find the ease with which the fuel tested would burn at a given temperature. Tests were conducted in the Cleveland's (Open cup) apparatus as per ASTM D92 standards.

4.3.3 Tests for kinematic viscosity and density

The viscosity is a measure of the internal fluid friction of fuel to flow, which tends to oppose any dynamic change in the fluid motion. Viscosity is the major reason why Jatropha curcas oil is transesterified to methyl ester (or biodiesel). Tests are performed as per ASTM standard D445 using the Redwood viscometer at standard 40^0C.

4.3.4 Calorific Value

The heating value or calorific value of a fuel is the magnitude of the heat of reaction at constant pressure or at constant volume for the complete combustion of unit mass of fuel. A bomb calorimeter was used to find the calorific value for various blends as per ASTM D4809 standards.

Table 4.2 Results different fuel properties investigated

Diesel and fuel blends	Calorific value	Flash point	Fire point	Kinematic viscosity	Density
	(MJ/kg)	($^{\circ}$C)	($^{\circ}$C)	cSt	(kg/m^3)
Diesel	42.50	74	81	3.15	830
Jatrophacurcas biodiesel	39.20	183	195	4.35	875
JB10	41.77	79	86	3.45	836
JB20	41.14	84	89	3.60	842
JB30	40.40	88	92	3.85	850

4.4 Experimental research methodology

4.4.1 Scheme of engine experimental studies

The engine experimental study involves three distinct stages. Initially experiments have been conducted with modified engine setup for pure petro-diesel at various loads for 2500 rpm engine speed at steady state condition. Further experiments have been continued with various biodiesel blends and EGR ratiosfor particular fixed fuel injection pressure and injection timing and lastly fuel injection pressure, injection timings were varied independently to evaluate the engines performance, combustion and emission characteristics. Effect of multiple injectionstrategy is applied for optimized condition from single injection mode.

Jatropha curcas biodiesel produced from transestification is used for blending with diesel, blends of this biodieselJB10, JB20 and JB30 are used as fuel in a high pressure common rain direct injection engine along with Exhaust Gas Recirculation (EGR) ratios of 10%, 20% and 30%. Fuel injection pressure (FIP) 600bar, 800bar and 1000 bar was used along with 10^0CA BTDC, 14^0CA BTDC and 18^0CA BTDC injection timing. Load is applied from zero to 100%, in the step of 25% interval.

Before starting the engine for experimentation it is need to be ensured that steady and continuouscooling water circulation for eddy current dynamometer, pressure-crank angle measurement sensor, engine and calorimeter cooler.Then engine test rig need to be started and allowed to run for minimum of 4-5 minutes at no load conditions to ensure, the engine was fully warmed and ready for conducting experiments. Then set the engine for required speed and load and allow engine to attain steady state condition by running it for 3 to 5 minutes. The all required readingsrelated for performance, combustion and emission were noted for particular specified test condition. Three readings were noted for each test condition and average of three readings is taken for analysis purpose to avoid errors during experimentation.

4.4.2 Plan and procedure

Research methodology of the current research workexplained above is planned is shown in flow chart in Figure 4.4

To analyse the engine performance emission and combustion characteristics for single as well multiple injection strategy following engine parameters are noted for various fuel compositions with varying EGR ratio and load conditions.

- Brake power, Brake thermal efficiency
- Brake specific fuel consumption, Brake specific energy consumption
- Exhaust Gas Temperature, Mean Gas Temperature
- Net heat release rate
- Pressure vs Crank angle, Peak pressure
- NOx , CO, HC and smoke emissions

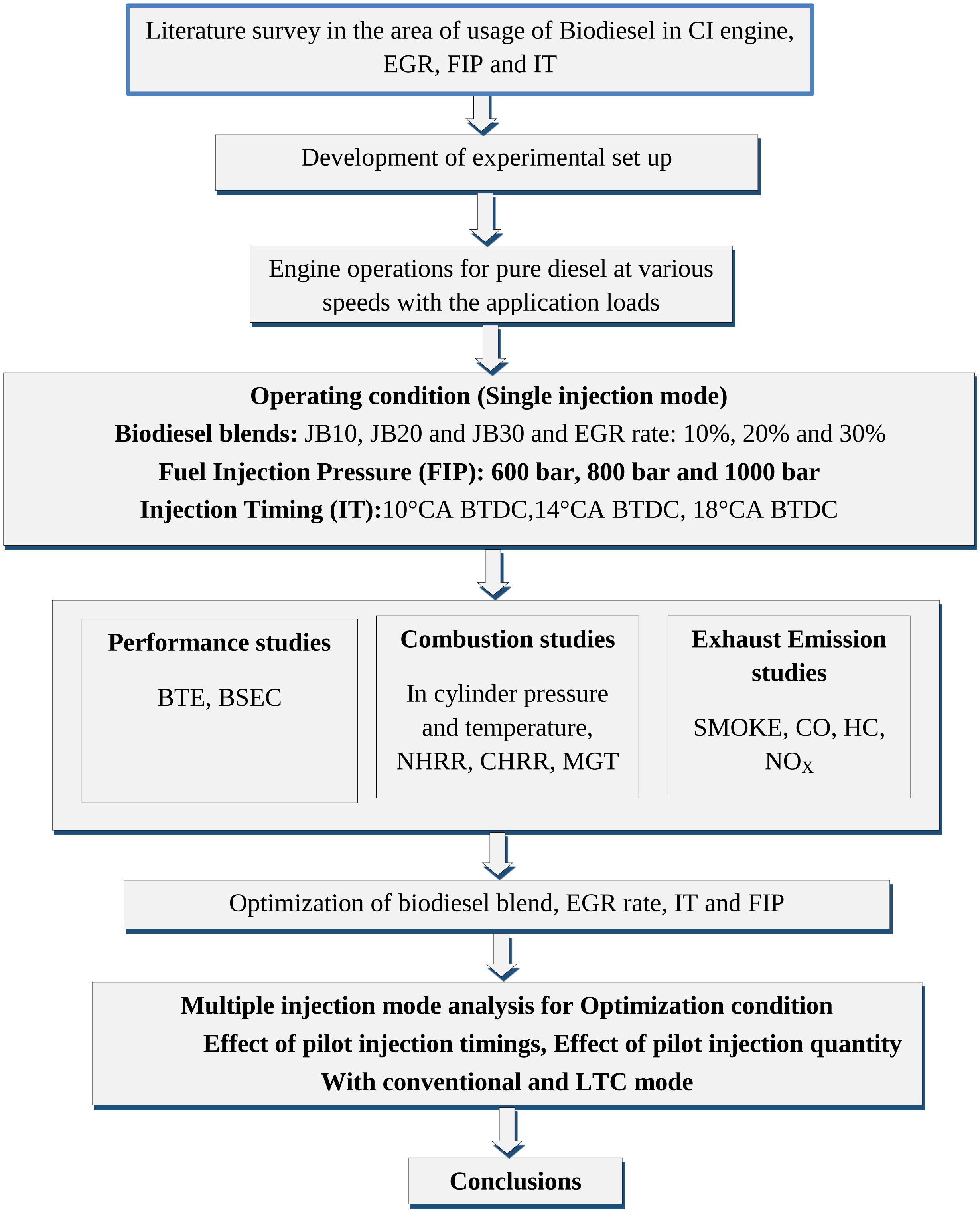


Fig. 4.4 Flow chart of experimental plan

4.4.3 Plan and procedure

For better accuracy and precision in the measurement of different engine parameter NIRA i7r open ECU (engine control unit) is deployed in test rig, NIRA i7r is an effective engine control unit developed by NIRA control AB intended to optimize performance advanced diesel engines. NIRA i7r comes with an application tool to control loading of software program, turning of engine data, monitoring etc. Hence combining all the features it offers the operator full access to control the engine functions. Featuring support for advance fuel injection strategies with multiple injections per stroke NIRA permits the engine tuner to attain extreme performance while retaining necessary characteristics such as quick engine response time and precise control of torque output. NIRA is also designed to handle sensors actuators related to EURO 5 and tier 4 regulation, including EGR and particulate filters. Online monitoring and recording of results obtained from the experiments conducted with different engine operating parameters are capturedand shown in Figures 4.5 to 4.10.

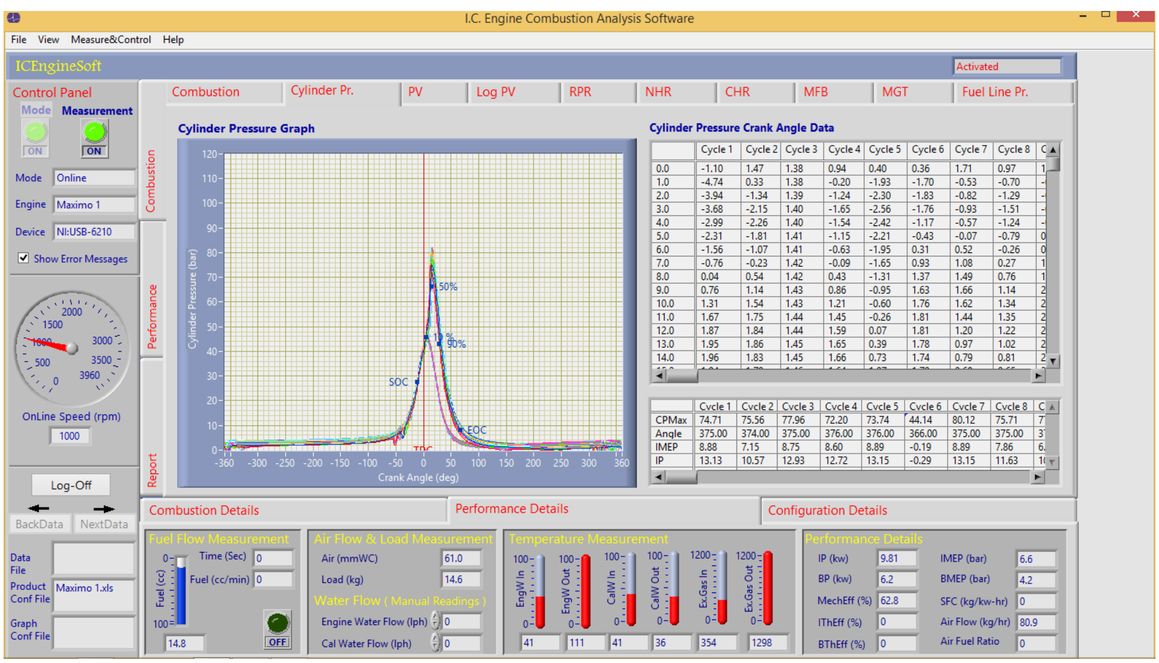


Fig. 4.5 online recording of pressure vs crank angle

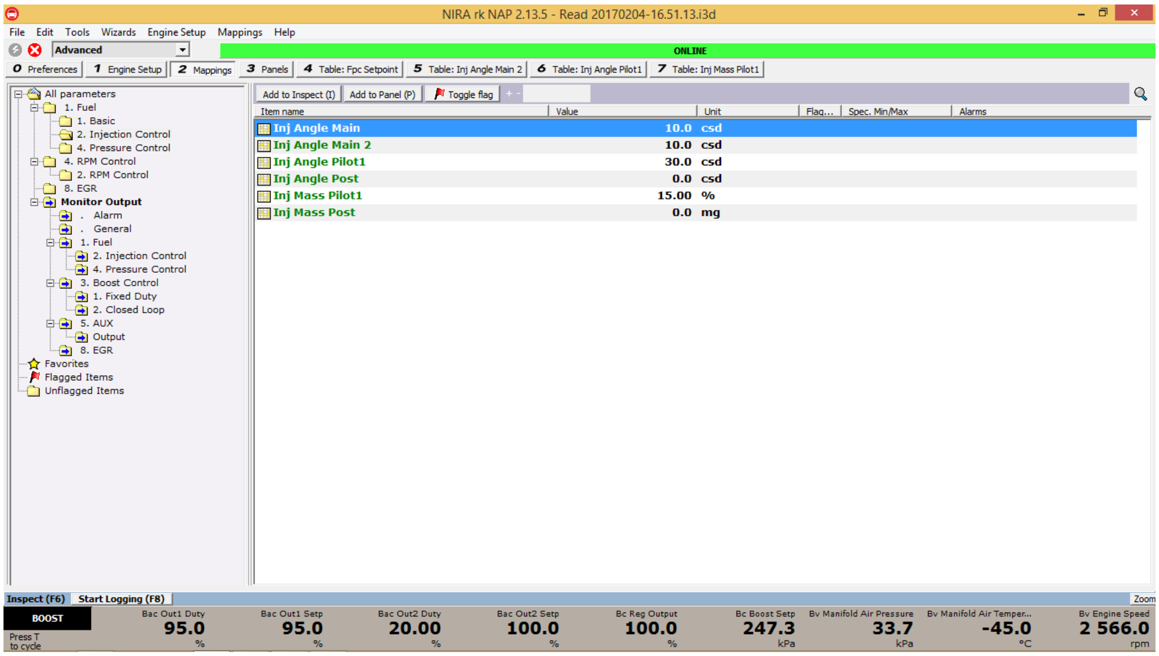


Fig. 4.6 online input of various engine fuel injection control setting

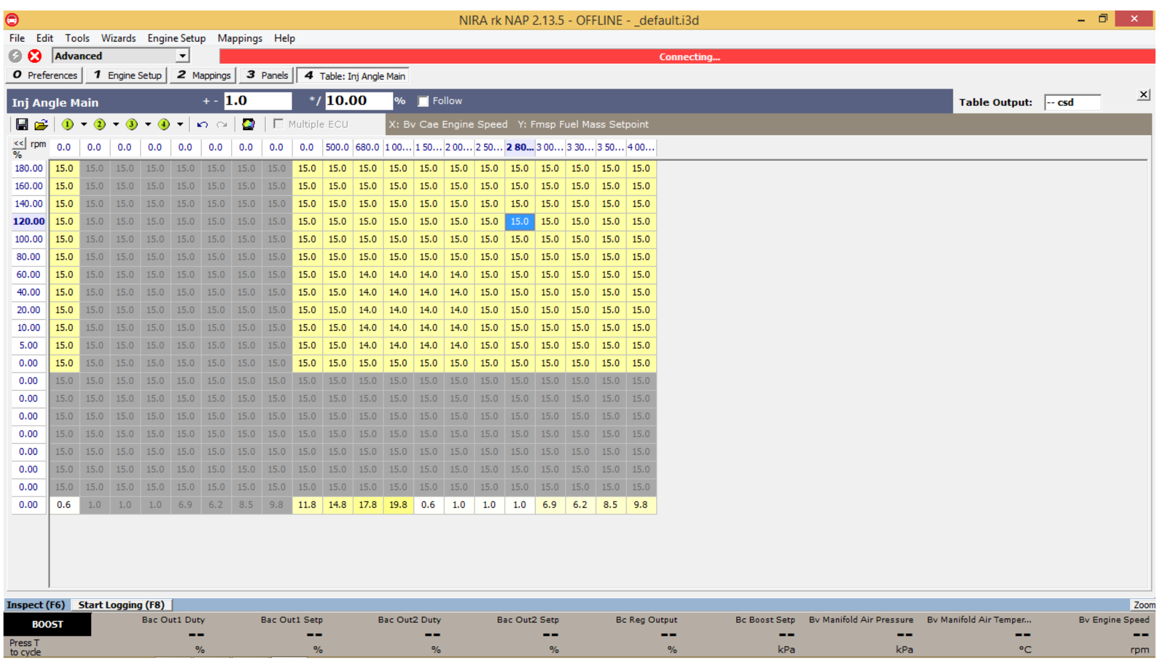


Fig. 4.7 online setting of fuel injection timing

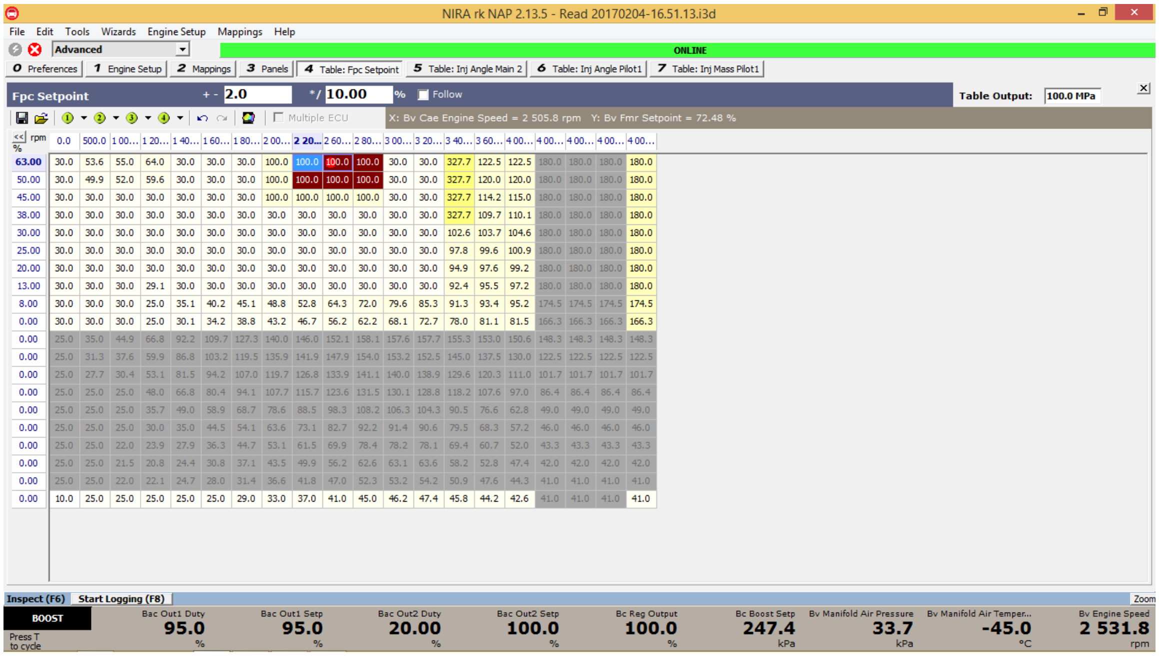


Fig. 4.8 online setting of fuel injection pressure

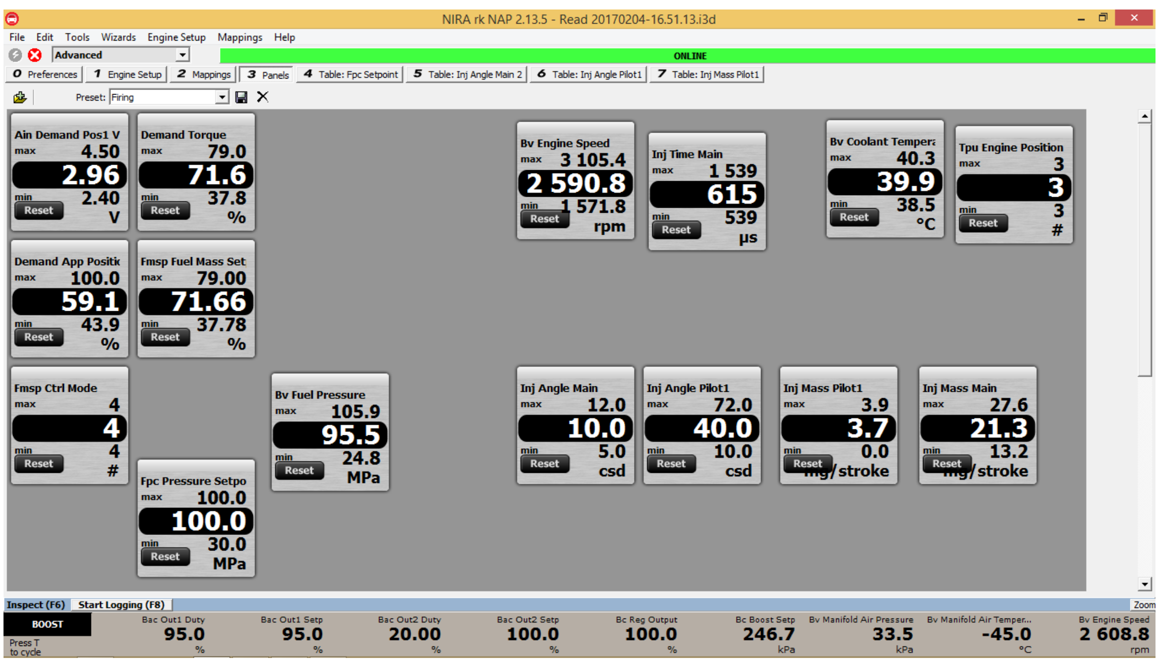


Fig. 4.9 online display of control panel showing all engine operating parameter

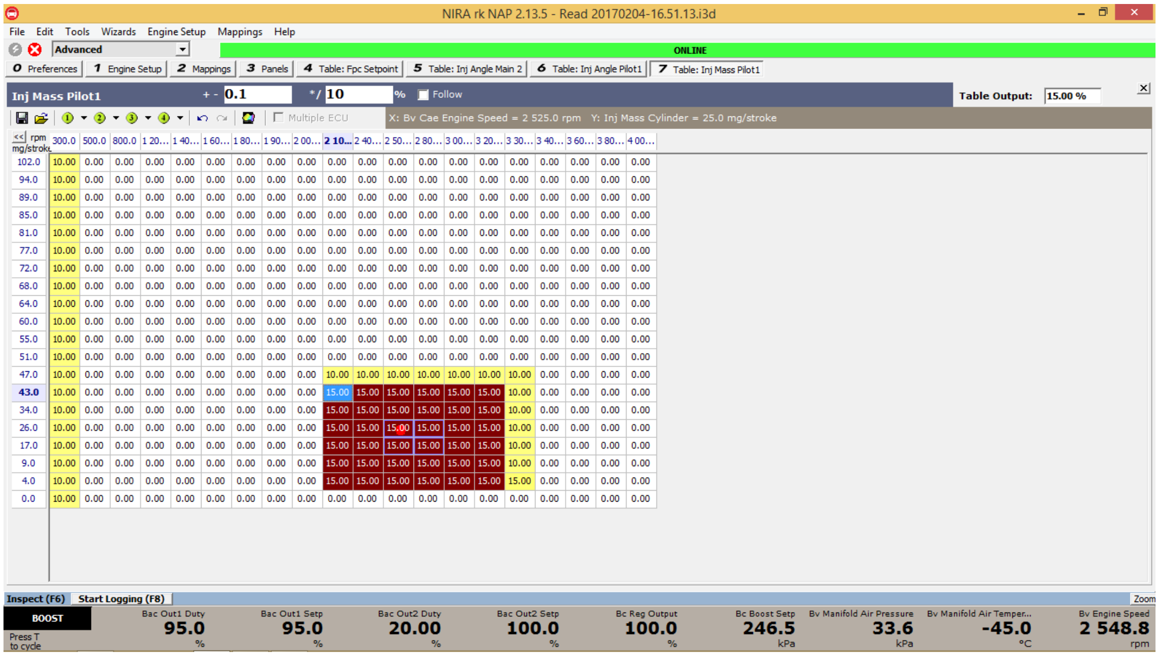


Fig. 4.10 online setting of fuel injection quantity

4.5 CALIBRATION OF INSTRUMENTS

All instruments are calibrated prior to their use in the tests. The dynamometer, exhaust gas analyser and pressure sensor are factory calibrated by the suppliers. The temperature sensors are calibrated with reference to standard thermometers. Rotameters are calibrated by manual measurement of the compressed air flow through a known time. While conducting the experiments due care is taken to check the repeatability of readings. At each test point the engine is allowed to reach steady state operating condition by allowing it to run for sufficient time. Average of at least three readings at each test point is taken to minimize the experimental error.

4.6 COMBUSTION ANALYSIS

In-cylinder combustion pressure data is very useful information, cylinder pressure is the constantly changing pressure inside the cylinder for all 4 strokes. There are certain characteristics which could be used to quantify the combustion behaviour of the fuels inside the engine, like peak (maximum) cylinder pressure near TDC, or peak cranking compression pressure (pressure at TDC without combustion), or average cylinder pressure (IMEP, indicated mean effective pressure). They are all just measures of cylinder pressure at certain times (or averaged over certain times) in the 4 strokes. Additionally, engineers, researchers also perform estimated rate of heat release, mass-burned fraction, and the charge temperature for design and optimization of engine

4.6.1 **Rate of heat release analysis**

To analyze the combustion phenomena critically, it is necessary to separate the effects of volume change, heat transfer and mass loss. This can be done by calculating heat release rate, which has the benefit of identifying the combustion indices like ignition delay, combustion duration, heat release rate and its crank angle position etc. Heat release analysis estimates the amount of heat required to be added to the cylinder, in order to produce the observed pressure variations. In the present work, an effort is made to determine a single zone heat release rate and combustion temperature in a C I engine, using experimentally obtained average pressure-crank angle data. A Lab view based Engine software is used to calculate standard heat release rate equations, correlations and constants which are discussed below. Heat release rate is computed for 100 consecutive combustion cycles at every test point.

This numerical analysis is based on the thermodynamic first law during the closed part of the engine cycle. The combustion parameters are obtained from the standard heat release rate equations during a cycle, i.e. the energy conservation equation. The first law of thermodynamics is applied by considering cylinder contents as a single open system, whose thermodynamic state and properties are being uniform throughout the cylinder and are specified by:

$$\frac{dQ}{dt} - p\frac{dV}{dt} + \sum_i \dot{m}_i h_i = \frac{dU}{dt} \qquad (4.1)$$

Here, Q represents heat transferred in Joules, p denotes pressure in Pascal, V is the volume m^3, m_i is the mass of fuel injected in kg/s, h_i is the enthalpy in J/kg, and U represents internal energy in J. Assuming that the internal energy and enthalpy are sensible terms (at room temperature) and only the mass transferred from the system is the injected fuel. Rewritingthe above equation as:

$$\frac{dQ}{dt} = p\frac{dV}{dt} + \frac{dU}{dt} \tag{4.2}$$

Complexity of heat transfer across the system boundary arises only at the end of combustion with the rise in temperatures. Assuming the contents of the cylinder as an ideal gas, equation 2 can be written as:

$$\frac{dQ}{dt} = p\frac{dV}{dt} + mC_v\frac{dT}{dt} \tag{4.3}$$

Here, $C_v$is the specific heat at constant volume.

Differentiation of the perfect gas law with R assumed constant, provides a means of eliminating the temperature term which is generally unavailable in pressure analysis to give

$$\frac{dQ_{Net}}{dt} = \left(1 + \frac{c_v}{R}\right)p\frac{dV}{dt} + \left(\frac{c_v}{R}\right)V\frac{dp}{dt} \tag{4.4}$$

Substituting the specific heat ratio γ, provides the final equation used in the analysis with the result being equally valid when substituting the independent variable θ or crank angle, for time t, then the net heat release combustion model of Krieger and Borman is obtained [15]

$$\frac{dQ_{Net}}{d\theta} = \frac{\gamma}{\gamma - 1}p\frac{dV}{d\theta} + \frac{1}{\gamma - 1}V\frac{dp}{d\theta} \tag{4.5}$$

Where, γ is the ratio of specific heats, Q_{Net} is the net heat release rate in Joules per degree, p is the in-cylinder pressure in Pascal, and V is the in-cylinder volume in m^3.

Calculating the cylinder volume (V) from crank angle for a slider-crank mechanism as follows

$$V = V_c + \frac{\pi B^2}{4}\left(l + R - R\cos\theta - \sqrt{\left(l^2 - R^2\sin^2\theta\right)}\right) \tag{4.6}$$

Where, V_C is clearance volume at TDC in cubic meter, B is bore in meter, l is connecting rod length in meter, R is crank throw in meter (= stroke/2), θ is crank angle measured from the beginning of the induction stroke in radians.Specific heat at constant pressure is given as:

$$C_p = \frac{R}{1 - \frac{1}{\gamma}} \tag{4.7}$$

A temperature dependent equation for specific heat ratio γ obtained from experimental data is used [16].

$$\gamma = 1.338 - 6x10^{-10}T + 1x10^{-8}T^2 \tag{4.8}$$

where T is the mean charge temperature found from equation of state, pV=mRT. Since the molecular weights of the products and the reactants are similar, the mass m and gas constant R can be assumed as constants. If all thermodynamic states (p_{ref}, T_{ref}, V_{ref}) are known or evaluated at a given reference condition such as inlet valve close (IVC), then the mean charge temperature T is calculated as

$$T = pv\frac{T_{ref}}{p_{ref}V_{ref}} \tag{4.9}$$

The cylinder volume at IVC is computed using the cylinder volume given in the above equation for $\theta_{IVC}$and is therefore considered to be known. The two other states at IVC (P_{IVC}, T_{IVC}) are considered unknown and have to be estimated.

4.6.2 In cylinder temperature

The in cylinder combustion temperature of an engine is an important parameter to depict the thermodynamic state of combustion process. It also affects the performance and life of an IC engine. The cylinder temperature can vary from low to high. But the high temperatures are most destructive and produce adverse effect on engine characteristics.

The measurement of cylinder temperature with respect to crank angle was done by using equation 4.10.

$$T = \frac{P*V*\ M_g}{m.R} \qquad (4.10)$$

Where P= in cylinder pressure V= volume M_g= molar mass of gas

m= total mass of charge R= universal gas constant

4.2.3 Rate of pressure rise

The rate of pressure rise with respect to crank angle was measured by using equation 4.11.

$$\text{Rate of pressure rise} = \frac{P_n - P_{n-1}}{\theta_n - \theta_{n-1}} \qquad (4.11)$$

Where Pn = cylinder pressure at n^{th} crank angle P_{n-1} = cylinder pressure at $n\text{-}1^{th}$ crank angle

θ_n = crank angle at n^{th} crank angle θ_{n-1} = crank angle at $n\text{-}1^{th}$ crank angle

4.3 ERROR AND UNCERTAINTY ANALYSIS

Error is associated with Experimental investigation and the theoretical calculations of all the performance parameters associated with the subject. Errors and uncertainty in the experimental work can arise from instrument selection, condition, calibration, environment, observation, reading and test planning. Thus uncertainty analysis is needed to support the experiment precision.

The uncertainty in any measured parameter is estimated based on Gaussian distribution method with confidence limit of ±2σ (95.45% of measure data lie within the limits of ±2σ of mean). Thus uncertainty of any measured parameter is given by:

$$w_i = \frac{2\sigma_i}{\bar{x}} 100 \qquad (4.12)$$

Experiments are coordinated to obtain the mean ($\bar{x}$) and standard deviation (σ_i) of any measured parameter (x_i) for a number of readings. This is done for speed, load, time for a specified amount of air and fuel flow etc. For the analysis, 20 sets of readings are taken at the same operating condition. The uncertainty values for speed, torque, air flow rate, fuel flow rate, exhaust gas temperature and emissions of NO_x, HC, CO, CO2 are calculated using equation (4.12).

A method of estimating uncertainty in experimental results has been presented by Kline and McClintock (1953). The method is based on careful specifications of the uncertainties in the various primary experimental measurements.

Suppose a set of measurements is made and the uncertainty in each measurement may be expressed with the same odds. These measurements are then used to calculate some desired results of the experiments. The uncertainty in the calculated result can be estimated on the basis of the uncertainties in the primary measurements.

If an estimated quantity R depends on 'n' independent measured parameters x_1,x_2, x_3, x_4…, x_n. Then R is given by

$$R=R(x_1,x_2, x_3,x_4\ldots x_n) \qquad (4.13)$$

Let w_R be the uncertainty in the result and w_1, w_2…w_n be the uncertainties in the independent measured parameters. R is the computed result function of the independent measured parameters x_1, x_2, x_3….x_n as per the relation $x_1 \pm w_1$, $x_2 \pm w_2$,…. $x_n \pm w_n$). If the uncertainties in the independent variables are all given with the same odds, then the uncertainty in the result having these odds is given as (Adnan et al. 2012):

$$w_R = \left(\left[\frac{\partial R}{\partial x_1} w_1 \right]^2 + \left[\frac{\partial R}{\partial x_2} w_2 \right]^2 + \ \ldots\ldots + \left[\frac{\partial R}{\partial x_n} w_n \right]^2 \right)^{1/2} \qquad (4.14)$$

Using the equation (4.30) for a given operating condition, the uncertainties in the computed quantities such as mass flow rates of air and fuel, brake power, brake thermal efficiency are estimated.

The estimated uncertainty values at a typical operating condition are given in the next section.

4.3.1 Sample Calculations

One set of sample calculation is performed and is as mentioned below for twin cylinder CRDI engine operation at 2500 rpm

Speed=2500 rpm

Fuel consumption rate =2.094 kg/hr

Torque=34 N-m

Calorific value = 42500 kJ/kg

(i) Brake Power (BP)

In the experimentation, brake power is obtained from:

Brake Power:, $BP = \frac{2\pi NT}{60000} kW$ (4.15)

Where, N = Engine speed in rpm,

$$BP = \frac{2\pi x 2500 x 34}{60000} = 8.90 kW$$

(ii) Brake Specific Energy Consumption

Brake specific energy consumption,

$BSEC = \frac{m_f\, CV}{BP}$ MJ/kWh (4.16)

Where, m_f= Mass flow rate of fuel in kg/hr

BP = Brake power in Kw

$$BSEC = \frac{2.094 x 42500}{8.90}$$

= 10000 kJ/kWhr

(iii) Brake Thermal Efficiency

Brake Thermal Efficiency (%)

$$\text{B}TE = \frac{BP \times 3600 \times 100}{Cv \times m_f} \tag{4.17}$$

$$BTE = 8.9X3600x100\,\frac{8.9x3600x100}{42500x2.094}$$

BTE = 36%

(iv) Uncertainty analysis

BP=f(N, W)

$$BP = \frac{2\pi \times N \times T}{60000} = NT \times 1.05E^{-4}$$

$$\mathrm{w_{BP}} = \left(\left[\frac{\partial \mathrm{BP}}{\partial \mathrm{N}}\mathrm{w_N}\right]^2 + \left[\frac{\partial \mathrm{BP}}{\partial \mathrm{W}}\mathrm{w_W}\right]^2\right)^{1/2}$$

$$w_N = 0.11$$

$$w_T = 0.73$$

$$\frac{\partial \mathrm{BP}}{\partial \mathrm{N}} = 0.315$$

$$\frac{\partial \mathrm{BP}}{\partial \mathrm{W}} = 1.89\mathrm{E}^{-3}$$

$$\mathrm{w_{BP}} = ([0.315\mathrm{X}\ 0.11]^2 + [1.89\mathrm{E}^{-3}\ \mathrm{X}0.73]^2)^{1/2}$$

= 0.035 kW

= 0.6%

4.3.2 Uncertainty of various parameters

Table 4.3. Uncertainty of various parameters

. No.	**Parameter**	**Uncertainty (%)**
1	Speed	±0.15
2	Torque	±0.55
3	Air flow rate	±1.0
4	Fuel flow rate	±0.7
5	Exhaust gas temperature	±1.4
6	NO_X emission	±1.8
7	HC emission	±2.5
8	CO emission	±2.4
9	Brake power	±0.6
10	Brake thermal efficiency	±0.25

CHAPTER-5

RESULTS AND DISCUSSION

This chapter gives the experimental results of investigation on twin cylinder CRDI engine with mineral diesel, Jatropha curcas biodiesel blends for different EGR rates, Injection timing (IT), and Fuel injection pressure (FIP) by varying engine load with single injection mode. Jatropha curcas biodiesel blends JB10, JB20 and JB30 are used in a CRDI engine having facility of injecting fuel at high pressure of 600, 800 and 1000 bar FIP using common rain direct injection fuel system. Exhaust Gas Recirculation (EGR) ratios of 10%, 20% and 30% were also applied for different injection timings of 10°CA BTDC, 14°CA BTDC and 18°CA BTDC and Load is applied from zero to 100%, in the step of 25% interval at constant speed of 2500 rpm. Each experimental test are conducted to evaluate the performance parameters such as brake thermal efficiency (BTE), brake specific energy consumption (BSEC), emission characteristics like carbon monoxide (CO), oxides of nitrogen (NO_x), hydrocarbon (HC) and smoke opacity. Also combustion characteristics like in-cylinder pressure, mean gas temperature, net heat release rate from the engine were analysed. From single injection mode analysis biodiesel blend ratio, IT, FIP and EGR rates are optimized and experiments with multiple injection mode carried out for the optimized values to study effects of pilot injection timing, pilot injection quantity on combustion, performance and emission on CRDI engine with conventional and as well as low temperature combustion mode and are presented and discussed.

5.1 Effects of usage of biodiesel blends, EGR and high pressure fuel injection (FIP) and injection timing (IT) on performance, emission and combustion characteristics of engine with single injection mode.

5.1.1 Brake Thermal Efficiency (BTE)

Brake Thermal Efficiency (BTE) is the important parameter which plays important role to evaluate conversion of chemical energy of a fuel to mechanical output. The effect of fuel, fuel blends, fuel injection pressure, and injection timing on thermal efficiency is shown in Figure 5.1-5.2. The engine was operated at 600 fuel injection pressure with injection timing of 14° CA BTDC, the fuel blends are varied from 10 to 30% by volume of biodiesel. It is observed that, the brake thermal efficiency increases with increase in load when engine operated with different fuels/blends, this is due to steadily decrease in heat losses (Bhanapurmth 2004 and Heywood 2011).

It is also observed from the Figure 5.1(a), that the usage of JB10 and JB20 blends has resulted in improvement of thermal efficiency by 0.8% and 1.75% at 600 bar FIP, 14°CA BTDC. The improvement in BTE for lower blend concentration of JB10 and JB20 is may be due to the presence of oxygen molecules in the biodiesel which helps in improving the combustion in comparison with diesel operation (Agarwal A.K 2015 and Dinesh 2014). Further increasing in biodiesel concentration to 30% (JB30) in petrodiesel, the reduction in thermal efficiency is observed. The decrement value is found to be around 1% and this is due to lower heating and higher viscosity of the blend as listed in the Table 4.2.

The outcome of application of exhaust gas recirculation (EGR) at the rate 10%, 20% and 30% for various Jatropha curcas biodiesel blends is also indicated in Figure 5.1(b-d). The 10% EGR rate application has resulted in improvement of performance at all loads as compared against neat diesel fuel operation (Liu 2013 and Ong 2014). With the application of 20% EGR rate, the improvement in performance is observed at lower loads only but reduction in performance is noticed for higher load of 75% and 100% condition and are performance values are relatively lower than 10% EGR rate. This improvement with EGR application, is mainly due to re-burning of unburnt hydrocarbons during combustion which are entered into combustion chamber by mixing with fresh air (Agarwal.D 2011 and saleh 2009) and In addition to these, oxygen content of the biodiesel fuel compensates to some degree of lack of oxygen caused by EGR application and promotes the combustion rate. An improvement of about 0.7% with JB20 for 10% EGR rate is noticed at 600 bar FIP, 14°CA BTDC.

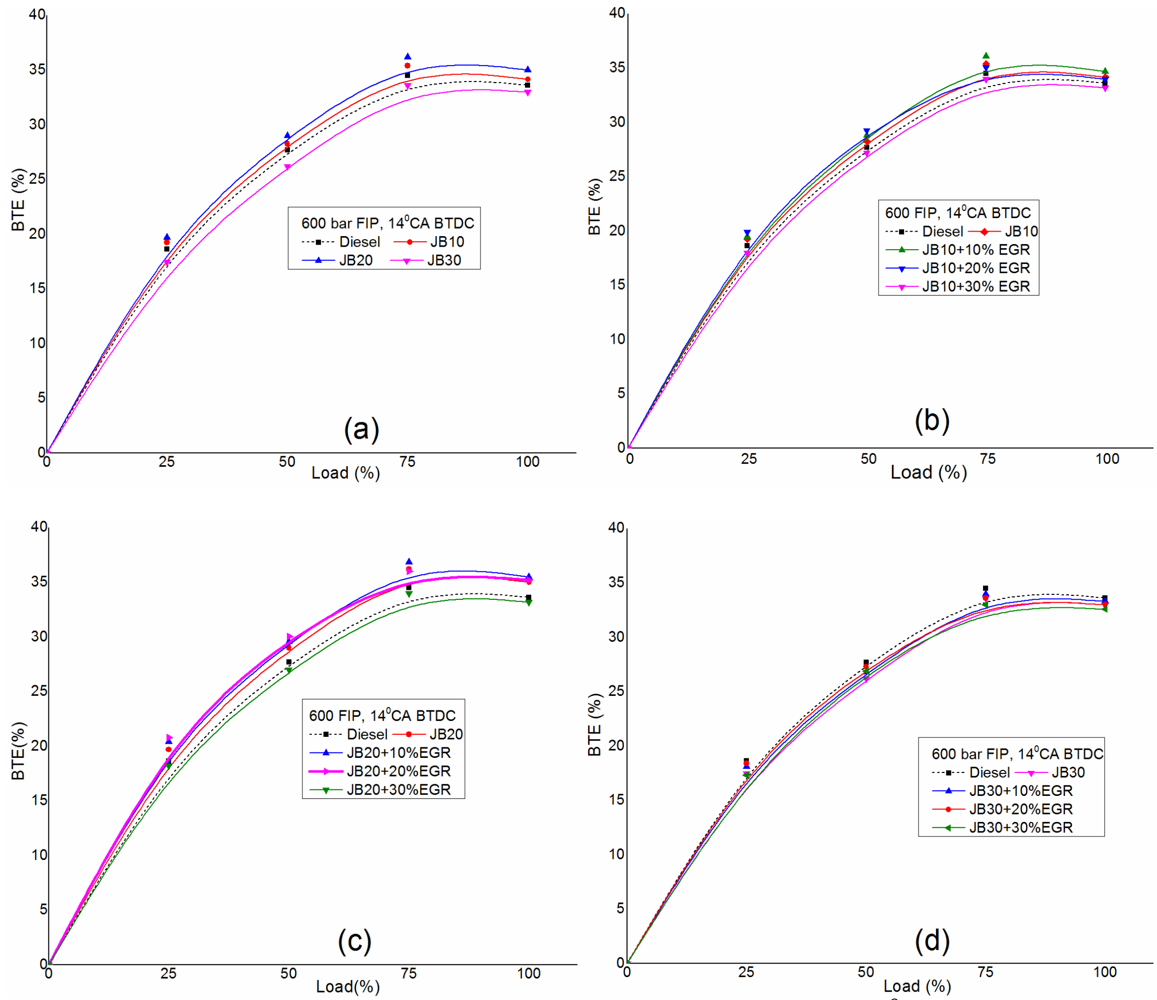


Fig. 5.1. Effect of fuel blends on BTE at 600 bar FIP and 14^0 CA BTDC

At 30% EGR rate, the reduction in efficiency is noticed and this reduction is due to the fact that the exhaust gases may contain higher amounts of CO_2, which limits the peak combustion temperature in the combustion chamber along with the oxygen availability as well decreased volumetric efficiency or breathing capacity of the engine (Agarwal.D 2011). Hence the remarkable re-burning of HC is not achieved. The reduction is found by around 2% at 600 bar FIP, 14^0CA BTDC.

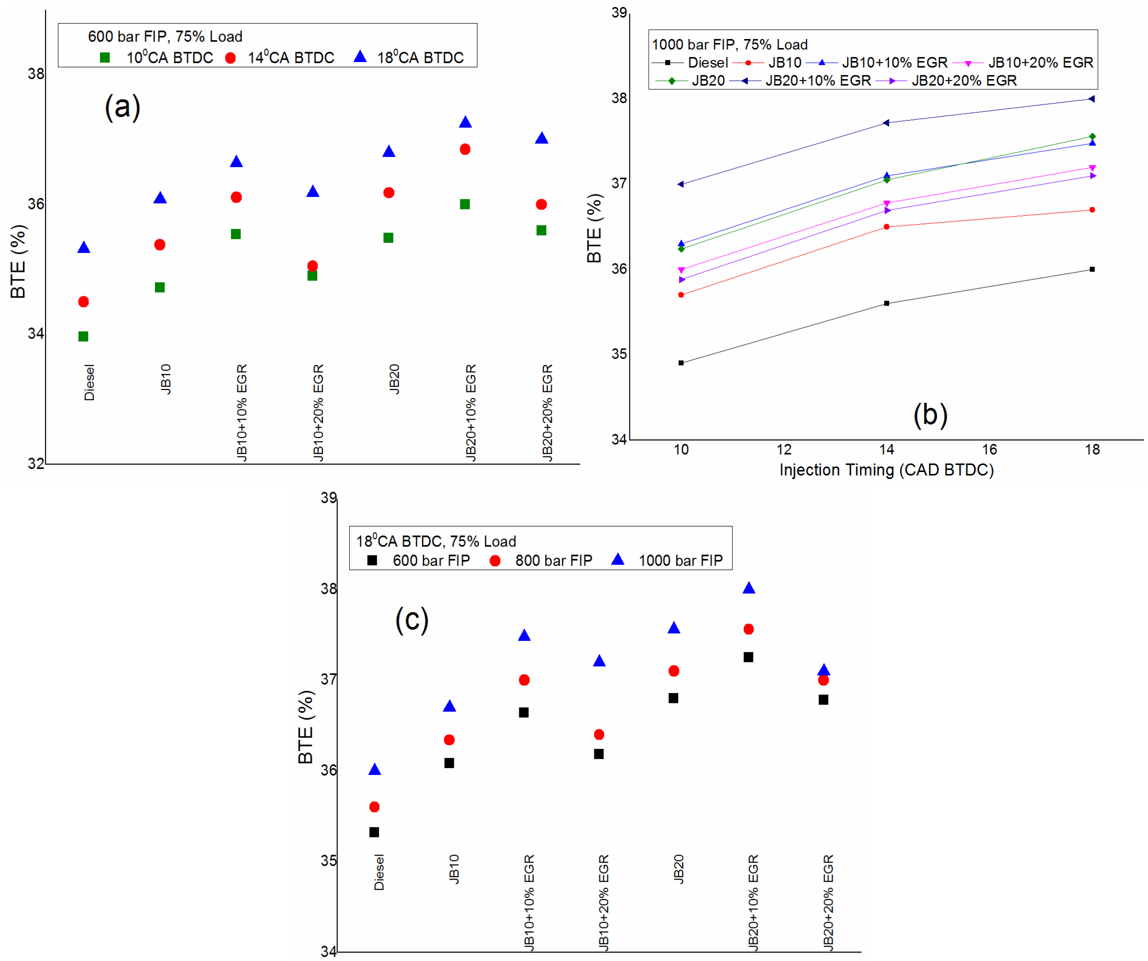


Fig. 5.2. Effect fuel blends on BTE at various injection timings and FIP

Further, the behavior of fuel blends/different EGR rate for various Injection timings (IT) and Fuel injection pressure (FIP) were studied. With regard to this two more fuel injection timings are chosen such that out of two, one injection timing is advanced and another is retarded from the 14°CA BTDC for performance, emission and combustion analysis. Also for any fixed injection timing, fuel injection pressure is increased from 600 bar to 800 bar and then to 1000 bar to study the effect of high pressure fuel injection on engine behavior with various operating conditions.

When injection timing is advanced by 4°CA BTDC from 14°CA BTDC i.e., 18°CA BTDC, the improved efficiency was observed for diesel as well as biodiesel blends is shown in Figure 5.2(a &b). This is because of fuel mixture operates at longer ignition delay (physical delay) hence advanced injection timing leads to better mixing, which results in better combustion leading to maximum BTE at advanced timing. For same fuel injection pressure of 600 bar, the Maximum of 1.1% improvement in BTE is noticed for JB20 blend with 10% EGR rate at 18°CA compared to 14°CA BTDC for 75% load condition (Gumus 2012).

Retarding fuel injection timing by 4° CA (from 14°CA BTDC) i.e., 10 °CA BTDC results in decrease of performance and this is because of since fuel injected nearer to TDC and has less time available for combustion of fuel hence incomplete combustion and reduced efficiency. At 10°CA BTDC, the BTE is reduced by 0.85% by compared to standard injection timing of 14°CA BTDC for 600 bar FIP (Gumus 2012).

Figure 5.2(c) shows effect of increase of fuel injection pressure (FIP) on the twin cylinder engine, the increase of FIP reduces the injection duration also higher FIP atomizes fuel spray into fine droplets which culminates into better and faster combustion due to better air entrainment at increased FIP and hence further improvement in the engine performance is observed. The maximum improvement in BTE is observed at 1000 bar FIP by 0.5% and by 0.3% at 800 bar as compared to 600 bar FIP at fixed IT of 18°CA BTDC for JB20 blend without EGR condition.

5.1.2 Brake Specific Energy Consumption (BSEC)

Brake specific energy consumption (BSEC) is the product of the BSFC and heating value of fuel and it is used as a tool for comparing the performance of fuels with different heating values. It measures how much energy is being consumed in one hour to develop a unit power output. With the increase in the load BSEC decreases due to better energy conversion efficiency for all the tested conditions.

The variations of BSEC with respect to change of load for Jatropha biodiesel blends JB10, JB20, JB30 with EGR application and diesel fuel is shown in the Fig.5.3(a-d). It is clear from the figure that the brake specific energy consumption is lower than the conventional diesel due to better combustion. Lowest BSEC is noticed for JB20 blend at 75% load condition and highest BSEC is noticed for JB30 at 25% load. The BSEC is lower by 4.6%, 2.5% for JB20, JB10 and increased by 2.7% for JB30 compared at 75% loading condition with 600 bar FIP and 14° CA injection timing.

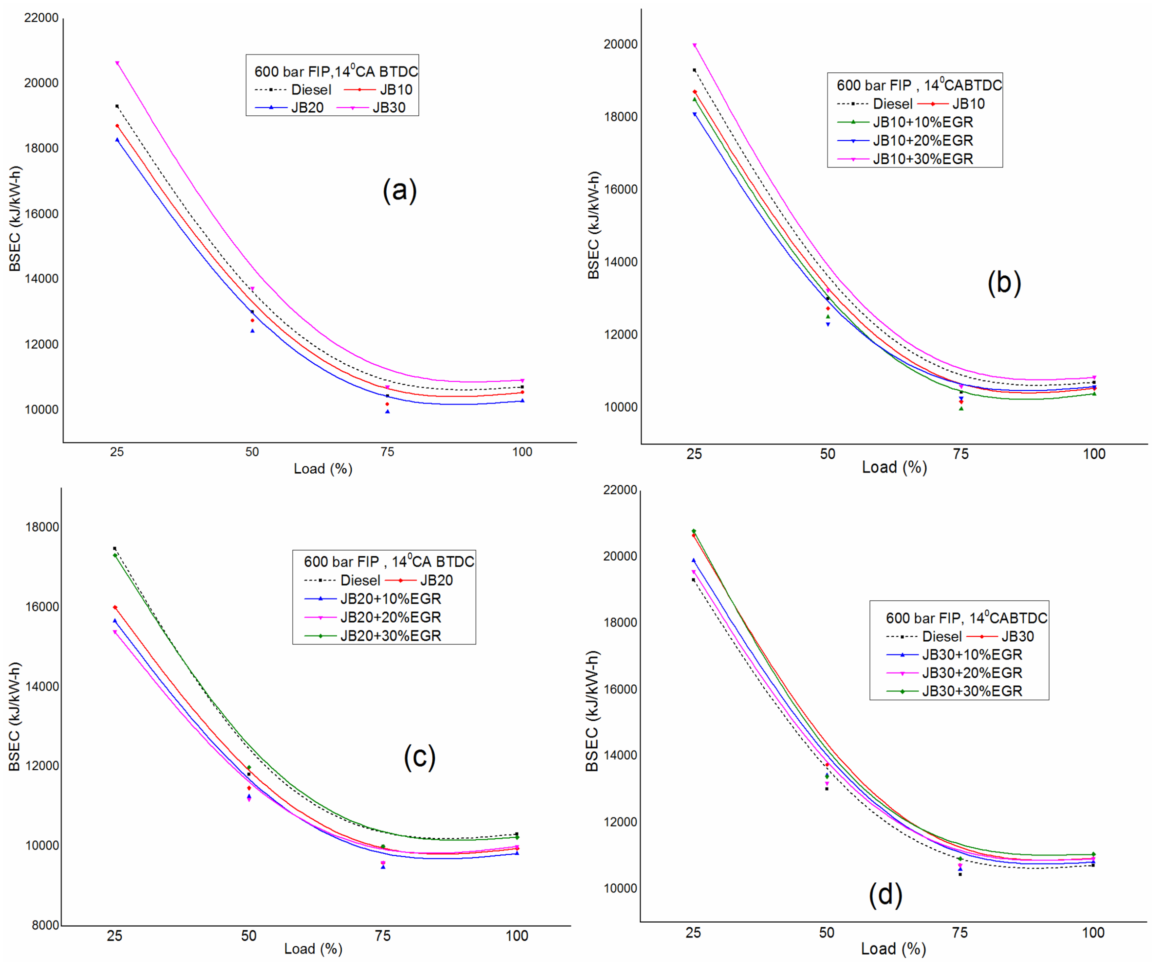


Fig. 5.3 Effect of fuel blends on BSEC at 600 bar FIP and 14^0 CA BTDC

With the application of EGR rate a slight decrease of BSEC value is noticed compared against pure JB blends, this is mainly due to improvement in combustion and reburning of unburnt hydrocarbons.

Large variations in brake specific energy consumption were noticed at advanced fuel injection timing and higher fuel injection pressure, as the injection timing is advanced to 18^0 CA BTDC by 4^0 CA from 14^0CA BTDC, the reduction in BSEC is observed for 18^0CA due to better mixing of fuel and air. Retarding the injection timing by 4^0CA BTDC, the increase in BSEC is noticed for 10^0CA BTDC with the same operating conditions as shown Figure 5.4(a-b), BSEC in respect of 18°CA injection timing is decreased by 1.7% compared to 14°CA and increased by around 2% with retarded timing to 10°CA. With respect to FIP, lowest BSEC is noticed at higher fuel injection pressure due to improved atomization of fuel at 1000 bar FIP as shown in Figure 5.4(c).

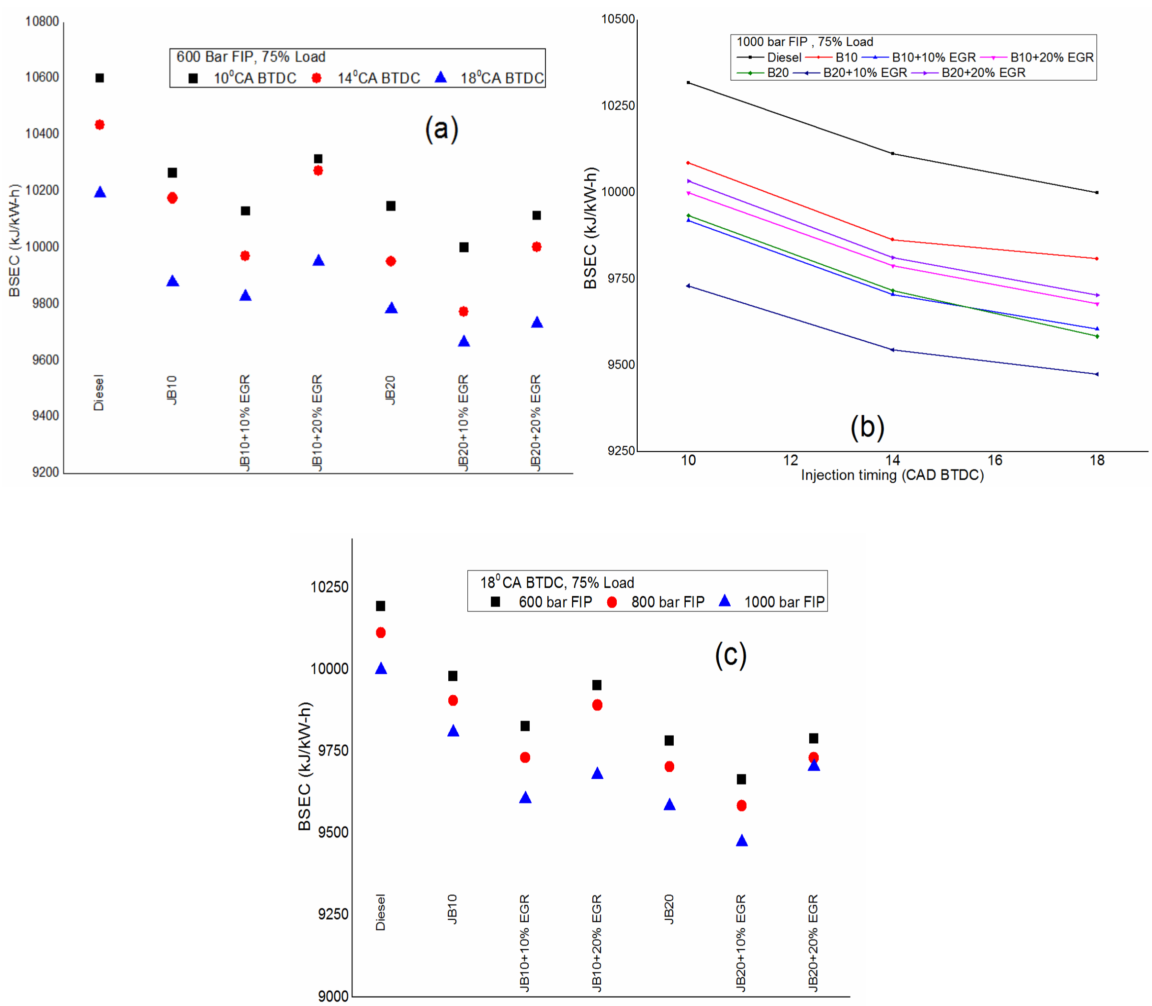


Fig. 5.4 Effect fuel blends on BSEC at various injection timings and FIP

5.1.3 Emissions

5.1.3.1 Nitrogen Oxides (NO_x) Emissions

Nitrogen and oxygen from air combined at very high combustion temperatures to form oxides of nitrogen. Once the tiny particles of fuel sprayed into the engine, the fuel particles and oxygen interacts at the boundary surface between the air and fuel and leads to start of combustion and consequently burning of fuel droplets. The localized temperature in the vicinity of the fuel particle spray surpasses the limit resulting in NO_x formation. In general high temperatures and availability of oxygen are two causes for

NO_x formation. Usually C I engines operate at high compression ratios with lean air-fuel mixture leading formation of high temperature and this favorable condition makes way for formation high-level NO_x emissions (Saleh 2009). Here higher NO_x in engine exhausts are due to higher oxygen molecule level in the biodiesel and elevated combustion temperature. The general trend is nitrogen oxides (NO_x) emission increases with usage of the biodiesel blend ratio i.e. the NO_x emission increases with increase of Jatropha curcas blends in the mineral diesel. Highest emissions of oxides of nitrogen are observed at JB20 blend at all tested conditions. For 600 bar FIP at 14^0 BTDC, the increase in NO_x emission is observed as 14%, 17.4%, 9.5% respectively for JB10, JB20 and JB30 at full condition compared with diesel fuel as shown in Figure 5.5(a).

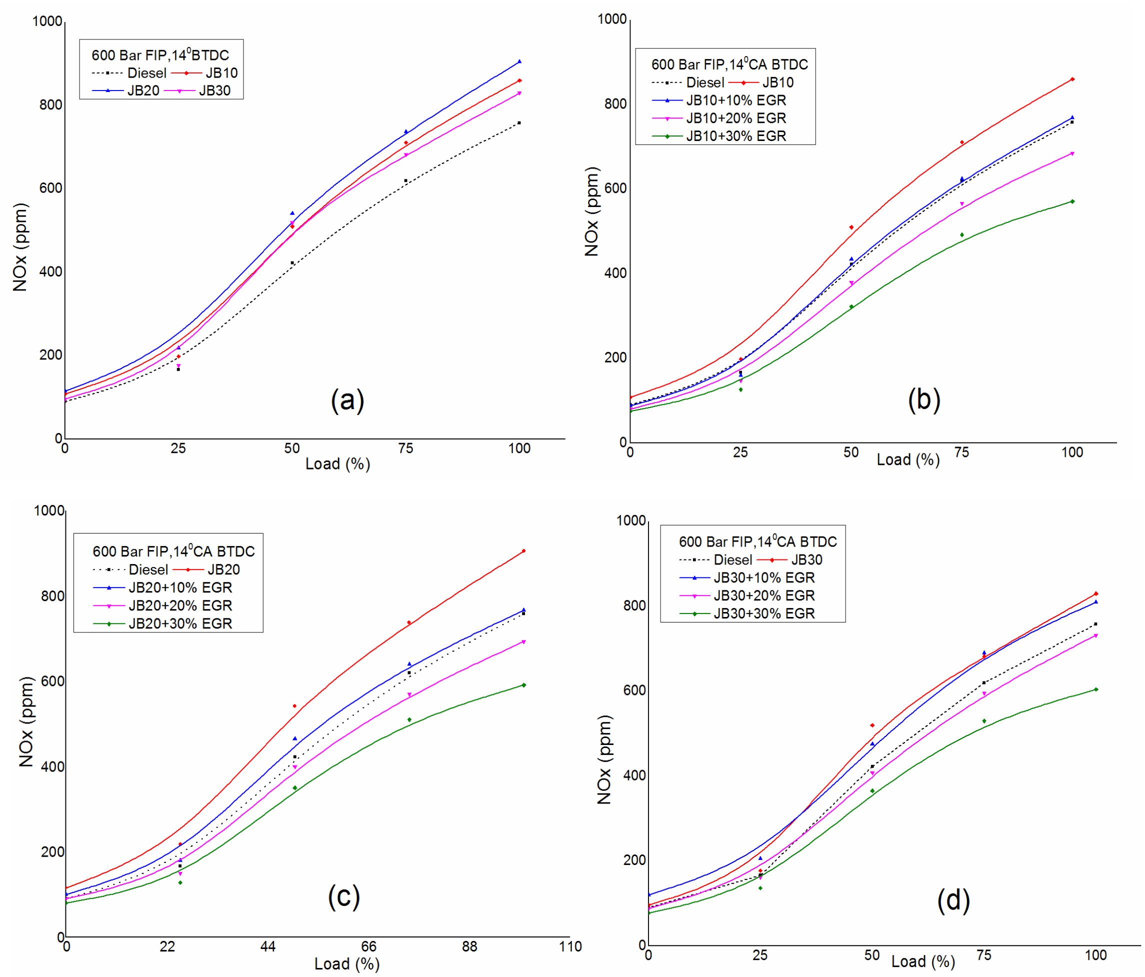


Fig. 5.5 Effect of fuel blends on NO_x at 600 FIP and 14^0 CA with different EGR rate

However, NO_x evolution decreases moderately with the application of EGR rate as shown in Figure 5.5 (c-d). This is owing to the fact that EGR dilutes oxygen concentration inside the cylinder, thereby limiting the flame temperature which reduces the NO_x evolution due to low temperature combustion environment. Reduction of NO_x when compared relatively with 10% and 20% EGR rate, maximum reduction has taken place at 30% EGR rate and at full load condition since more reduced volumetric efficiency at that condition, hence it can be concluded that NO_x level decreased and the reduction rate is proportional to EGR rate. For same speed and load condition, the decrement is around 14%, 22% and 34% for 10%, 20% and 30% EGR rate for JB20 blend ratio compared with standard injection timing of 14°CA BTDC at full load condition.

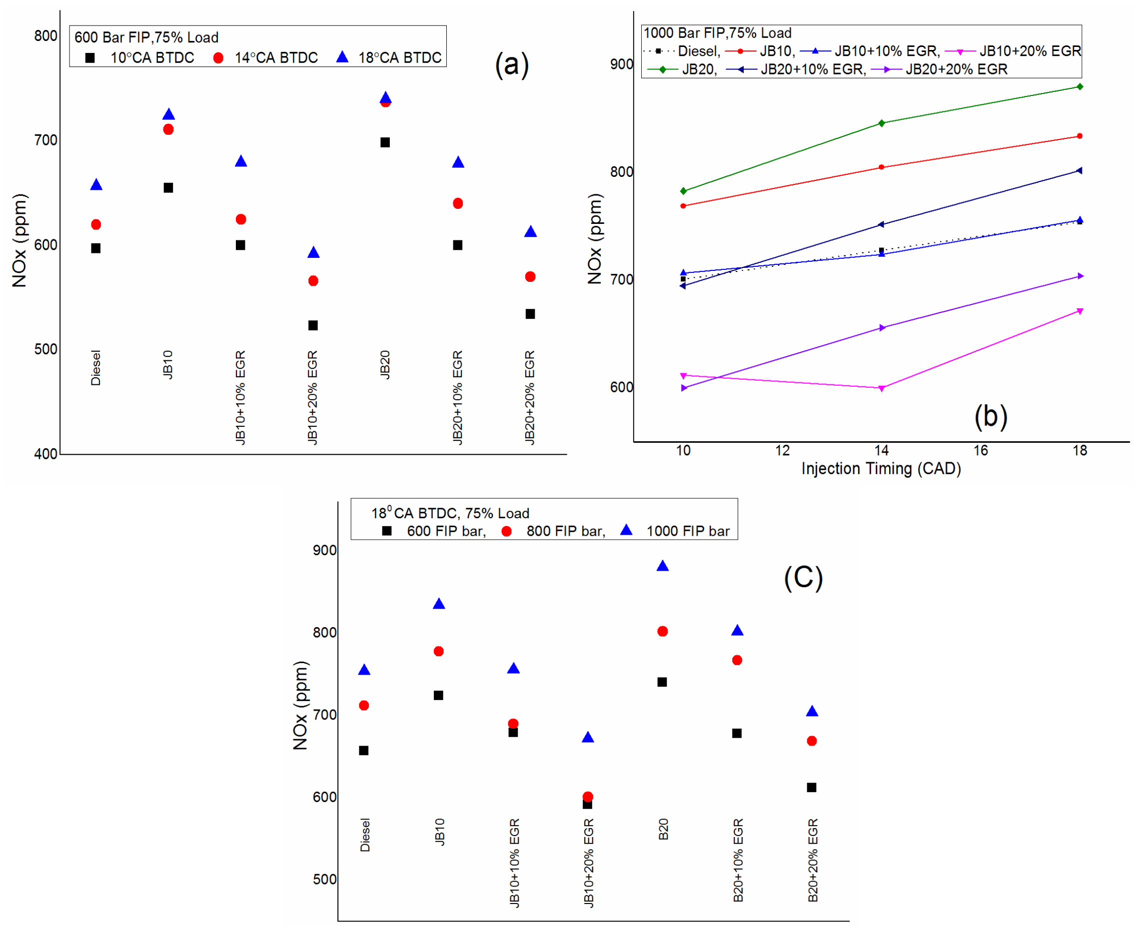


Fig. 5.6 Effect fuel blends on NO_x emission at various injection timings and FIP

The advancing of injection timing to 18°CA BTDC, NO_x emissions increase because more fuel burns before TDC resulting in higher peak cylinder pressures and peak temperatures. Also when the injection timing was retarded to 10°CA BTDC, this decreases the peak cylinder pressure because of more fuel burns after TDC. Lower peak cylinder pressures results in lower peak temperatures, as a consequence, the NO_x concentration starts to diminish. Compared to 10° CA BTDC injection timing, the increasing order is around 6% and 10% for 14°CA and 18°CA BTDC respectively for JB20 fuel blend at 75% load condition with 600 bar FIP and is shown in Figure 5.6 (a-b).

Further, Increase of fuel injection pressure increases oxides of nitrogen from engine exhausts due to higher in cylinder pressure and temperature because of better atomization of fuel spray as shown in Figure 5.6(c). Highest increase is observed by 19% at 1000 bar FIP and at advanced 18°CA injection timing for JB20 fuel compared to 600 bar FIP at the same injection timing and 75% load.

5.1.3.2 Smoke Opacity

Smoke formation is mainly due to incomplete combustion of fuel and it is also strongly affected by air-fuel ratio, mixture distribution and fuel composition. From the present experimental study, it is observed that with the usage of biodiesel blends in diesel engine, the smoke opacity emission trend is lower than that of neat diesel under all operating conditions.

It is decreased by 36% for JB20 biodiesel blend ratio at 18^0 CA BTDC at full load condition without EGR application. This is due to increase of oxygen concentration along with decrease in the carbon content in biodiesel (Hwang 2014 and Ozener 2014). Here oxygen molecules of the biodiesel help to promote stable and complete combustion by delivering oxygen to fuel rich zones thereby reducing locally over rich regions to limit primary smoke formation (Hwang 2014) and also an absence of aromatics in biofuel is added advantage for lower smoke. Smoke opacity level is reduced by 21%, 31% and 17% for JB10, JB20 and JB30 compared to neat diesel operation and is shown in Figure 5.7 (a).

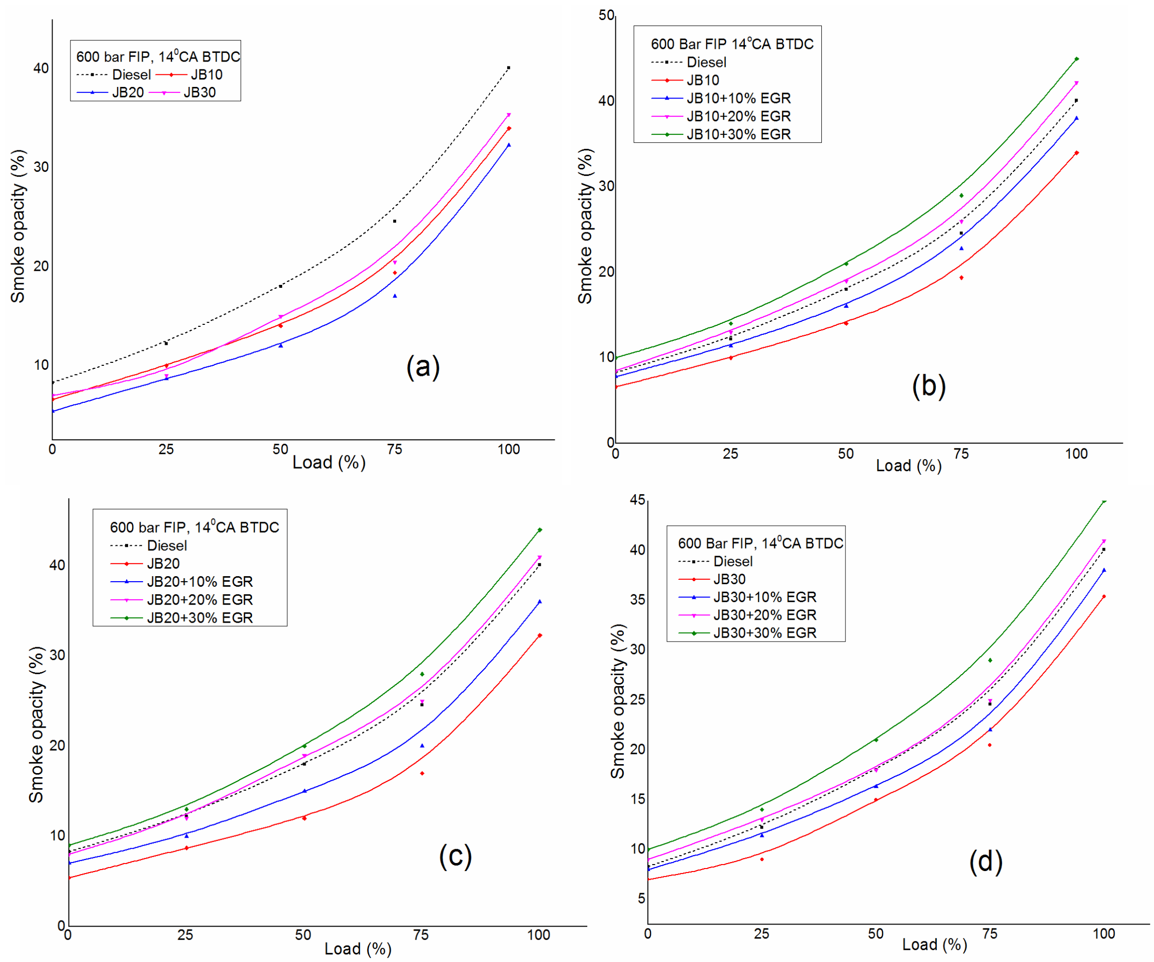


Fig. 5.7 Effect of fuel blends on smoke opacity at 600 FIP and 14^0 CA

With the application of EGR rates 10%, 20% and 30%, smoke opacity in the exhaust is slightly increased compared relatively with the biodiesel blend ratios. Further it is observed that, smoke opacity is lower at 10% of EGR compared to neat diesel operation at all load conditions. Engine performance worsened at 20%, 30% EGR rates above medium load condition and it is shown as in Figure 5.7(c-d). This is due to the fact that exhaust gas recirculation reduces the oxygen required for proper combustion of fuel inside the combustion chamber leading higher origination of smoke.

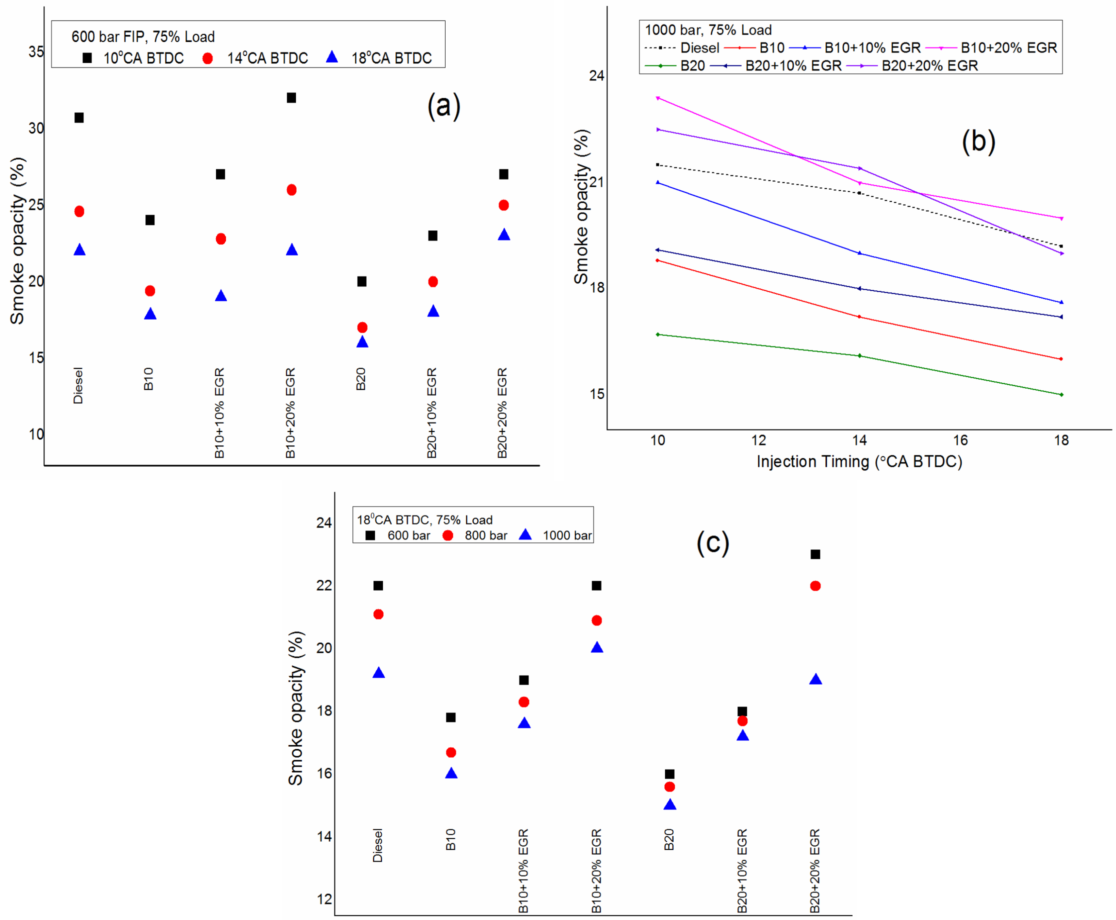


Fig. 5.8 Effect fuel blends on smoke opacity at various injection timings and FIP

Compared with standard injection timing of 14^{0}CA BTDC, when engine is operated at advanced injection timing, it observed that, the smoke opacity level is decreased due to better combustion. Retarding the injection timing decreases ignition delay and reduces the time available for proper combustion and hence more smoke observed at retarded injection timing 10^{0}CA BTDC compared to advanced injection timing of 18^{0}CA BTDC. Smoke opacity is reduced by 20% at 18^0 CA compared to 10^0 CA BTDC for JB20 fuel blend at 75% load condition as shown in Figure 5.8(a-b)

In regard to fuel injection pressure, it was revealed that as FIP increases smoke level steadily decreases. This is due to better atomization with increase in injection pressure. At 75% load condition, smoke level decreased by 12% with 1000 bar compared to 600 bar FIP for JB20 blend at 18°CA and shown in Figure 5.8(c).

5.1.3.3 Hydrocarbon (HC) emissions

Figure 5.9 (a), shows the variation of hydrocarbon (HC) emissions of Jatropha curcas biodiesel blends with various load for different EGR rates in comparison to baseline diesel. The HC emissions were reduced when the engine operated with biodiesel fuel blends and it varies with volume of blends. The HC emissions were reduced by 20%, 29% and 17% for JB10, JB20 and JB30 fuel blends respectively at 75% load for 600 FIP at 14^0CA BTDC injection timing compared to diesel fuel. The possible cause for the reduction in the HC emissions when fuelling with biodiesel blends is availability of oxygen molecules in the biodiesel compared with the diesel fuel. This can also decrease HC from the locally over rich mixture and also lower carbon and hydrogen ratio (Gopa 2014) when compared with the diesel fuel (Ozener 2014, Agarwal.A.K 2015).

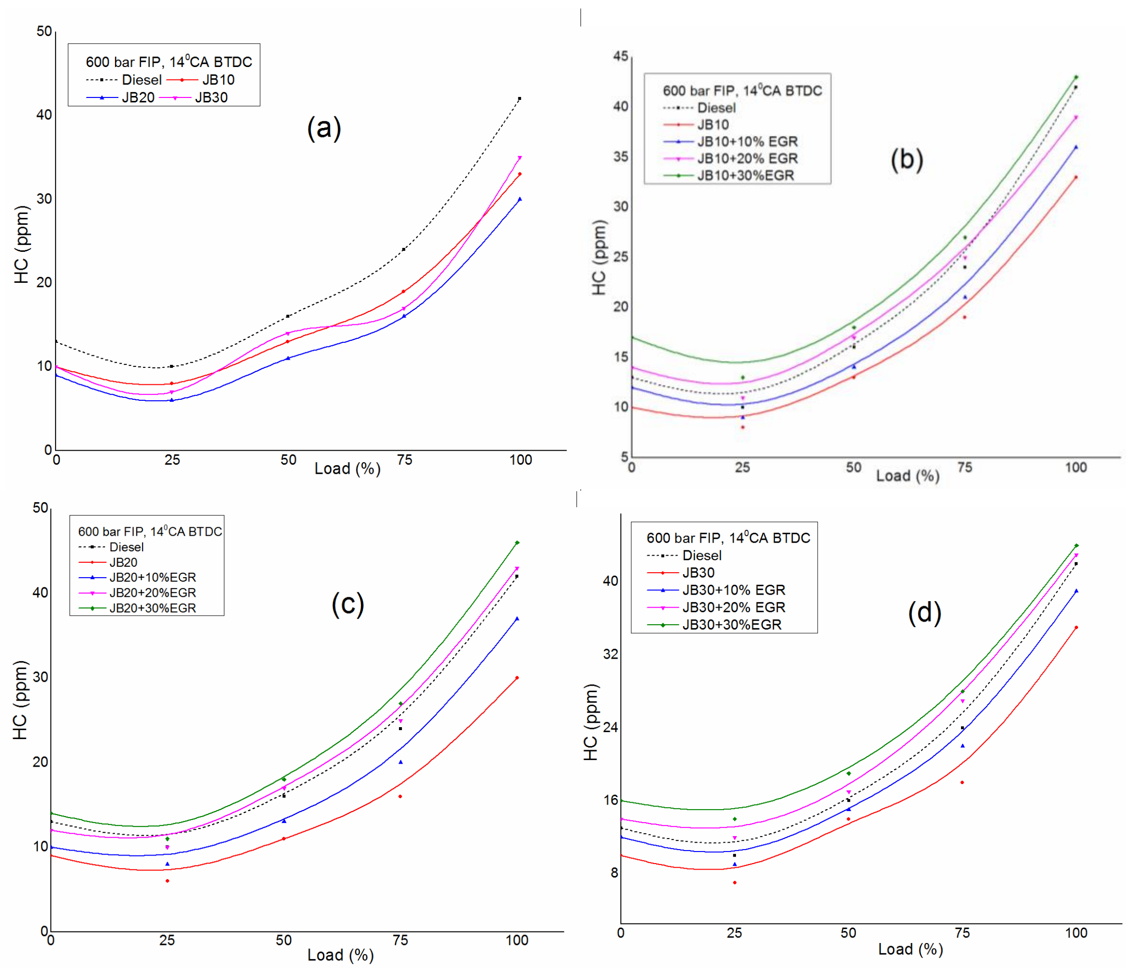


Fig. 5.9 Effect of fuel blends on HC at 600 FIP and 14^0 CA with different EGR rate

These combined factors help in improved and more complete combustion process and reduces the HC emissions from engine exhausts.

With the application of EGR rates 10%, 20% and 30%, it is noticed that the HC emissions increases compared relatively with the biodiesel blends as shown in Figure 5.9 (c-d). Further the value of HC at 10% EGR rate with all blends is still lower than of diesel operation due to oxygen content of blends. Whereas HC is increased by 4.1% and 8.3% for 20% and 30% EGR rate with JB20 blend compared to diesel fuel at 75% load for 600 FIP at 14^0 CA BTDC injection timing. The increase of HC emission with EGR rate may due to lower excess oxygen presence which results rich air-fuel mixture at different locations inside the combustion chamber during combustion. Hence, this non-homogeneous mixture does not combust accordingly and develops more HC emissions.

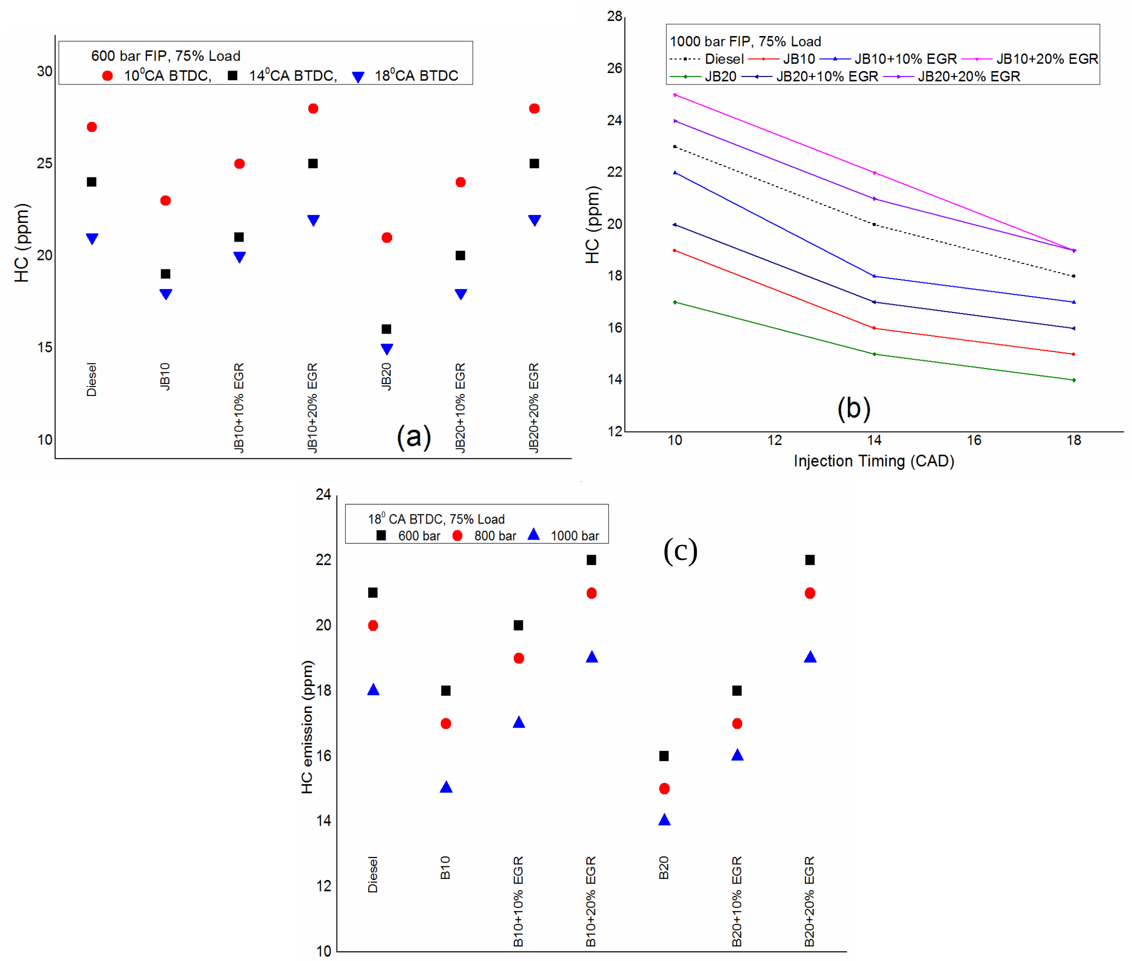


Fig. 5.10 Effect fuel blends on HC emissions at various injection timings and FIP

With respect to engine operation at different injection timing, the advancing the injection timing by 4 crank angle degrees from 14°CA to 18°CA BTDC caused the HC emission reduction, advanced injection timing produced the higher cylinder temperature and increasing oxidation process of HC during expansion and exhaust processes. However, retarding the injection timing to 10^0 CA BTDC caused increase in the HC emission at the same test as discussed above. HC for JB20 blend at 18^0 CA BTDC is reduced by 6.25% compared to 14^0 CA at 600 bar FIP, whereas for 10°CA BTDC, it is increased by 18% at same FIP with 75% load as shown in Figure 5.10 (a-b).

The HC emissions are reduced with increase of FIP from 600 bar to 1000 bar because of complete combustion at higher injection pressures as shown in Figure 5.10(c). HC value for JB20 is lower by 13% and 20% respectively for 800 and 1000 bar FIP compared to 600 bar at 18 CA BTDC for 75% loading condition.

5.1.3.4 Carbon monoxide (CO) emissions

CO is one of the intermediate compounds formed during the combustion process. Formed as the result of the incomplete combustion of the fuel, which contain less or no oxygen in their molecular structure (Ozener 2014). CO formation mainly depends on air fuel equivalence ratio, fuel type, combustion chamber design and fuel injection parameters such as injection timing and pressure (Gopa 2014). Blending biodiesel with diesel decreases the CO emissions, this is the implication of adding oxygenate fuels which can decrease CO from the locally over rich mixture and the decrease level vary with volume of blends. Moreover oxygen enrichment is also favorable to the oxidation process of CO during expansion and exhaust processes, CO emissions are shown in Figure 5.11(a). With fuelling JB10, JB20 and JB30 blend carbon monoxide emission are lowered by around 18% and 25% and 13% respectively when compared to diesel fuel at 600 bar FIP,14^0 BTDC at 75% load condition.

With the application of various EGR rates to blends, the CO emissions in the engine exhaust increase rapidly, due to reduced oxidation of CO to CO_2, here EGR dilutes oxygen concentration making inlet mixture to become richer than stoichiometric, also

EGR application lowers combustion temperature. Maximum increase in CO emission is noticed for 30% EGR rate at all conditions. From the Figure 5.11(c-d), it clear that CO emissions are still lower than diesel operating condition by 6% at 10% EGR rate due to usage oxygenated fuel, however CO emissions are higher than diesel operating condition at 20% and 30% EGR rate. CO values are found to be higher by 18%, 31% respectively for 20% and 30% EGR rate compared to neat diesel operation for JB20 blend with 75% load, 600 bar FIP with 14^0 CA injection timing.

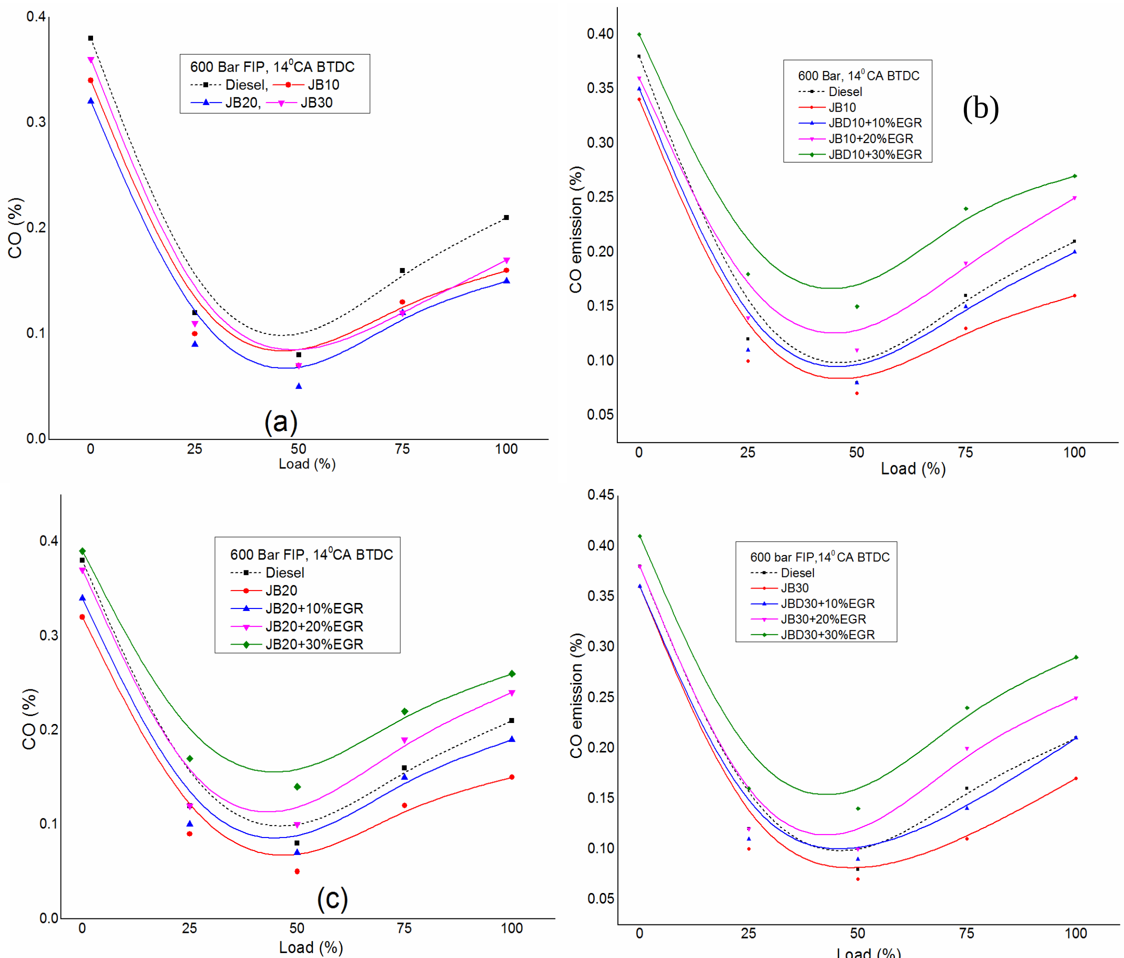


Fig. 5.11 Effect of fuel blends on CO at 600 FIP and 14^0 CA with different EGR rate

Figure 5.12 (a-b) shows the variations in carbon monoxide emissions at different injection timing and fuel injection pressure for various biodiesel blends with different EGR rates compared with respective baseline diesel operation. CO were lowest in case of advanced injection timing, here advanced injection caused increase of ignition delay which resulted in lean fuel zone and helps in faster atomization of fuel injected for 18^0 CA BTDC at fixed FIP, hence decreased CO emissions at this condition. Retarding fuel injection timing to 10^0 CA BTDC caused CO emissions to increase because fuel is injected near TDC and has less ignition delay leading inferior mixing. The fuel injected at this condition promotes majority of combustion takes place during the expansion stroke, which reduced the temperature and pressure (Agarwal.A.K 2015). At 18^0 CA BTDC, CO value is found to be lower by 20% compared 14^0 CA and at 10^0 CA is higher by 10% for the same operating condition.

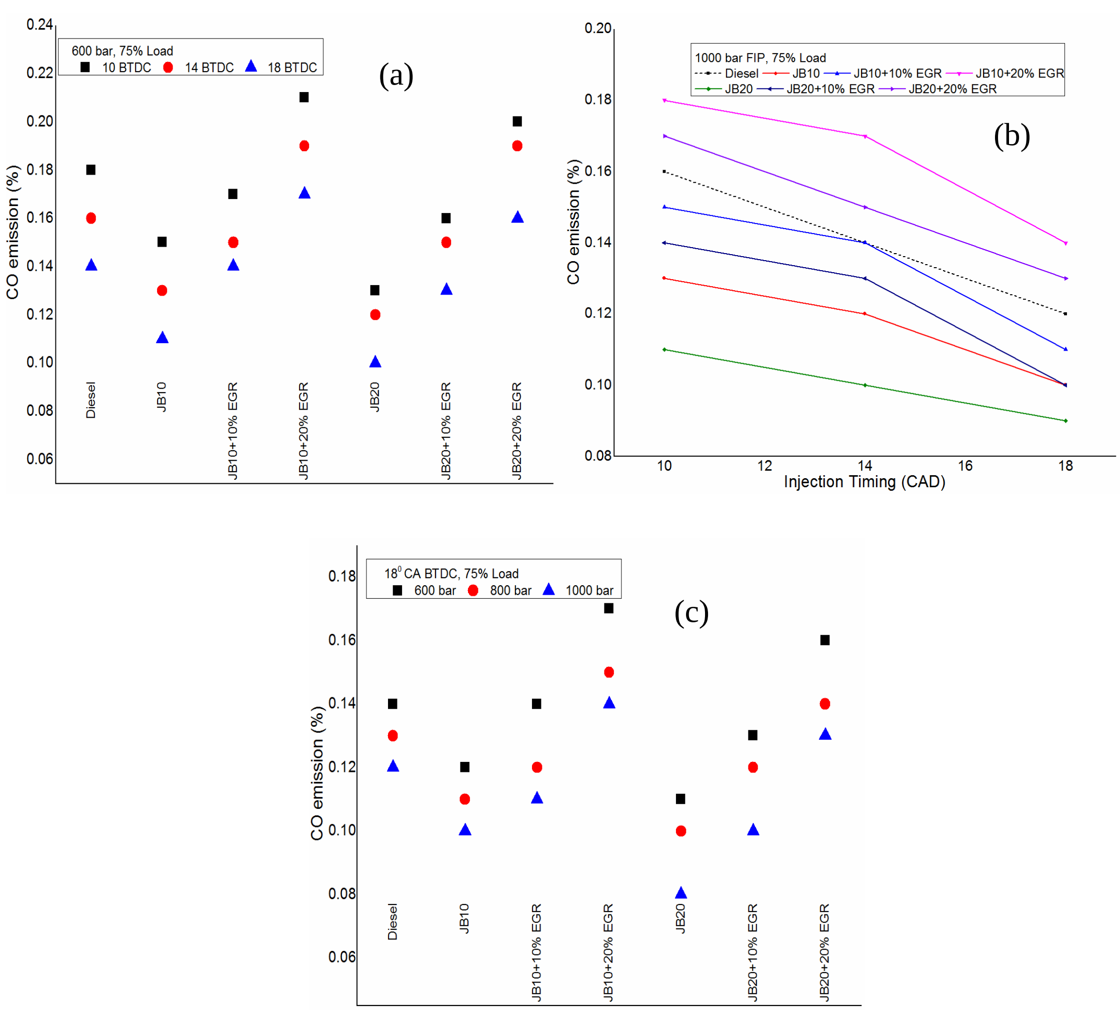


Fig. 5.12 Effect fuel blends on CO emissions at various injection timings and FIP

As noticed from from the Fig.5.12(c), that the higher FIP has lowest CO emissions for all blends and at all injection timings, this result implicates that increase of fuel injection pressure has reduced the value CO emission due to better mixing and superior atomisation of a fuel injected at that condition compared to lower FIP. CO for 800 and 1000 bar FIP is observed to be lower by 9% and 27% compared to 600 bar FIP at 18^0 CA BTDC injection, 75% load with JB20 blend.

5.1.4 Variation of Performance and Emission with 15% EGR for JB20 blend at 1000 bar FIP and 18°CA BTDC

From the above discussion it is clear that, biodiesel blend JB30 usage has resulted in reduction in performance due to lower heating and higher viscosity of the blend and with respect to application of EGR, 10% rate has given improvement performance with less HC, CO and smoke emissions than diesel fuel at all load conditions. Further application of 20% and 30% EGR rate is found to be less-beneficial in terms of performance at higher loads. Though NO_x emissions were much lower at this condition, but other engine exhausts emissions like HC, CO and smoke increased.

With respect to fuel injection strategies, out of 10^0CA, 14^0 CA and 18^0CA BTDC injection timings and 600, 800 and 1000 bar FIP, when engine operated with injection timing of 18^0CA BTDC and 1000 bar FIP has yielded better results in terms of performance and HC, CO and smoke emissions for JB20 blend except NO_x emissions.

Hence biodiesel blend JB20, 10% EGR rate, 1000 bar FIP and 18^0CA BTDC at 75% load are optimzed engine operating paramers for better performance and reduced emissions.

With the intention to reduce NO_x emissions further, engine was operated with 15% EGR rate for optimized values at 1000 bar FIP at 18°CA BTDC, from this experiment performance in terms of efficiency is found to be relatively better at 75% and full load condition with JB20 blend condition and is shown in Figure 5.13 (a).

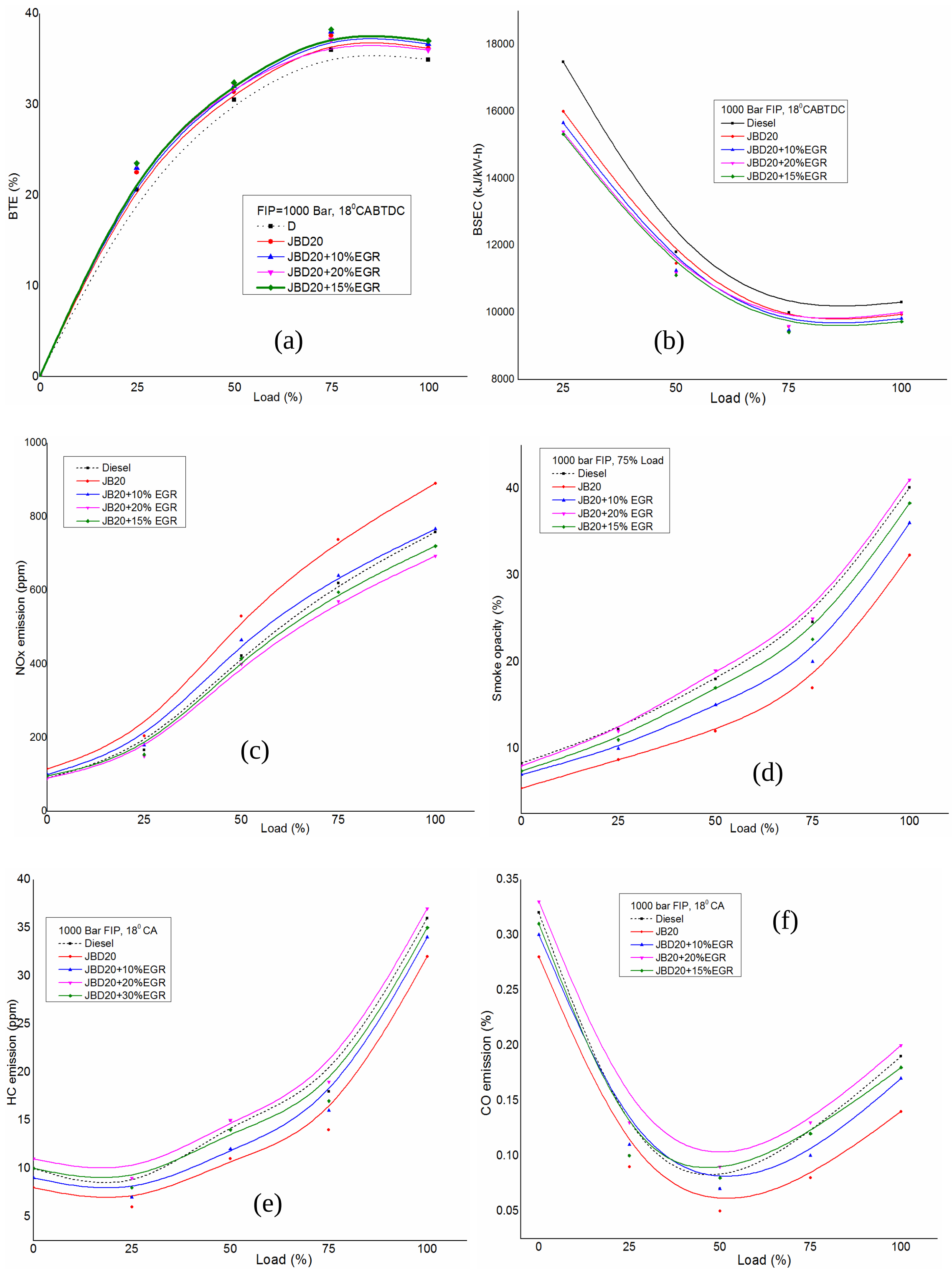


Figure 5.13 Effect of 15% EGR rate on performance and emissions at 1000 bar FIP and 18°CA BTDC with JBD20.

Among all EGR rates, maximum efficiency was observed at 15% EGR rate and also the lowest BSEC value is noticed at same condition due to better combustion at this operating conition as shown in Figure 5.13 (b). NO_x conversion efficiency is also noticed to be better and is lower than the conventional diesel operation due to lower combustion temperature and is shown in Figure 5.13(c).

From Fig. 5.13 (d-f), it is observed that 15% EGR produces smoke opacity, HC and CO emission value lower than or nearer to diesel operation, here oxygen defieciency caused by EGR is compansated by combined effect of fuel borne oxygen and better mixing at higher fuel injecion pressure. Hence 15% EGR rate is an optimised condition where engine can operate with less harmfull exhaust emissions without affecting performance.

5.1.5 Combustion analysis

5.1.5.1 In-cylinder pressure

Figure 5.14-5.17, displays the variation in cylinder pressure and position of peak pressure with respect to various biodiesel blends, EGR rate, different SOI timing, FIP relation to crank angle degrees (CAD). In a C I engine, the cylinder pressure depends on the burned fuel fraction and cylinder pressure characterizes the ability of the fuel to mix well with air and burn.

It is also observed that the as load increases, peak pressures also increase with all the fuels for a given speed, FIP and injection timing. This is because, at higher load more fuel is supplied and greater amount of mixture undergoes combustion which in turn results in higher compositional pressure. The difference in peak or maximum in cylinder pressure for different fuels is very minimal such results actually indicate better conversion efficiency of heat energy of a fuel into actual mechanical one for reformed fuel.

Figure 5.14(a), shows Jatropha curcas biodiesel blend JB20 has maximum peak pressure in comparison with the diesel fuel and other blends, this is may be due to better combustion because of more amount oxygen in the fuel and also added advantage of shorter ignition delay which is a result of higher cetane index of Jatropha curcas blend and hence more fuel burnt in diffusion stage combustion. Blend JB20 has a peak pressure of around 80.57 bar at 369^0 CA for 600 FIP and 14^0 CA BTDC.

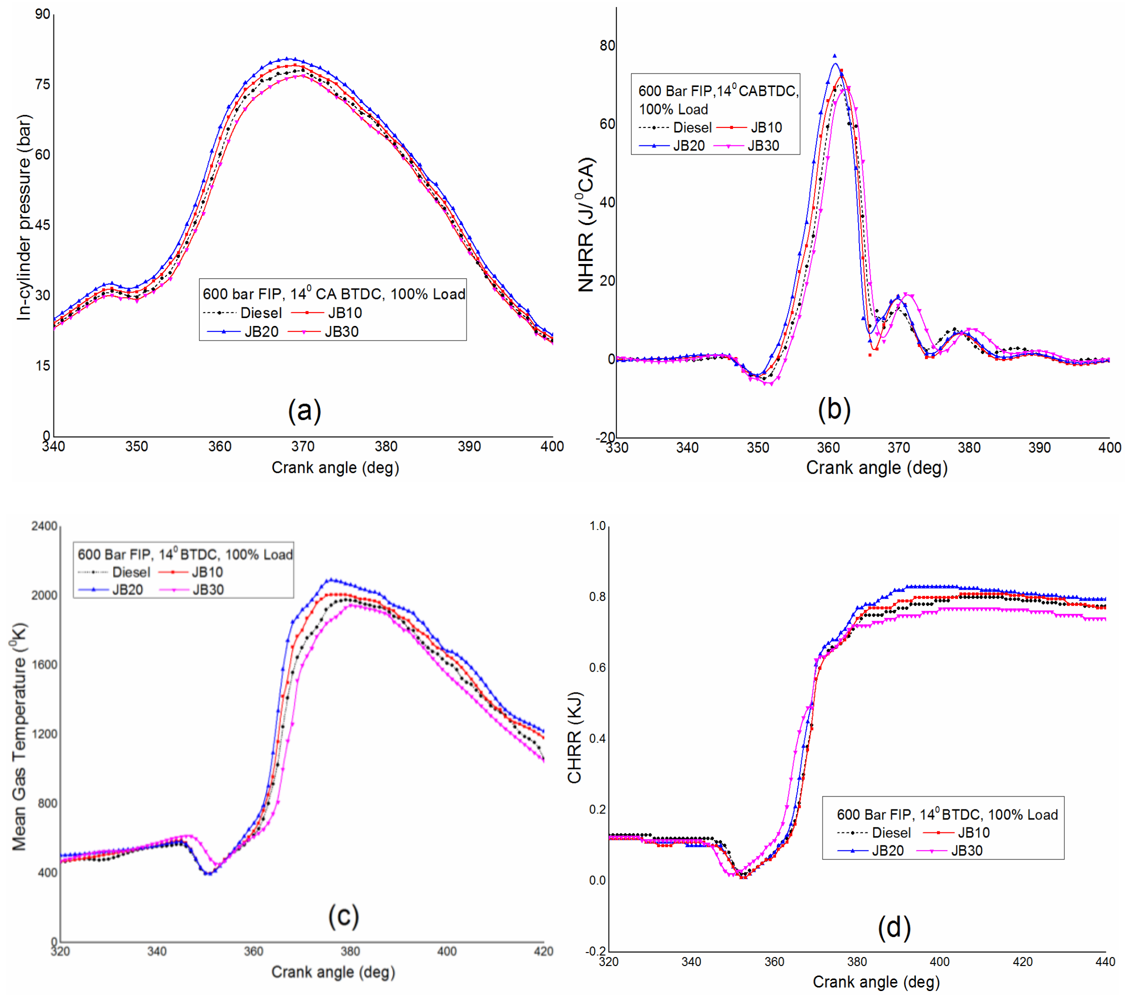


Fig.5.14 Variation of in-cylinder pressure, NHRR, MGT and CHRR for JB20 blend

Figure 5.15(a), shows the in cylinder pressure history for 20% biodiesel fuel blend with various EGR rates. with the application of EGR rates 10%, 20% and 30% for JB20 blend, The decrease in the peak pressure is observed and this reduction is may be due to dilution, chemical, thermal and ignition delay effect of an EGR.

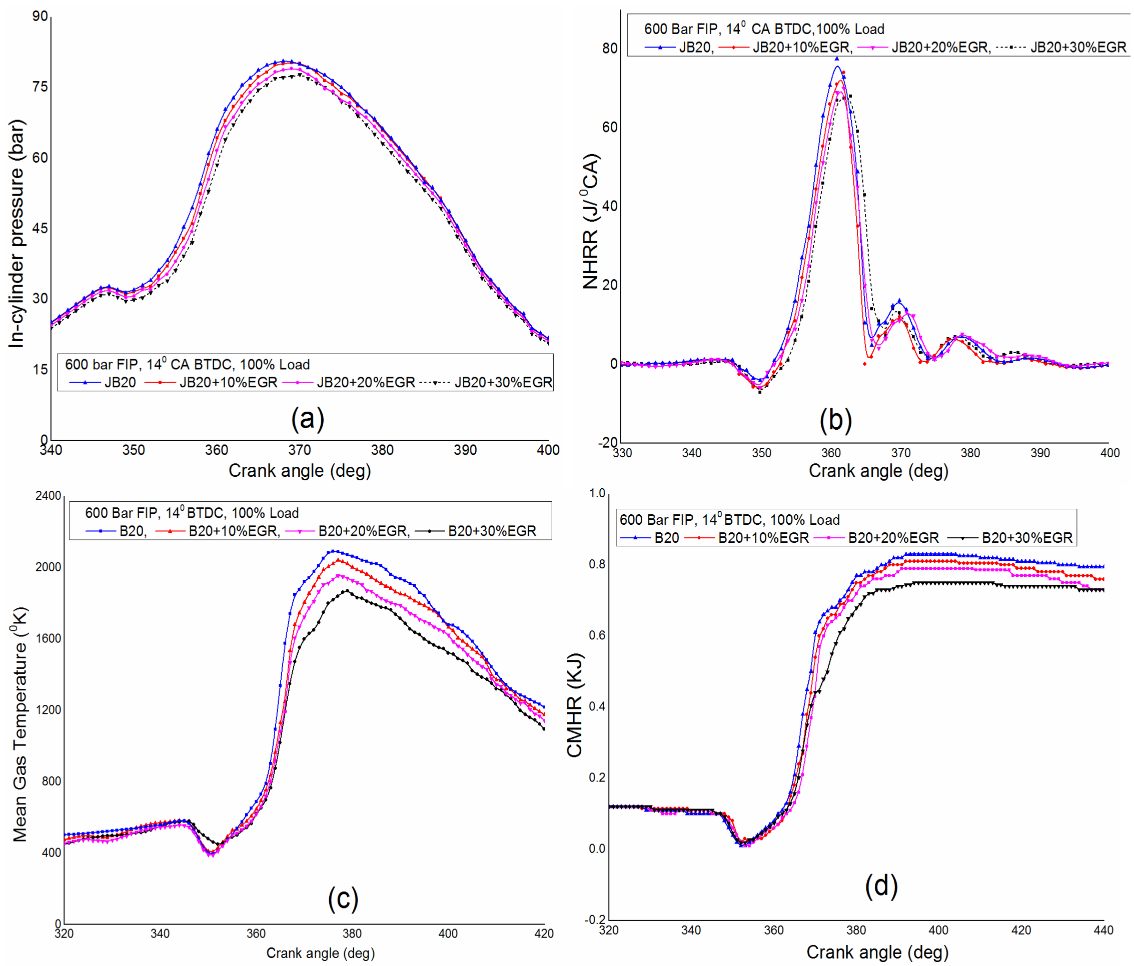


Fig.5.15 Variation of in-cylinder pressure, NHRR, MGT and CHRR for JB20 blend with different EGR rates.

It can be observed from Figure 5.16(a), that the change of injection timing significantly affects the cylinder pressure trends of diesel as well as Jatropha curcas biodiesel blends for full load and 600 bar FIP condition. The start of combustion occurs a little earlier for JB20 blend at 18^0 CA as compared to 14^0 C and 10^0 CA BTDC injection timing for full load due to reduced ignition delay at this condition. At 18^0 CA BTDC, JB20 blend has maximum peak due to better premixing in initial stage of combustion i.e. in premixed combustion zone. Here fuel mix well with air and burn hence high peak pressure and also maximum rate of pressure rise correspond to large amount of fuel burned in premixed combustion stage.

Figure 5.17(a), shows the variation in maximum cylinder pressure and position of maximum pressure with 600, 800 and 1000 bar FIPs at fixed fuel injection timing. For all tested fuels, maximum cylinder pressure increased with increase of FIP, this is due to improved fuel–air mixing at high fuel injection pressure.

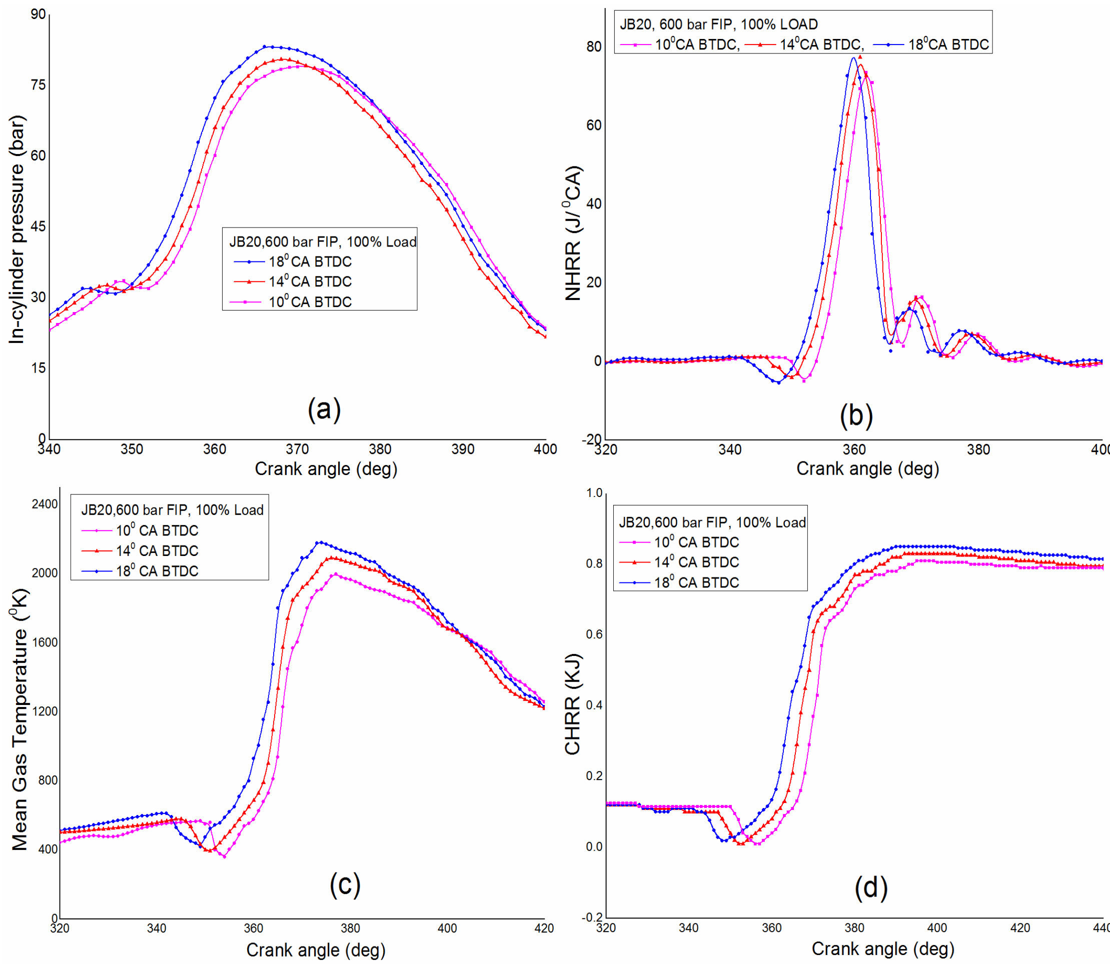


Fig.5.16 Variation of in-cylinder pressure, NHRR, MGT and CHRR for JB20 blend with different injection timings.

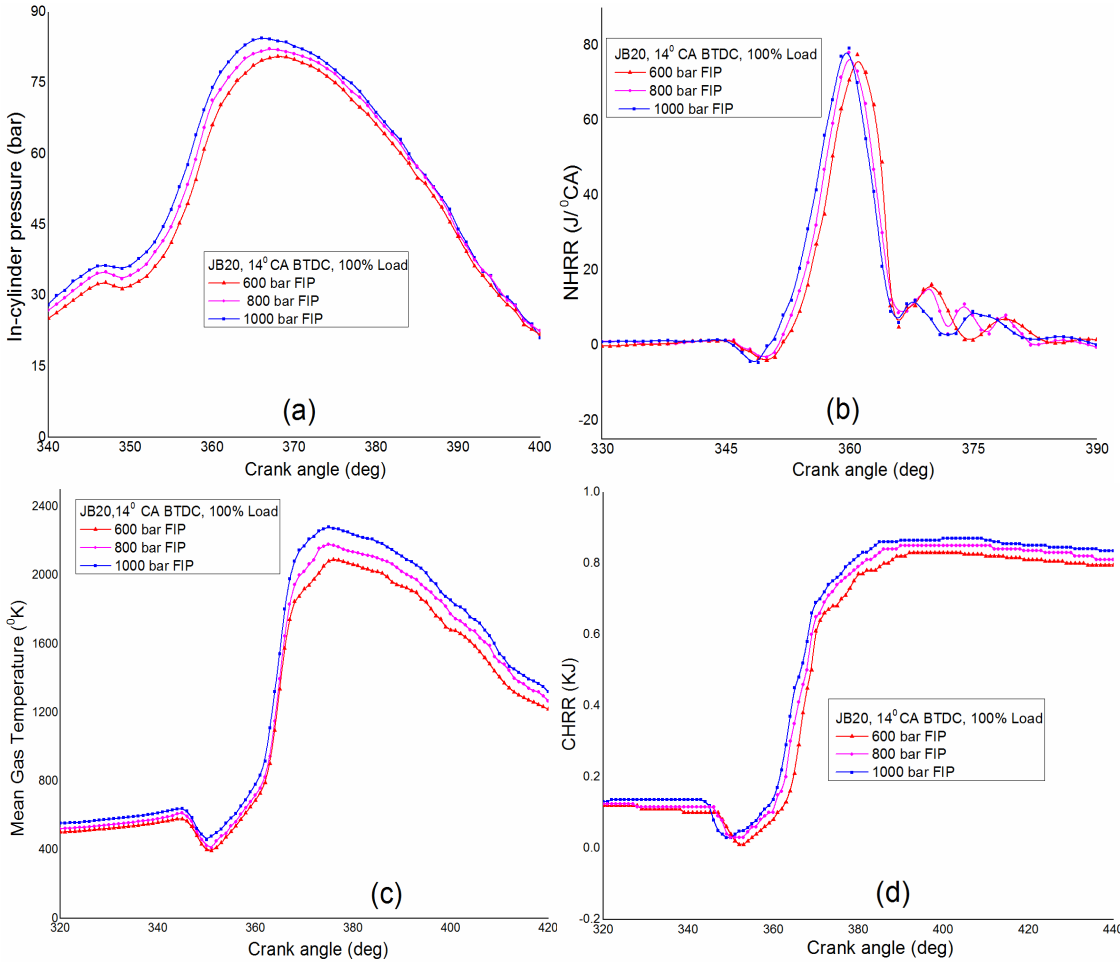


Fig.5.17 Variation of in-cylinder pressure, NHRR, MGT and CHRR for JB20 blend with different FIP.

5.1.5.2 Net heat release rate (NHRR)

The calculation of net heat release rate (NHRR) is an effort to get more information in relation to combustion process inside the cylinder. Moreover, physical and chemical properties of the fuel used for experimentation affect the heat release rate. Negative heat release was observed for all tested fuels after injection of fuel and before TDC due to cylinder charge cooling because of vaporization of the fuel accumulated during the ignition delay period. HRR becomes positive after the start of combustion (SOC). After the ignition delay, premixed air–fuel mixture burns rapidly followed by diffusion combustion when the HRR is controlled by rate of air–fuel mixing. The net heat release trends for diesel and JB20 blend for various EGR rates is as shown in Figure 5.14(b) -

5.17(b). Peak value for NHRR is observed for JB20 blend and earlier than other fuel conditions, this is due to better combustion of fuel borne oxygen. With respect to application of EGR rate, the NHRR is lesser because of higher EGR rates may lead to formation of low temperature flames and also the increased concentration of CO_2 and H_2O in the mixture will lead to slower reaction rate. Hence combination of low temperature combustion and reduced O_2 concentration during combustion period may have resulted in slow formation and propagation of flame rates in turn decrease of net heat release rates compared to JB20 blend without EGR conditions. It can be observed that shift in NHRR curves is consistent with shift in SOI timings and change in FIP. Start of heat release was slightly advanced for all biodiesel blends in comparison to other diesel fuel and this advancement was higher at advanced SOI timings and higher fuel injection pressure as shown in Figure 5.16 (b) and 5.17 (b).

5.1.5.3 Mean Gas Temperature (MGT)

Figure 5.14(c) - 5.17(c), in the cylinder mean gas temperature (MGT) at different engine operating conditions. With respect to Figure 5.14(c), among all tested fuels, JB20 blend has maximum gas temperature because higher in cylinder pressure. Figure 5.15(c), shows gas temperature for various EGR rates, here with application of EGR rate for JB20 blend MGT are reduced due to the replacement of oxygen by recirculated exhaust gases resulting in lower combustion temperature. It is well known that EGR has many effects like including the thermal effect, dilution effect and chemical effect. These combined effects of EGR result in lower gas temperature and lower combustion temperature. With increase of fuel injection pressure from 600 bar FIP to 1000 bar FIP a slight increase of in-cylinder temperature is observed due to better atomization and mixing at higher fuel pressure as shown in Figure 5.16 (c). Fuel injection timing plays important role in combustion characteristics, here advance injection timing of 18°CA BTDC has increased MGT mainly because of increased ignition delay vice versa MGT are less with retarded fuel injection timings reduces the in-cylinder pressure as shown in Figure 5.17(c).

5.1.5.4 Cumulative Heat Release Rate (CHRR)

Figure 5.14 (d) - 5.17 (d) shows the cumulative heat release at different engine operating conditions. Cumulative heat release ascends with higher engine load for all the tested fuels because of increase in quantity of fuel burnt during combustion with increasing load condition.

Figure 5.14 (d), shows a cumulative heat release rate for JB10, JB20 and JB30 blend in comparison with mineral diesel fuel at 600 bar FIP and 14°CA BTDC. It can be noticed from the Figure that JB20 has peak CHRR release rate due better combustion qualities. CHRR is reduced with lower blend ratios in mineral diesel may be due to lower oxygen concentration of the blends. With increase of EGR rate CHRR is lowered mainly due to dilution effect of EGR as shown in Figure 5.15 (d). The trend of CHRR with respect to fuel injection strategies is shown in Figure 5.16 (d) and 5.17(d), when engine is operated with various injection timings for a fixed FIP, the peak value of cumulative heat release rate is obtained for 18°CA BTDC and also same trend of results are obtained when the engine is operated at high pressure fuel injection of 1000 bar FIP at fixed injection timing for same load conditions. These similar trends of result are mainly due to the improved combustion characteristics of fuel.

5.2 Performance, emission and combustion characteristics of CRDI engine with multiple injection mode

5.2.1 Combustion analysis

The present investigation is to study the effects of pilot injection timings and the pilot injection quantity on the performance, emissions and combustion of CRDI Engine in conventional and low temperature combustion mode.

Multiple injection system divide the total quantity of fuel into two or more injections per combustion event. A pilot injection is also usually defined as an injection where 15% or less of the total mass of fuel is injected in the first injection (Mobasheri 2012).

The present experimental facility has a NIRA ECU (Engine Control Unit), this facilitates the operation of common rail fuel injection system through which many parameters such as fuel injection timing, injection pressure and pilot injection timing and quantity can be varied. Besides, NIRA adjusted the opening of EGR valve and control the EGR ratio.

To ensure better thermal efficiency and stable operation of the engine, the exhaustive experiments are conducted with single and various pilot injection timings. From the compared results, the injection timing of 10°CA BTDC is fixed for single main injection and pilot injection timings was selected in the range of 20° to 40°CA BTDC with high pressure fuel injection of 1000 bar operating with JB20 fuel blend for 15% EGR and without the EGR condition. Also, the pilot-injection fuel mass (quantity) was set at 5%, 10% and 15% of total fuel mass injected per cycle for above mentioned conditions.

The results of in-cylinder combustion pressure and net heat release rate of a twin cylinder CRDI engine under different injection strategies are shown in Figure 5.18

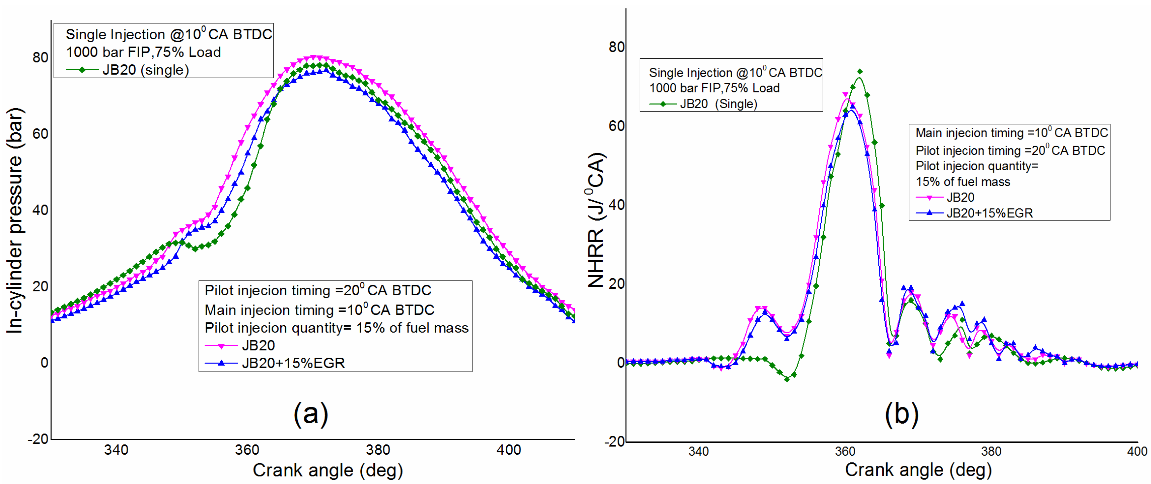


Fig.5.18 In-cylinder pressure and NHRR vs CAD for single and pilot injection

Figure 5.18(a), shows an in-cylinder combustion pressure for single injection compared with multiple injection strategy for fuel JB20 fuel and also with EGR rate 15% for fuel injection pressure of 1000 bar at 75% loading condition. Single injection has occurred at 10°CA BTDC and in case multiple injection, one pilot injection has been selected 20°CA BTDC and main injection was kept at the same as single injection.

In multiple injection combustion mode, a small amount of fuel is injected during the compression stroke which has a low in-cylinder pressure and temperature when fuel is pre-injected. A portion of injected pilot fuel was burnt and caused the ambient condition to have high pressure and temperature for the main injection event, once main fuel injected to this condition will enable fuel reaching the ignition point quickly, hence combustion of main injected fuel will be rapid and faster due to pilot fuel heat release rate as shown in NHRR Figure 5.18(b). Hence pilot fuel injection at 20° CA with 10° CA main injection has induced higher combustion pressure and lower peak heat release rate compared to the single-injection event at 10° CA BTDC (Huang 2016,Jeon 2015). Combustion pressure and heat release rate are lower with respect to EGR, compared to other conditions due to dilution effects (Huang 2015, Huang 2016).

5.2.2 Effect of different pilot injection timings on in-cylinder pressure and NHRR

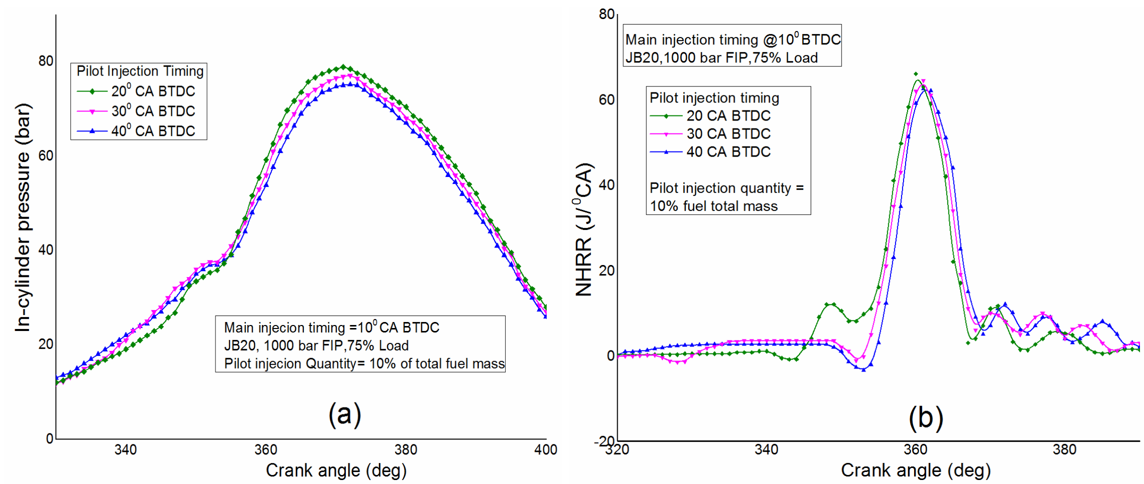


Fig.5.19 Effect of different pilot injection timings on in-cylinder pressure and NHRR

Figure 5.19(a), shows the results of in-cylinder pressure and heat release rate using the JB20 fuel with various pilot injection timings and 10°CA main injection. The peak combustion pressure and peak rate of heat release reduced with advance pilot injection timings. The pilot quantity of fuel injected at advanced timings, creates lean mixture due to larger cylinder volume and may improve air utilization and reduce fuel-rich regions (Suh 2011) and also additionally ignition delay prolonged during entire process. Hence lower combustion pressure and NHRR compared to retarded pilot timing 20° CA BTDC.

With advanced pilot injection timing, ignition delay will be longer, this makes the mixture of air and fuel much more dilute, making it difficult to combust until main injection event to happen. In these cases combustion will be activated only after injection of main fuel. The in-cylinder pressure for the pilot injection timing at 20°CA BTDC is higher than that of 30°CA and 40°CA BTDC. This is obviously high combustion pressure mainly because of the high ambient condition when pilot fuel is injected at less-interval between the pilot-main injection time. These condition makes combustion to occur rapidly and affect the subsequent burning, leading to the cylinder pressure increase (Huang 2015, Huang 2016, Yao 2010).

5.2.3 Effect of pilot quantity on in-cylinder pressure and NHRR

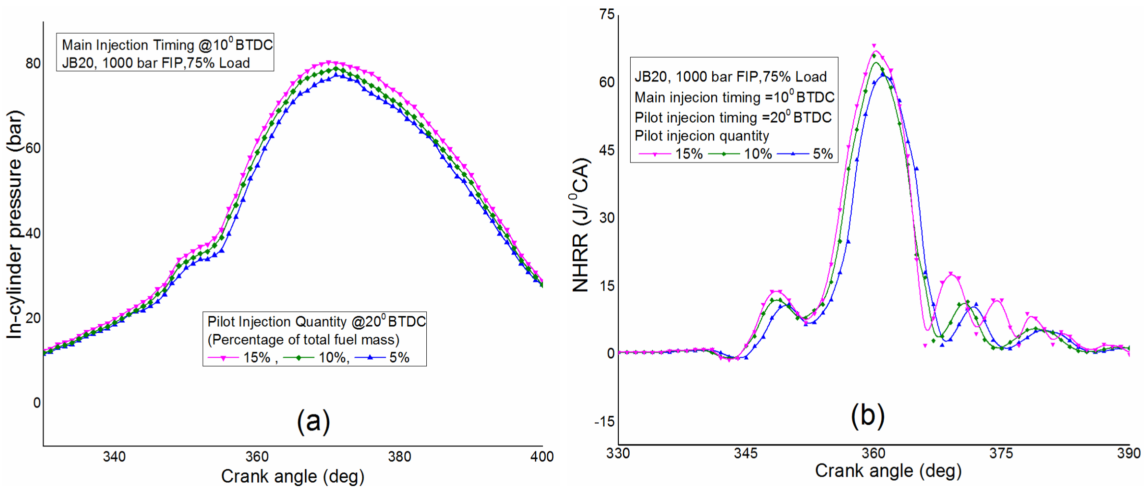


Fig.5.20 Effect of pilot quantity on in-cylinder pressure and NHRR

Figure 5.20 (a) & (b) shows the effect of various pilot fuel quantities on in-cylinder pressures and heat release rates of JB20 under different multiple injection strategies. As the injection event is split into two stages, main and pilot injections and with introduction of pilot fuel, the entire combustion period showed two different stages of heat release, specifically the heat release due to pilot fuel combustion and heat release from main fuel combustion. An increase in the pilot quantity led to a more amount of heat release during pilot injection stage and which shortened the ignition delay time making ambient condition to have elevated temperature and pressure for the main fuel injection which caused in an increased peak pressure value in the main-injection phase for higher amount of pilot injection stage (Huang 2016, Jeon, 2015, Yao 2010, Suh 2011).

5.2.4 Brake Thermal efficiency (BTE)

Figure 5.21 shows BTE for JB20 fuel with and without EGR condition for different multiple injection strategies. Pilot quantity of fuel is injected at 20°, 30° and 40° CA pilot injection timing and main injection has occurred at 10° CA BTDC. The BTE has highest value for 20°CA BTDC pilot injection compared to other injection timing, here pilot quantity of fuel mass injected before main injection creates lean and even mixture due to larger volume of cylinder and may improve air utilization and reduce fuel-rich regions compared to single main injection, hence higher brake thermal efficiency for multiple injection compared to single injection. BTE with15% EGR rate is more than JB20 fuel blend and this is due to reburning of un-burnt HC.

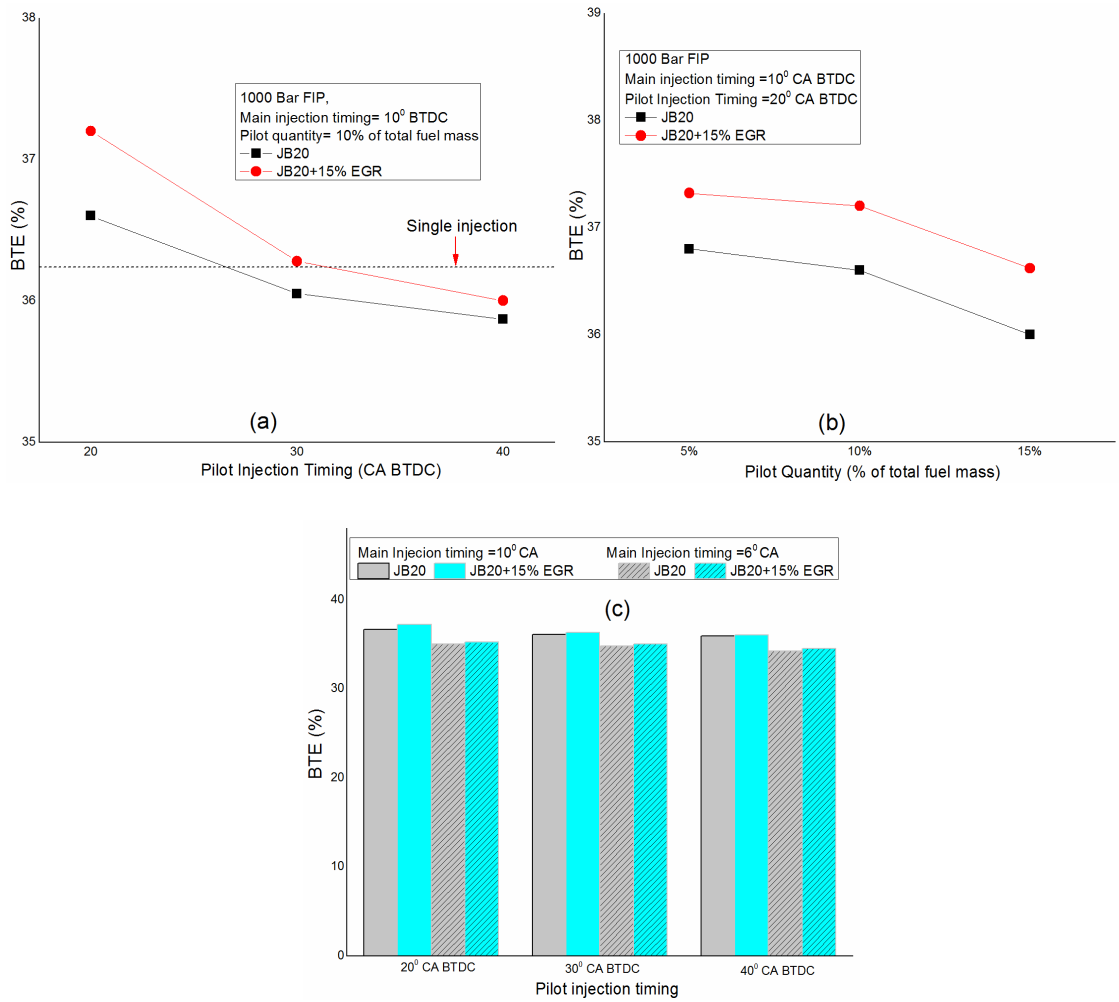


Fig.5.21 Variation of BTE for various pilot fuel mass and pilot injection timings

With further advancement of pilot injection timing, BTE is reduced because of lower cylinder pressure and temperature of the mixture than the 20°CA BTDC pilot injection timing conditions. The BTE improved for JB20 blend at 20°CA BTDC pilot injection timing in compared to JB20 blend with single injection. From Figure 5.21(a), it is observed that, the BTE of JB20 blend is improved by around 1% (without EGR) and

2.6% (with 15% EGR) for the 20°CA BTDC pilot injection in contrast with the single injection (shown as dotted horizontal line).

Figure 5.21 (b), shows effect of fuel injection quantity on BTE at 20°CA BTDC pilot timing, with the increase of pilot quantity more heat before the start of main combustion and this may negatively effect during compression and more work has to be done. Hence lower pilot quantity of 5% and 10% has more BTE compared to 15% pilot quantity.

Figure 5.21(c) shows the effect of main injection period for multiple injection strategy with different pilot injection angles. From the Figure it is clear that 10°CA BTDC main injection event has higher BTE than 6°CA BTDC main fuel injection with all pilot injection timings. This mainly because of increased in cylinder pressure and better combustion due more time available for combustion of main injected fuel at 10°CA BTDC. In case of 6°CA BTDC main injection, the combustion starts after TDC and lower peak pressure and peak temperature due to expansion process hence lower BTE compared to 10°CA BTDC.

5.2.5 Brake specific energy consumption (BSEC)

Figure 5.22 shows results obtained for BSEC for various pilot injection timings, pilot quantity and with two main timings. As compared to the single-injection mode (which is illustrated as dashed lines in Figure 5.22 (a)). The pilot injection conditions had a lower BSEC over a multiple injection in case for wide range of injection timings. In general 20°CA BTDC pilot injection has better combustion and energy efficiency over other pilot injection timing conditions compared to single injection strategy at 10°CA main injection. When fuel is pre-injected, start to burn and making ambient conditions have high in-cylinder pressure during the compression stroke for biodiesel. Reduction in BSEC up to 2.5% was achieved compared to the single injection for pilot fuel injection at 20°CA BTDC. Since retarded pilot injection timings are nearer to top dead center, they make main combustion more sensitive by ensuring that their combustion is complete near the end of the compression stroke compared to advanced injection timings. As a consequence, the decrease of BSEC value is noticed in case of retarded pilot timings than advanced injection timings.

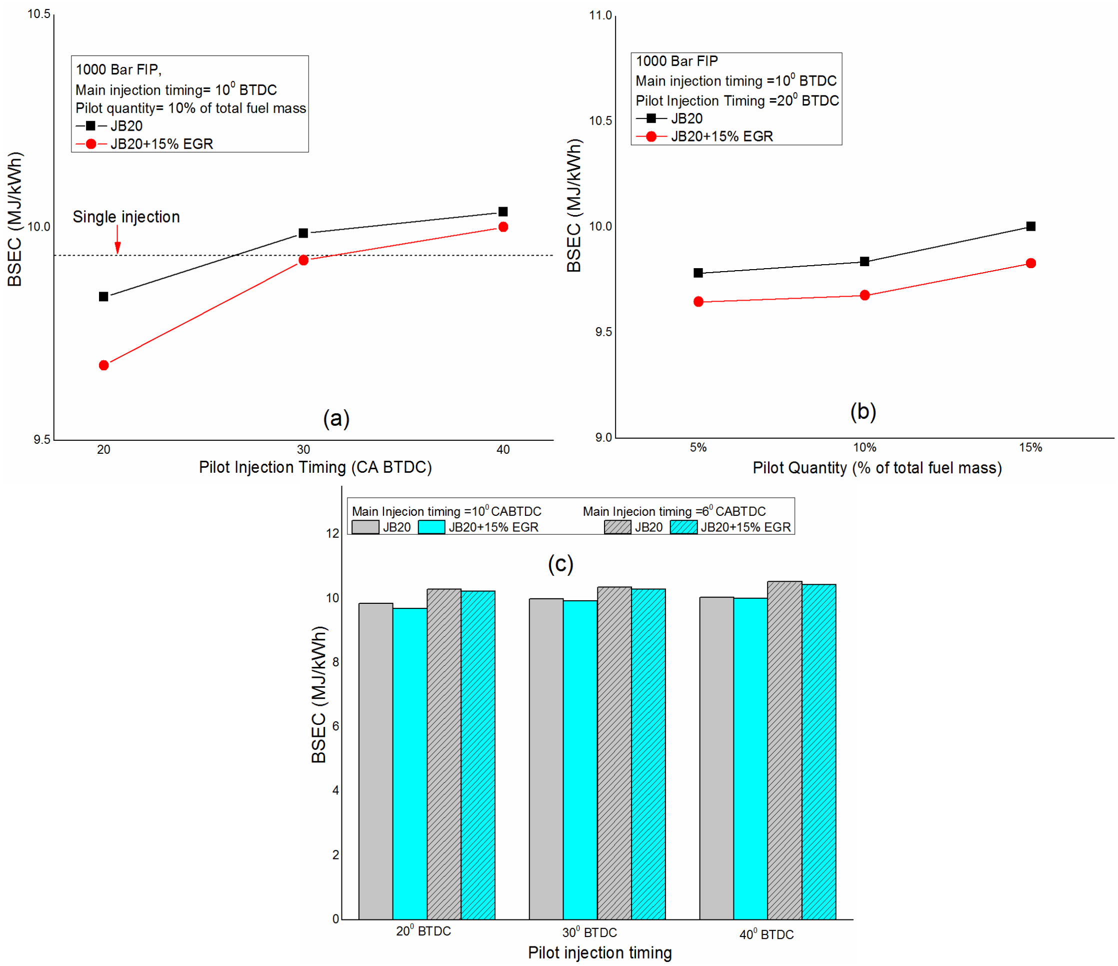


Fig.5.22 Variation of BSEC for various pilot fuel mass and pilot injection timings

Figure 5.22(b), shows the effect fuel mass on the BSEC, it is observed that the combustion was improved by introduction of pilot injection. The BSEC is lower with lower amount of pilot fuel quantity. This means that energy efficiency of the pilot injection was improved with a small injection quantity.

Figure 5.22(c) shows the effect of main injection period for BSEC in multiple injection strategy. From the Figure it is clear that 10° CA main injection event has lower BSEC than at 6° CA main fuel injection with all pilot injection timings. This is mainly because of better combustion 10° CA BTDC main injection.

5.2.6 Effect of multiple injection on emissions

5.2.6.1 Oxides of nitrogen (NO_x)

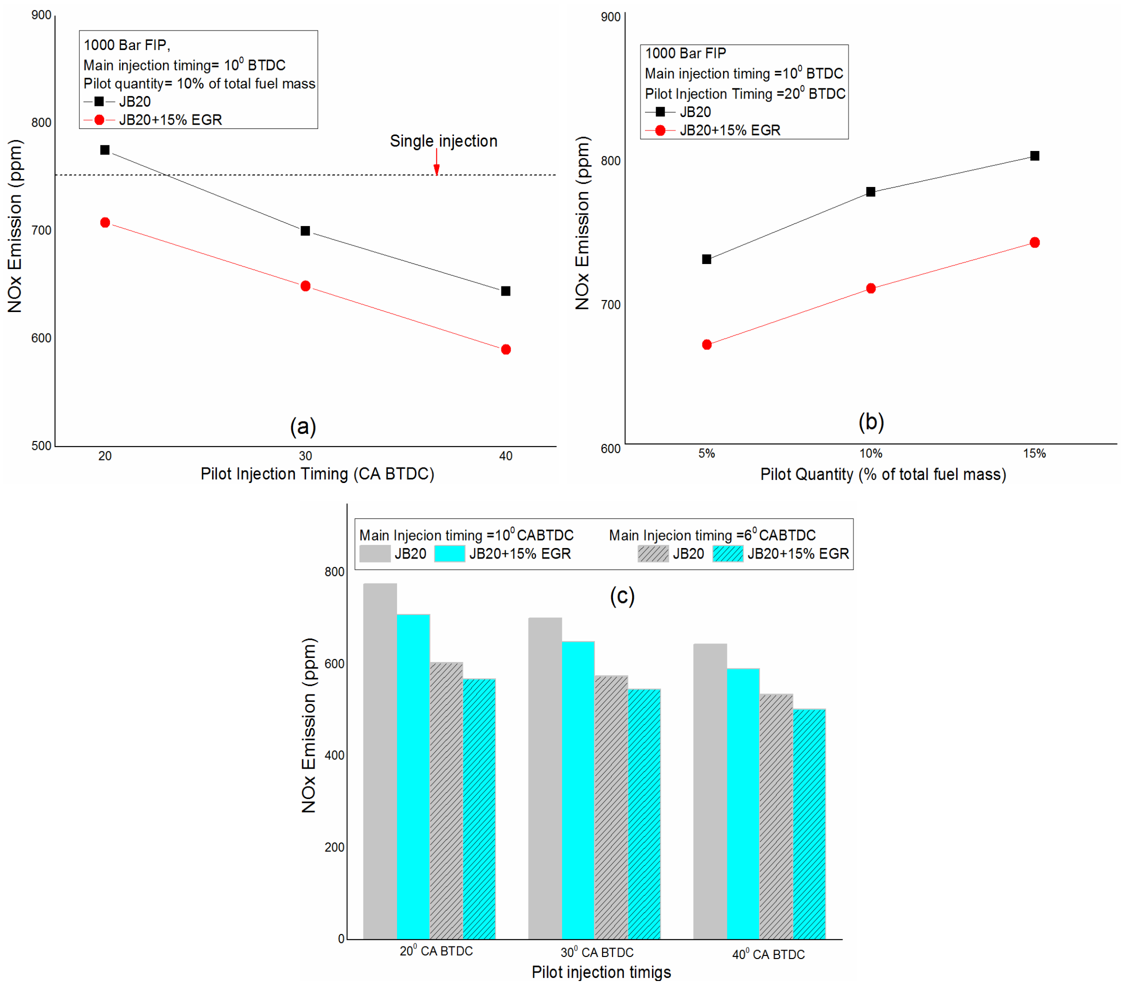


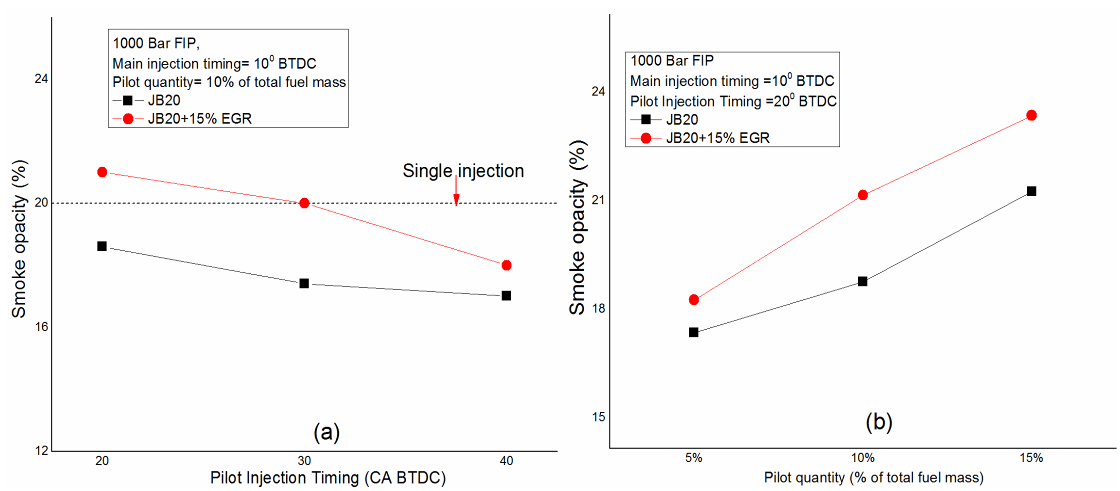
Fig.5.23 Variation of NO_x for various pilot injection timings and pilot quantity

Figure 5.23 shows the effect of pilot injection timing on NO_x emission for JB20 fuel blend with and without EGR. From Figure 5.23(a), NO_x emissions decrease with respect advancing pilot injection timing, here NO_x formation mainly depend on in-cylinder pressure and temperature which are reduced due to injection of fuel at these conditions (Huang 2015). The in-cylinder pressure and temperature are lower when fuel is pre-injected at advanced injection timings, here excessive leaner fuel/air mixture is formed due to extended ignition delay, NO_x reduces due to lower combustion temperature. The effect of retarded pilot injection on NO_x emissions was higher due to higher in-cylinder pressure and temperature because of shorter ignition delay period. NO_x level increased by 3% at 20° CA pilot timing, whereas NO_x found to be lower with application of EGR rate and reduced by around 9% at same pilot timings due to lower combustion temperature.

Figure 5.23(b) shows the effect of pilot injection fuel mass on NO_x emissions at multiple injection strategies with EGR rate is 15%, with the increase of pilot injection fuel mass, will enhance the overall mixture temperature leading to higher NO_x emissions.

Figure 5.23(c) shows effect of different main injection period for multiple injection strategy with different pilot injection angles. Here 10°CA BTDC main injection has higher NO_x emissions at all pilot injection timings compared to 6°CA BTDC injection. This is mainly due to increased in cylinder pressure and better combustion as more time available for combustion. In case of 6° CA BTDC main injection, a significant reduction of maximum temperature due to starting the fuel injection at the later injection timing in order to shift the main combustion phase into the expansion stroke. It also reduces residence time of high temperature combusted gas in the combustion chamber where NO_x forms actively (Suh 2011).

5.2.6.2 <u>Smoke opacity</u>



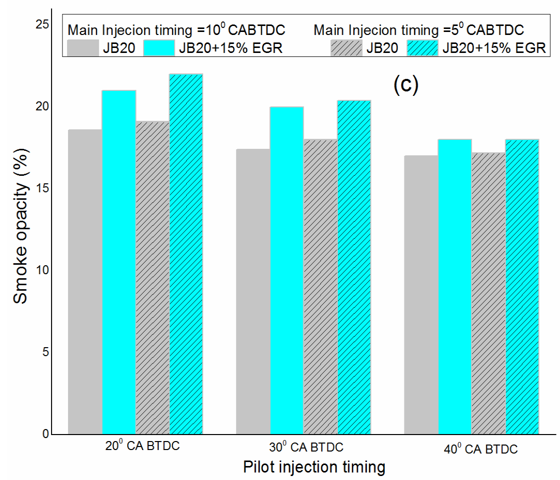


Fig.5.24 Variation of smoke opacity for various pilot injection timings and pilot quantity

Figure 5.24 (a) shows the effect of pilot injection timing on smoke opacity emission at different conditions in comparison with the single injection is shown as dashed horizontal line. From the graph it is clear that soot emission is lower than single injection for all pilot timings, especially at with the advance of pilot injection timings. As advanced pilot timings ensures the fuel having adequate time to mix with air and this could improve air utilization and reduce fuel-rich regions, both are beneficial to soot reduction and the smoke opacity emission decreases also due to longer ignition delay (Yao 2010, park 2015, Roh, Thurnheer 2011). With the introduction of EGR rate causes soot emissions ascending and smoke opacity is increased to 13% for 15% EGR rate compared to single injection, whereas without EGR condition smoke value is lower by 7% for same pilot injection timing of 20° CA BTDC.

Figure 5.24 (b) shows effect of pilot fuel mass injection at 20°CA pilot timing, here more mass means comparatively less reduction because increased amount pilot quantity injected before main injection, induces locally rich zone for combustion which results higher amount smoke level (Thurnheer 2011, Park 2015).

Figure 5.24(c) shows effect of main injection period for multiple injection strategy for two main injection angles for various pilot injection angles. Here 10° CA main injection has lower smoke opacity compared to 6°CA main injection at all pilot injection timings, this is mainly due to increased in cylinder pressure and better combustion as more time available for combustion. In case of 6^0 CA main combustion starts after TDC and lower peak pressure and peak temperature due to expansion process hence lower smoke opacity reduction. From another point of view, the early injection timing can significantly lessen soot emissions and low temperature due to the late injection timing influences on the less soot oxidation (Suh 2011).

5.2.6.3 **Hydrocarbon emissions (HC)**

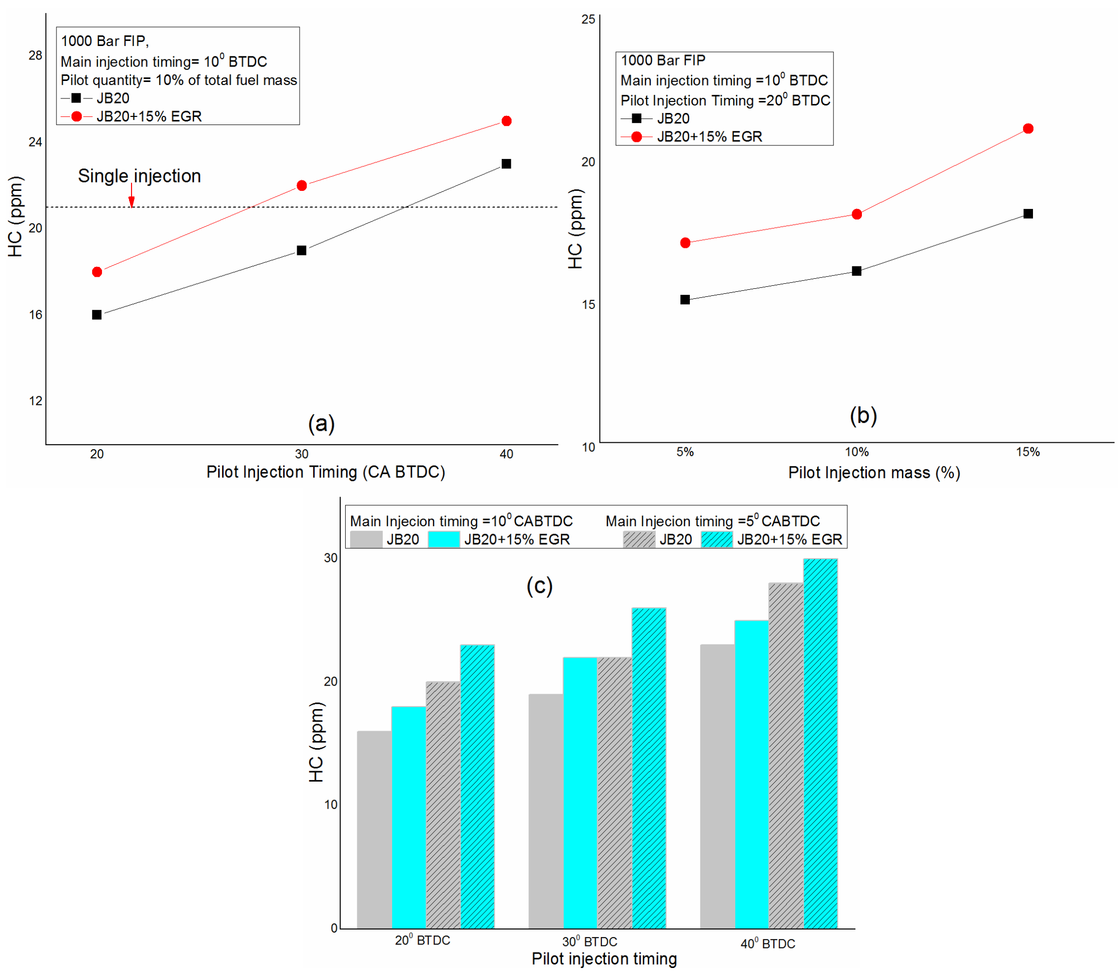


Fig.5.25 Variation of HC for various pilot fuel mass and pilot injection timings

Figure 5.25 (a) shows the effect of multiple injection on HC emissions compared single injection strategy. HC emissions decreased with retardation of pilot injection timings. Multiple injection could improve air utilization and reduce fuel-rich regions and both are beneficial to HC reduction. If too much advancement of the pilot injection occurred, HC emission is increased compared to retarded injection timings (Roh 2015, Suh 2011). In these regions excessively diluted air prevents the combustion process from either starting or going to completion. Thus it reduces the chances of complete combustion of main injection as a result more unburned HC emission remained during the increased gap of pilot and main injection compared to single injection (Park 2011, Yun 2011).

HC emission 20° CA pilot injection timings is reduced by 23% without EGR condition and is increased by 12% without EGR condition for same pilot timing due to lower combustion temperature which is a result of incomplete combustion. At the same time, the rich mixture is expected in higher fuel mass pilot strategies, thus increased HC emission are observed for higher pilot fuel ratio as shown in Figure 5.26(b).

Figure 5.26(c) shows effect of main injection period for multiple injection strategy with different pilot injection angles. From the Figure it is clear that 10° BTDC main injection has lower HC at 20° CA BTDC pilot injection compared to 6° BTDC injection this is mainly due increased in cylinder pressure and better combustion due more time available for combustion. In case of 6°CA BTDC main combustion starts after TDC and lower peak pressure and peak temperature due to expansion process hence more HC compared to 10° BTDC.

5.2.6.4 <u>Carbon monoxide (CO)</u>

The CO formation is primarily owing to the incomplete combustion of fuels in combustion chamber and is an intermediate product during the combustion process on account of localized oxygen deficit. Combustion temperature and excess air critically affect the oxidation process of CO to CO_2. Figure 5.26(a) compares the CO emissions of JB20 blend and EGR rate under different pilot injection strategies.

In case of multiple injections, when pilot fuel injected at much advanced conditions, portion of pilot fuel injected will either enter ultra-lean and too-low temperature zones or reach the vicinity of the combustion. Here, proper the mixing process between pilot fuel and air lead to lean mixture which will burn with more amount of heat release rate resulting in less CO emissions or enhancement of mixing process of pilot injected fuel and air make charge to become too-lean zone where combustion does not occur resulting lower amount pilot injection heat release rate, hence higher amount CO formation (Yao 2010, Yun 2016, Park 2015, Lee 2015). And also temperature is lower compared to retarded pilot injection timing due to the longer ignition delay of pilot injection and hence low CO oxidation rate compared to retarded pilot injection. In case of shorter interval between main-pilot injection strategy, the temperature is high hence induces low CO formation and reduced CO emissions (Yao 2010, Yun 2016, Park 2015, Lee 2015). CO emission for retarded pilot injection timings is decreased by 16% with JB20 blend without application of EGR. The introduction of a EGR ratio, since it causes CO emissions ascending by 10%, because the addition of EGR lowers combustion temperature leading to the incomplete combustion for same pilot timing.

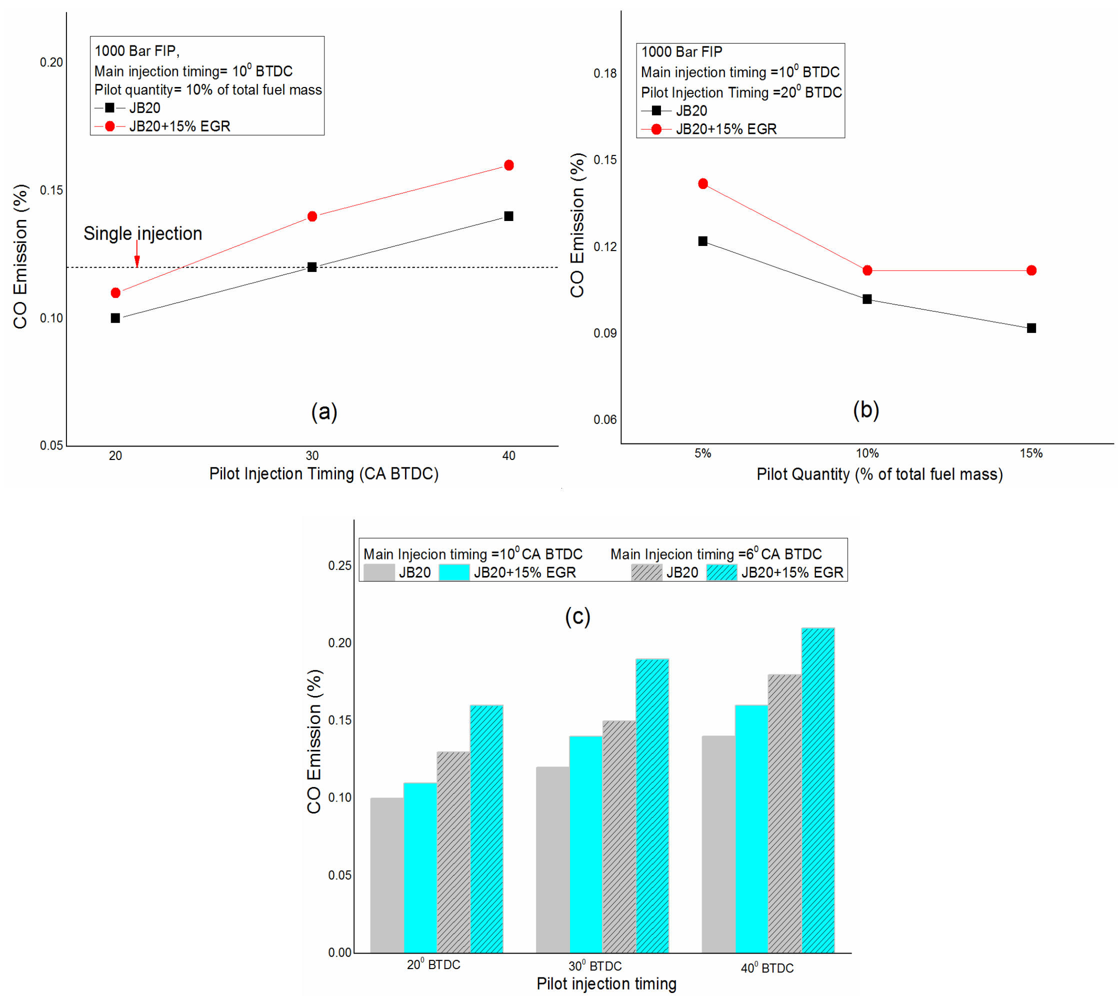


Fig.5.26 Variation of CO for various pilot fuel mass and pilot injection timings

With rising pilot quantity has reduced the CO emissions decrease evidently, The rise in in-cylinder temperature before main combustion promotes the oxidation of CO and finally less CO emissions are emitted at higher pilot fuel mass as shown in Figure 5.26(b).

Figure 5.26(c) shows effect of main injection period for multiple injection strategy for two main injection angles for various pilot injection angles. From the Figure it is clear that 10° CA main injection has lower CO emission at compared to 10° CA main injection at retarded pilot timings, this is mainly due to better combustion at 10°CA main combustion compared to 6° CA main injection timing.

CHAPTER 6

CONCLUSIONS AND SCOPE FOR FUTURE RESEARCH

The present experimental study was carried out to investigate the effect of biodiesel blends, fuel injection timings (IT), higher fuel injection pressure (FIP) and multiple injection with low temperature combustion (LTC) mode. LTC is achieved by the use of various flow rates of EGR on a twin cylinder CRDI for performance, emission and combustion characteristics by varying loads, the results obtained are analysed and the following conclusions are highlighted.

6.1 CONCLUSIONS

6.1.1 Effects of usage of biodiesel blends, EGR and fuel injection pressure (FIP) and injection timing (IT) on performance, emission and combustion characteristics with single injection mode.

- The usage of biodiesel blends in a CRDI engine has resulted in improved brake thermal efficiency (BTE) for JB10 and JB20 blends due to better combustion, vice versa the reduction in thermal efficiency is observed with JB30. With respect to Brake specific energy consumption (BSEC), reduction in BSEC is noticed compared to conventional diesel fuel and lowest BSEC is noticed for JB20 blend at 75% load by 4.6%. Smoke opacity, hydrocarbons (HC) and carbon monoxide (CO) emissions are also decreased with the usage of Jatropha biodiesel blends, however NO_x emissions increased by 18%. With respect to combustion analysis, JB20 fuel blend has peak combustion pressure, temperature and NHRR due to better combustion.
- Improvement in BTE, lower BSEC noticed with the application of 10% EGR rate vice versa decrease in performance and increase of BSEC is noticed for 20%, 30% EGR rate. NO_x emissions decreased for all EGR rates at all engine operating conditions, whereas increase of smoke, HC and CO emissions is noticed for 20% and 30% EGR rate. At 10% EGR rate smoke, HC and CO emissions are still lower than diesel operating conditions.

- Higher BTE, lower BSEC is obtained in respect of advanced injection timing of 18°CA BTDC compared standard injection timing 14°CA BTDC due to increased in-cylinder pressure, vice versa lower BTE, higher BSEC is obtained in respect of retarded injection timing of 10°CA BTDC because of reduced ignition delay and less time available for proper combustion of fuel and hence more exhaust emissions. Meantime, when engine is operated at higher fuel injection pressure with aid of CRDI fuel system, increase in BTE and NO_x with lower BSEC, HC, CO and smoke obtained at all conditions due to improved atomization of fuel at higher fuel injection pressure of 1000 bar.
- Among all EGR rates, maximum efficiency and lowest BSEC was obtained at 15% EGR rate for 75% load at 1000 bar FIP with JB20 blend due to better combustion, hence implementation of 15% EGR rate is better in terms of performance at all operating conditions. 15% EGR produces smoke opacity, HC and CO value lower than or nearer to diesel operation, here oxygen defieciency is caused by EGR is compansated by combined effect of fuel borne oxygen and better mixing at higher fuel injecion pressure. Hence 15% EGR rate is an optimised condition where engine can operate with less harmfull exhaust emissions without affecting performance.
- The change of injection timing significantly affects the cylinder pressure trends of various fuels, the start of combustion occurs a little earlier and has maximum peak pressure for JB20 blend at 18^0 CA as compared to 14^0 CA and 10^0 CA BTDC injection timing.
- Maximum cylinder pressure and MGT increased with increase of FIP, this is due to improved fuel–air mixing at high fuel injection pressure. With respect to application of EGR rate, the NHRR is lesser because of higher EGR rates may lead to formation of low temperature flames and also the increased concentration of CO_2 and H_2O in the mixture will lead to slower reaction rate

6.1.2 Performance, emission and combustion characteristics of CRDI engine with multiple injection mode.

- Pilot fuel injection at 20°CA BTDC with 10°CA BTDC main injection has induced higher combustion pressure and lower peak heat release rate compared to the single-injection event at 10°CA BTDC. The peak combustion pressure and peak rate of heat release reduced with advance pilot injection timings. As the injection is split into two stages, the heat is released in two stages, an increase in the pilot quantity led to a more amount of heat release and also increases of peak pressure.

- The BTE has highest value for 20°CA BTDC pilot injection compared to single injection mode due to improved air utilization and reduced fuel-rich regions. BTE with15% EGR rate is more than JB20 blend and this is due to reburning of un-burnt HC. The BTE of JB20 blend is improved by around 1% (without EGR) and 2.6% (with 15% EGR) for the 20° CA pilot injection in contrast to with the single injection. NO_x level is also increased by 3% at same pilot injection timing.
- NO_x emissions were higher at retarded pilot injection due to higher in-cylinder pressure because of shorter ignition delay period. Smoke, HC and CO emissions for 20°CA pilot injection timing are reduced by 7%, 23% and 16% without EGR condition.
- 10°CA BTDC main injection event has higher BTE than at 6°CA BTDC main fuel injection with all pilot injection timings. This mainly because of increased in cylinder pressure and better combustion.
- BTE, NO_x are higher with 10°CA BTDC main injection event compared to 6° CA BTDC main fuel injection with all pilot injection timings. This mainly because of increased in cylinder pressure and better combustion due to more time available for combustion of main injected fuel at 10°CA BTDC.
- BSEC, smoke opacity, HC and CO emissions are lower with 10° CA BTDC main injection compared to 6° CA BTDC main fuel injection with all pilot injection timings.

The engine operated with high pressure fuel injection of 1000 bar, injection timing of 18^0CA BTDC for JB20 blend with LTC mode gives better performance with lower exhaust emissions compared diesel fuel. Further multiple injection mode improved the combustion, performance with lower smoke, HC, CO and NO_x emissions compared to single injection mode when engine is operated with low temperature combustion (LTC) mode.

6.2 Scope for future research

Based on the results of this experimental investigation, further research work could be recommended in this interesting area of research. In view of recent development, following recommendations are made:

- It is suggested to carryout performance testing with still higher fuel injection pressures, changing the shape of combustion chamber, changing number holes and etc
- It is suggested to perform the comparative study of with other types biodiesel blends along with different types of additives.
- Computational and numerical study analysis can be undertaken for multiple injection strategy under low temperature combustion with high pressure fuel injection.
- The further reduction of exhaust emissions can be done by using exhaust gas after treatment devices such as SCR, 3 way catalytic converter etc

APPENDICES

APPENDIX I: Technical specifications of the dynamometer

Paramter	**Details**
Model	AG10
Make	Saj Test Plant Pvt. Ltd
End flanges both side	Cardon shaft model 1260 type A
Water inlet	1.6 bar
Minimum kPa	160
Pressure lbf/in2	23
Air gap mm	0.77/0.63
Torque Nm	11.5
Hot coil voltage max.	60
Continuous current amps	5.0
Cold resistance ohms Load 3.5kg	9.8
Speed max.	10000 rpm
Weight	130 kg

APPENDIX II: Technical specifications of Pressure transducer

Parameter	**Details**
Sensor name	Hydraulic pressure transducer with built in amplifier
Make	PCB Piezotronics, INC
Model	M108A02
Range, FS (5V output)	10000 psi
Useful range (10V output)	20000 psi
Maximum pressure	50000 psi
Resolution	0.4 psi
Sensitivity	0.5 mV/psi

Resonant frequency	300 kHz
Rise time	2 μs
Discharge time constant	1000 s
Linearity (zero based BSL)	2 %
Output impedance	100 ohms
Acceleration sensitivity	0.01 psi/g
Temperature coefficient	0.03 %/0F
Temperature range	-100 to +250 0F
Vibration	2000 g peak
Shock	20000 g peak
Sealing Hermetic	welded
Excitation (Constant current)	2 to 20 mA
Voltage to current regulator	+18 to 28 VDC
Sensing geometry	Compression
Sensing element	Quartz

APPENDIX III: Technical specifications of the AVL's Digas 444 Exhaust gas analyser

Measured Values	**Measurement Range**	**Resolution**
CO	0…10% Vol	0.01% Vol
HC	0…2000rpm	10ppm (0-2000)
CO_2	0…20% Vol	0.1% Vol
NO	0…5000 ppm	1 ppm

APPENDIX IV: Other technical specifications of the engine

Parameter	**Details**
Piezo sensor	Range 5000 PSI, with low noise cable
Crank angle sensor	Resolution 1 Deg, Speed 5500 RPM with TDC pulse
Data acquisition device	NI USB-6210, 16-bit, 250kS/s
Piezo powering unit	Make-Cuadra, Model AX-409
Temperature sensor	Type RTD, PT100 and Thermocouple, Type K
Load indicator	Digital, Range 0-50 Kg, Supply 230VAC
Load sensor	Load cell, type strain gauge, range 0-50 Kg
Fuel flow transmitter	DP transmitter, Range 0-500 mm WC
Air flow transmitter	Presure transmitter, Range (-) 250 mm WC
Software	“Enginesoft” Engine performance analysis software
Rotameter	Engine cooling 40-400 LPH; Calorimeter 25-250 LPH

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