Effect of operating parameters on the combustion, performance and emission characteristics of a diesel fueled CRDi engine

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ME14MTECH11011

A Dissertation Submitted to

Indian Institute of Technology Hyderabad

In Partial Fulfillment of the Requirements for

The Degree of Master of Technology

भारतीय प्रौद्योगिकी संस्थान हैदराबाद

Indian Institute of Technology Hyderabad

Department of Mechanical and aerospace Engineering

June, 2016

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Abstract

The ever growing energy demand and use of automobiles has made engines an important part of today's world. The slowly depleting fossil fuel reserves as well as the increasing environmental pollution made it necessary to develop efficient and cleaner engines. Diesel engines are known to have better efficiency but also higher amount of harmful exhaust emissions. Many techniques are being developed for improving fuel efficiency of diesel engine while simultaneously reducing the emissions.

For a diesel engine, compression ratio, fuel injection pressure and injection timings are very important parameters, which influence the engine performance, emissions, and combustion. A research diesel engine equipped with CRDi injection was used to experimentally determine the effect of compression ratio and fuel injection parameters on the combustion, performance and emission characteristics of the engine. Experiments were conducted at constant engine speed with two compression ratios (16 and 18) and three injection pressures (300, 500 and 800 bars) at variable engine loads. The combustion (peak cylinder pressure, HRR) and performance (BSFC, BTE) characteristics were found to be improved notably, while the exhaust emissions (CO, HC, NOx) were found to be reduced by significant amount.

Nomenclature

A	Cylinder area
AFR	Air-Fuel ratio
ABDC	After bottom dead center
ATDC	After top dead center
BBDC	Before bottom dead center
BDC	Bottom dead center
BMEP	Brake mean effective pressure
BP	Brake power
BS	Brake specific
BSFC	Brake specific fuel consumption
BTDC	Before top dead center
BTE	Brake thermal efficiency
CA	Crank angle
CI	Compression ignition
CO	Carbon monoxide
CO2	Carbon dioxide
CR	Compression ratio
CRDi	Common rail direct injection
DI	Direct injection
ECU	Electronic control unit
EGT	Exhaust gas temperature
FIP	Fuel injection pressure
Φ	Equivalence ratio

γ	Ratio of specific heats
HC	Unburned Hydrocarbons
HRR	Heat release rate
IC	Internal combustion
IMEP	Indicated mean effective pressure
JME	Jatropha methyl ester
L	Piston stroke
NOx	Oxides of nitrogen
p	Instantaneous in-cylinder pressure
PM	Particulate matter
ROHR	Rate of heat release
RPM	Revolutions per minute
SI	Spark ignition engine
SOI	Start of injection
SFC	Specific fuel consumption
TDC	Top dead center
θ	Crank angle
v	Instantaneous in-cylinder volume

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Chapter 1

Introduction

1.1 Background

The rate of consumption of the energy resources available continues to increase in spite of the significant progress being made to enhance the efficiency of their usage. Contributions to this increase in consumption include the rise in world population and improvements in the average standard of living. Despite increased utilization of wind, solar, and other emerging alternative energy sources, the combustion of naturally occurring fossil fuels still supplies the majority of the world's energy needs. It is expected that the combustion of fossil fuels will continue to be the prime source to rely on for energy for some time to come. This will be constrained, however, by the continued depletion of crude petroleum resources of quality, the need for ever-cleaner exhaust emissions, and the rapid progress being made in the development of renewable resources. Further, these depleting natural resources put more emphasis on developing highly efficient energy converters.

1.1.1 Internal combustion engines

The internal combustion engine (ICE) may be considered one of the most significant inventions that changed human life in recent times. It provides prompt and simple control of power generation while consuming a variety of commonly available fuels. They have been a primary power source for most of the transportation systems and small power generating stations, around the world for many decades.

Internal combustion engines date back to 1876 when Otto first developed the spark-ignition engine and 1892 when Diesel invented the compression-ignition engine. **[1]** Since that time these engines have continued to develop as our knowledge of engine processes has increased, as new technologies became available, as demand for new types of engines arose, and as environmental constraints on engine use changed. Internal combustion engines, and the industries that develop and manufacture them and support their use, now play a dominant role in the fields of power, propulsion and energy. The last twenty-five years or so have seen an explosive growth in engine research and development as the issues of air pollution, fuel cost, and market competitiveness have become increasingly important.

The basic purpose of Internal combustion engines is the production of mechanical power from the chemical energy contained in the fuel. Chemical energy of the fuel is first converted to thermal energy by means of combustion or oxidation with air inside the engine. Unlike *external* combustion engines, the energy is released by burning or oxidizing the fuel *inside* the engine. This thermal energy raises the temperature and pressure of the gases within the engine, and the high-pressure gas then expands inside the engine cylinder. Here, the fuel-air mixture before combustion and the burned gases after combustion are actual working fluids. The energy from these expanding gases is then directly converted into useful mechanical energy by the reciprocating piston-cylinder arrangement in the engine and supplied to a rotating crankshaft, which is the output of the engine. This rotating mechanical energy is then used for propulsion of vehicles, or to drive generators or pumps, and other purposes.

1.1.2 Classification of I C engines

Internal combustion engines are mainly classified into two categories based on the type of ignition.

(a) **Spark ignition (SI):** An SI engine starts the combustion process in each cycle by use of a spark plug. The spark plug gives a high-voltage electrical discharge between two electrodes which ignites the air-fuel mixture in the combustion chamber surrounding the plug. Commonly known as Petrol or Gasoline engine. They work on the principle of Otto cycle.

(b) **Compression ignition (CI):** The combustion process in a CI engine starts when the air-fuel mixture self-ignites due to high temperature in the combustion chamber caused by high compression. Commonly known as Diesel engine. They work on the principle of Diesel cycle. **[2]**

1.1.3 Diesel cycle:

In compression-ignition (CI) engine, fuel is directly injected into the cylinder which already compressed air at a higher pressure and temperature than the self-ignition temperature of the fuel. Hence the fuel ignites on its own without any dedicated ignition system like that in spark-ignition engine. Such engines work on heavy liquid fuels like diesel. Since the fuel and air are compressed separately and brought together at the time of combustion, the limitation on the compression ratio can also be overcome.

The compression-ignition engines work on an ideal cycle of operation known as Diesel cycle.

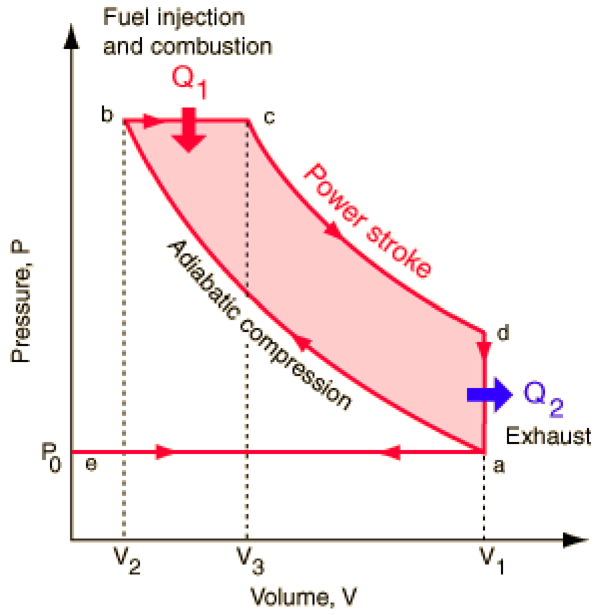

Figure 1: Ideal P-V diagram of Diesel cycle [2]

As shown in Figure 1, the ideal Diesel cycle consists of 4 major processes which are,

i. **Adiabatic compression or Compression stroke:** Fresh air charge taken inside the cylinder during suction is compressed adiabatically. The temperature and pressure of air rise to a high value.

ii. **Constant pressure heat addition or Combustion:** The fuel is injected into the cylinder, it forms a combustible fuel-air mixture and undergoes spontaneous combustion at constant pressure.

iii. **Adiabatic expansion or Power stroke:** The hot burnt gases expand inside the cylinder adiabatically and give useful work output.

iv. **Constant volume heat rejection or Exhaust:** The burnt gases are then driven out of the cylinder at constant volume. **[2]**

1.1.4 Diesel engine

Diesel engines operate on the principle Diesel cycle and rely on compression ignition of an atomized liquid fuel jet injected into the high-temperature and high-pressure cylinder air charge toward the end of the compression stroke.

Diesel engines need to have sufficiently high compression ratios as compared to spark ignition engines to ensure reliable, prompt, and well-controlled auto ignition of the injected fuel. They involve non homogenous fuel–air mixtures leading mainly to heterogeneous diffusion type combustion. The rapid energy releases produced require robust engine construction to withstand the resulting high mechanical and thermal loading rates.

The engine requires suitable liquid fuels that, in comparison to those of the spark ignition, are more prone to auto ignition and have higher cetane numbers, but at the same time very low octane numbers. Diesel engine usually operates on the overall lean equivalence ratio over the entire operating range i.e. excess air operation is employed throughout with intense turbulence and swirling fluid action provided so as to aid in the rapid atomization, vaporization, and subsequent mixing of the fuel and air.

The engine normally operates unthrottled and since it operates on comparatively lower fuel-air ratio, it tends to have superior work production efficiency and torque characteristics.

The exhaust emissions in diesel engines tend to show normally low levels of carbon monoxide and unburned hydrocarbons, but relatively high levels of NOx and particulates. At present, to render their exhaust gas emissions environmentally acceptable, most engine types require special treatment and costly equipment.

There are many improvements made in recent years in diesel engine design, manufacture, and performance, such as in their superior fuel injection systems, are very impressive. They led to significant improvements in key performance features, such as power production, brake torque characteristics, fuel utilization, and reduced exhaust emissions. [1]–[4]

1.2 Motivation

The global energy demand has shown drastic growth over the years. Contributions to this increase in energy consumption include the ever-rising world population and increasing urbanization and mechanization of the world. In spite of the increased utilization of non-conventional energy resources like wind, sunlight, tides, etc. fossils fuels are the major energy resource. This has led to a sudden rise in the demand of fuels derived from petroleum products. Petroleum products are extracted from crude oil and there are limited reserves in the world from which the crude oil is obtained. Statistics show that the present crude oil reserves in the world are going to deplete completely in coming 50 to 70 years if consumed at the current rate. As per the US department of energy, the world's oil supply will reach its maximum production and midpoint of depletion sometime around the year 2020.

Also, the exhaust emissions arising due to combustion of these fuels consists of harmful gases, particulate matter and smoke, resulting in continuous environmental pollution. To control this environmental degradation, governments are enforcing more and more stringent emission control legislations.

In today's world, gasoline and diesel are the most commonly used fuels for IC engines. Diesel engine offer advantage of higher compression ratios and are usually operated on overall lean equivalence ratio over the entire operating range. So it offers greater thermal efficiency than gasoline engines, but produces greater amount of exhaust emissions like particulate matter, NOx and smoke. Two methods are being used to reduce harmful engine emissions. One is to improve the technology of engines and fuels so that better combustion occurs and fewer emissions are generated. The second method is after-treatment of the exhaust gases. This is done by using thermal converters or catalytic converters that promote chemical reactions in the exhaust flow. Recent technological developments in diesel engine are focused on improving overall engine efficiency and keeping the exhaust emissions within acceptable limits. The engine operating parameters play an important role in reducing the emissions to an acceptable level. By varying the engine operating parameters like compression ratio, fuel injection pressure (FIP), and injection timing, engine performance in terms of power output and efficiency can be improved and the exhaust emissions can be reduced significantly. Thus improving the combustion, performance and emission characteristics of the diesel engine.

1.3 Objectives of research

To investigate the effect of compression ratio and fuel injection parameters (injection pressure, injection timing) on the combustion, performance and emission characteristics of a diesel fueled CRDi (Common Rail Direct Injection) engine.

The focus of this study is to find out the operational envelope of an existing facility of a compression ignition diesel engine equipped with CRDi injection and to carry out a detailed experimental testing and analysis of the effect of various engine operating parameters like compression ratio, fuel injection pressure (FIP), and injection timing on the performance, combustion and emission characteristics of the engine.

This study will help in understanding the basic knowledge of combustion process and operation of a compression ignition engine and assist in improving the overall engine performance as well as reducing the exhaust emissions.

Chapter 2

Literature review

2.1 Combustion in CI engine

The essential features of the compression-ignition or diesel engine combustion process can be summarized as follows.

Fresh air charge is taken inside the cylinder during suction stroke and compressed adiabatically during compression stroke. Fuel is injected by the fuel-injection system into the engine cylinder towards the end of the compression stroke. The liquid diesel fuel is injected at high pressures and high velocity as one or more fuel jets through very small nozzles present in the injectors. The fuel then atomizes into fine droplets and penetrates into the combustion chamber. The fuel vaporizes and mixes with the high-temperature high-pressure compressed cylinder air charge and forms a combustible fuel-air mixture. If the temperature and pressure inside the cylinder are above the fuel's ignition point, a spontaneous auto-ignition of the fuel-air mixture occurs after a certain delay period at different points inside the cylinder. The remaining unburned portion of the mixture is then further compressed due to the combustion and that helps in shortening the delay before ignition and the remaining mixture then burns rapidly. It also reduces the evaporation time for the remaining fuel. Injection continues until the desired amount of fuel is supplied in the cylinder. Atomization, vaporization, fuel-air mixing, and combustion processes continue as far as the fuel is being injected. The burned gases then expand inside the cylinder giving the useful work output during expansion stroke. Finally, these burned gases are driven out of the cylinder during exhaust stroke and the engine is ready for next cycle. **[1], [3]**

2.1.1 Stages of combustion in CI engine

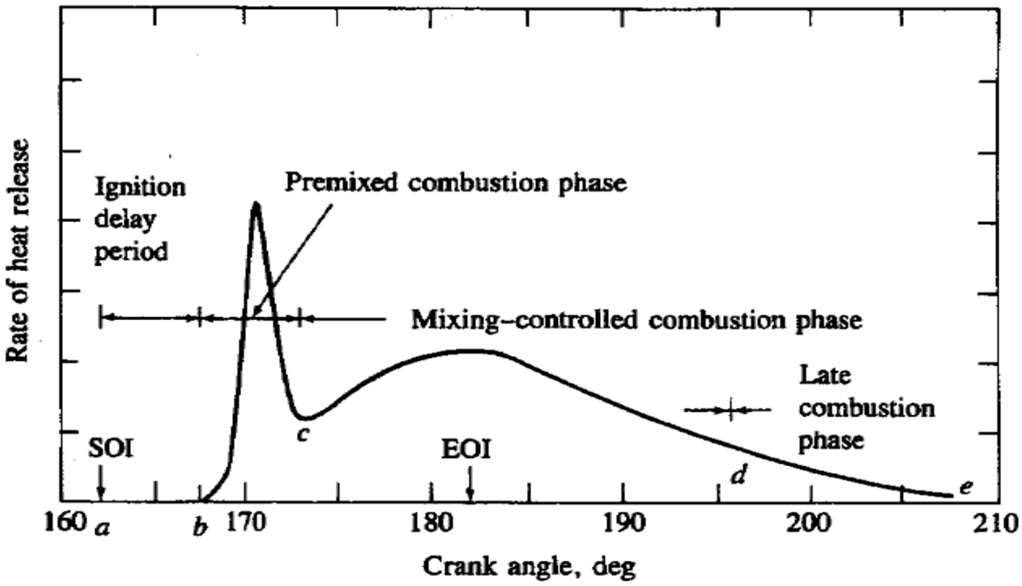

Figure 2: Stages of combustion in diesel engine [1]

Figure 2 shows a typical heat release rate diagram for a compression ignition engine. Overall, there are four stages in compression ignition combustion process as defined below.

i. **Ignition delay (a-b):**

It is the time period between the start of fuel injection into the combustion chamber and the start of combustion.

ii. **Premixed or uncontrolled combustion (b-c):**

The rich fuel-air mixture accumulated in the combustion chamber undergoes spontaneous rapid combustion during this stage. Uncontrolled combustion occurs until the peak pressure is achieved. The flame generated thereby spreads throughout the combustion chamber and ignites the remaining fuel. The heat release rate and pressure rise to a peak value during this stage.

iii. **Controlled or diffusion combustion (c-d):**

Once the fuel-air mixture premixed during the ignition delay is consumed, the heat release rate is controlled by the rate at which mixture becomes available for burning i.e. injection rate. The already established high temperature and pressure during uncontrolled combustion result into almost instantaneous vaporization and burning of the fuel that is then injected into the combustion chamber.

iv. **After-burn (d-e):**

A small portion of fuel molecules remain unburnt, as they fail to form a combustible mixture during the above stages. These least volatile fuel molecules undergo combustion during this stage. Reasons include incomplete fuel atomization, poor injection penetration, improper injection timing or a lack of sufficient air supply. **[1], [2]**

2.2 Exhaust emissions

Four major emissions produced by internal combustion engines are unburnt hydrocarbons (HC), carbon monoxide (CO), oxides of nitrogen (NOx), and solid particulates (PM).

Hydrocarbons are fuel molecules which did not get burned and smaller non-equilibrium particles of partially burned fuel. Carbon monoxide occurs when enough oxygen is not present to fully react all carbon to CO_2 or when incomplete air-fuel mixing occurs due to the very short engine cycle time. Oxides of nitrogen are created in an engine when high combustion temperatures cause some normally stable N_2 to dissociate into monatomic nitrogen N, which then combines with reacting oxygen. Particulates are the solid carbon soot particles that are generated in the fuel-rich zones within the cylinder during combustion due to low volatility of the fuel. **[1], [4]** In case of compression ignition or diesel engine, due to lean operating conditions, most of the fuel gets burned during combustion. A very small amount of fuel remains unburned which comes out in the exhaust contributing to a very low HC emission in diesel engine. Also, due to lean operating conditions, plenty of oxygen is available to convert all carbon atoms to CO_2. Hence CO emission is also less in diesel engine.

Due to higher compression ratios in CI engines, cylinder temperatures are very high as compared to SI engines, which aids in breaking down more and more diatomic nitrogen molecules (N_2) to nitrogen atoms (N), thus forming more amount of nitrogen oxides (NOx). Hence, NOx emissions are more in diesel engines.

Solid particulate emissions (PM) is also more in case of compression-ignition engines due to very low volatility if diesel fuel. **[3]**

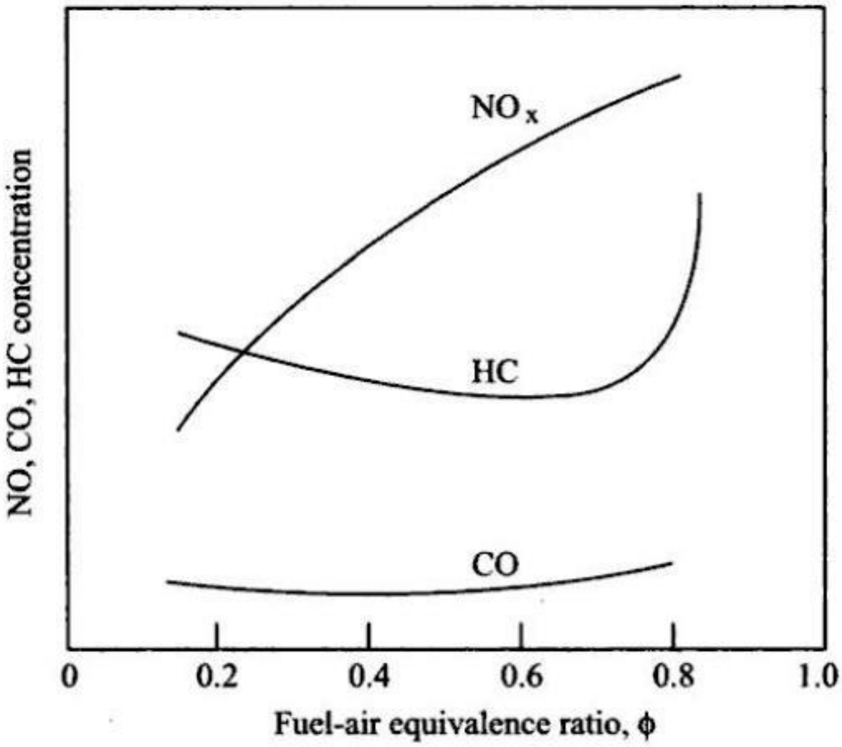

Figure 3: Emissions as a function of equivalence ratio for a CI engine [2]

Figure 3 shows a qualitative picture of HC, CO and NOx emissions with respect to equivalence ratio for a four stroke DI diesel engine. HC will decrease slightly with increase in the equivalence ratio due to higher cylinder temperatures making it suitable to burn any locally lean or rich fuel-air mixture. But, at high loads HC emission may rise again if the fuel in regions is too rich to burn. Due to available excess air, CO emissions will always be very low at all equivalence ratios. Since fraction of burnt gases increases in the cylinder contents during combustion, and also due to very high peak temperatures and pressures, NOx emission will increase gradually with increase in equivalence ratio. **[2]**

2.3 Common rail direct injection (CRDi)

CRDi stands for Common Rail Direct Injection meaning, direct injection of the fuel into the cylinders of a diesel engine via a single, common line, called the common rail which is connected to all the fuel injectors.

Whereas ordinary diesel direct fuel-injection systems have to build up pressure anew for each and every injection cycle, the new common rail (line) engines maintain constant pressure regardless of the injection sequence. This pressure then remains permanently available throughout the fuel line. The engine's electronic timing regulates injection pressure according to engine speed and load. The electronic control unit (ECU) modifies injection pressure precisely and as needed, based on data obtained from sensors on the cam and crankshafts. In other words, compression and injection occur independently of each other. This technique allows fuel to be injected as needed, saving fuel and lowering emissions.

Figure 4 shows a schematic diagram of the components in a CRDi system. In common rail systems, a high-pressure pump stores a reservoir of fuel at high pressure - up to and above 2,000 bars. The term "common rail" refers to the fact that all of the fuel injectors are supplied by a common fuel rail which is nothing more than a pressure accumulator where the fuel is stored at high pressure. This accumulator supplies multiple fuel injectors with high-pressure fuel. This eliminates the need of separate high-pressure pumps for each injector. The high-pressure pump only has to maintain a required pressure for a single reservoir.

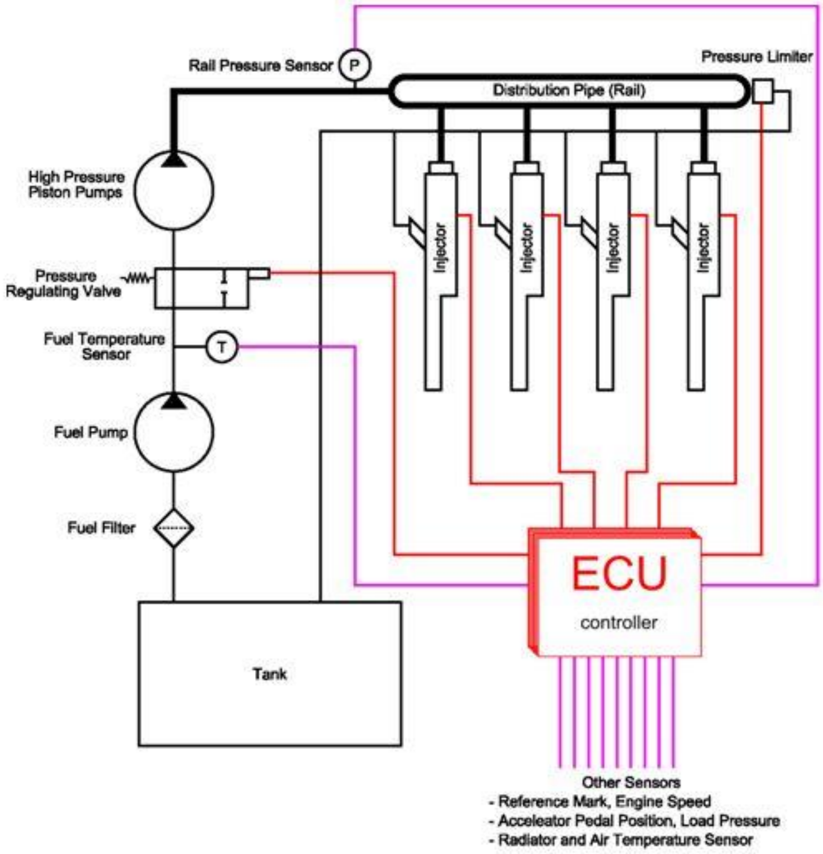

Figure 4: Schematic diagram of CRDi system [5]

In CRDi system, the fuel injectors are operated using solenoid valves which operate on electric current to operate the fuel Injection into the cylinder. The solenoid valves are operated by the central microcontroller of the CRDi control system based on the inputs from the sensors used in the system. This makes possible fine electronic control over the fuel injection time and quantity, and the higher pressure that the common rail technology makes available provides better fuel atomization. Based on the input from these sensors, the microcontroller can calculate the precise amount of the diesel and the crank angle when the diesel should be injected inside the cylinder. Using these calculations, the CRDi control system delivers the right amount of diesel at the right time to allow best possible output with least emissions and least possible wastage of fuel. The fuel injection pressure is independent of engine speed and load. The use of sensors and microcontroller to control the engine makes most efficient use of the fuel and also improved the power, fuel-economy and performance of the engine by managing it in a much better way.

CRDi system also gives the ability to employ pilot injection, post injection as well as multiple injections. In order to lower engine noise, the engine's electronic control unit can inject a small amount of diesel just before the main injection event ("pilot" injection), thus reducing its explosiveness and vibration, as well as optimizing injection timing and quantity for variations in fuel quality, cold starting and so on. Overall, CRDi system improves the power, response, efficiency and performance. It also significantly reduces the noise, emissions and vibration levels to a considerable extent. **[5]**

2.4 Injection pressure

In present diesel engines, fuel injection systems have designed to obtain higher injection pressure. So, it is aimed to decrease the exhaust emissions by increasing efficiency of diesel engines. When fuel injection pressure is low, fuel particle diameters will be larger and ignition delay period during the combustion will increase. Engine performance will be decrease since combustion process is not efficient. When injection pressure is increased, fuel particle diameters will become smaller. Since formation of fuel-air mixture is better during ignition period, engine performance will be increase. If injection pressure is too high, ignition delay period becomes very short. Possibilities of homogeneous mixing decrease and combustion efficiency falls down. The purpose of fuel injection system in a direct injection diesel engine is to achieve a high degree of atomization in order to enable sufficient evaporation in a very short time and to achieve sufficient spray penetration in order to utilize the full air charge and the combustion process to take place efficiently.

2.5 Effect of operating parameters

For a diesel engine, compression ratio, fuel injection pressure and injection timing are the most important operating parameters which influence the engine performance, combustion and emissions. Varying these parameters is exercised by many researchers as a practice for improving overall engine performance. **[6]–[14]** Higher compression ratios and fuel injection pressures help not only in maximizing performance characteristics of engine, but also in reducing exhaust emissions up to a certain extent.

2.5.1 Effect of fuel injection parameters

Agarwal et al. **[6]** investigated the effect of fuel injection pressure and timing on the combustion, emission and performance characteristics of a single cylinder research diesel engine at a two fuel injection pressures (FIP) wiz. 500 and 1000 bars and different start of injection (SOI) timings.

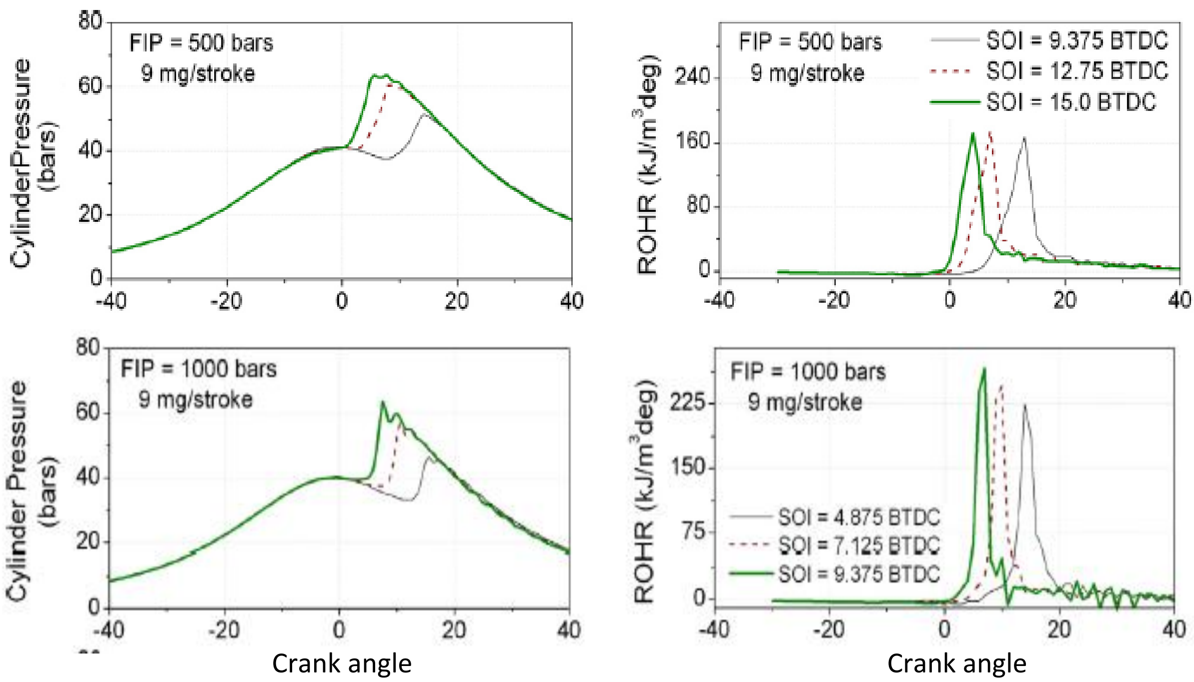

Figure 5: Effect of fuel injection parameters on combustion [6]

Figure 5 shows the variation in cylinder pressure and ROHR w.r.t. crank angle at different injection pressures and start of injection timings.

The peak cylinder pressure increases with increasing load due to increase in fuel quantity injected. This happens due to richer mixture formed inside the chamber, which burns more rapidly in early stages of combustion (premixed combustion) and remaining fuel burns in later stages (diffusion combustion). ROHR curve also follows similar pattern.

Advancing injection timing causes longer ignition delay, which leads to premixed combustion and higher peak cylinder pressure and higher ROHR can be seen. When injection timing was retarded and came closer to top dead center (TDC) in compression stroke, ignition delay became shorter, which caused higher fuel fraction burning in diffusion combustion thus lowering peak cylinder pressure. Due to this shorter ignition delay, pressure peak was smaller and it also shifted away from TDC position in expansion stroke as compared to earlier SOI conditions. This shift was clearly visible in ROHR curves where the peak of the curve shifted away from TDC in expansion stroke with retarded injection. Higher fuel injection pressure (1000 bars) gave extremely high ROHR. This was due to finer fuel atomization at higher FIPs, which promoted mixing and increases ROHR. However, knocking conditions can be noticed especially at higher engine loads. This knocking tendency increased with advanced injections due to availability of more fuel quantity in early stages of combustion, which promoted erratic combustion due to extremely high ROHR. **[6]**

Another study by Agarwal et al. **[7]** also reported similar results for effect of fuel injection pressure and injection timings of Karanja-biodiesel blends on combustion characteristics.

Figure 6 shows the effect of fuel injection parameters and load on the performance characteristics of the engine in terms of exhaust gas temperature (EGT), specific fuel consumption (BSFC) and thermal efficiency (BTE). It can be seen that, EGT increased with increasing engine load because of the increased fuel quantity injected.

However higher exhaust gas temperatures were seen for lower FIP (500 bar) compared to higher FIP (1000 bar). It happened due to larger droplet size distribution inside the combustion chamber, which led to more fuel burning in diffusion combustion,

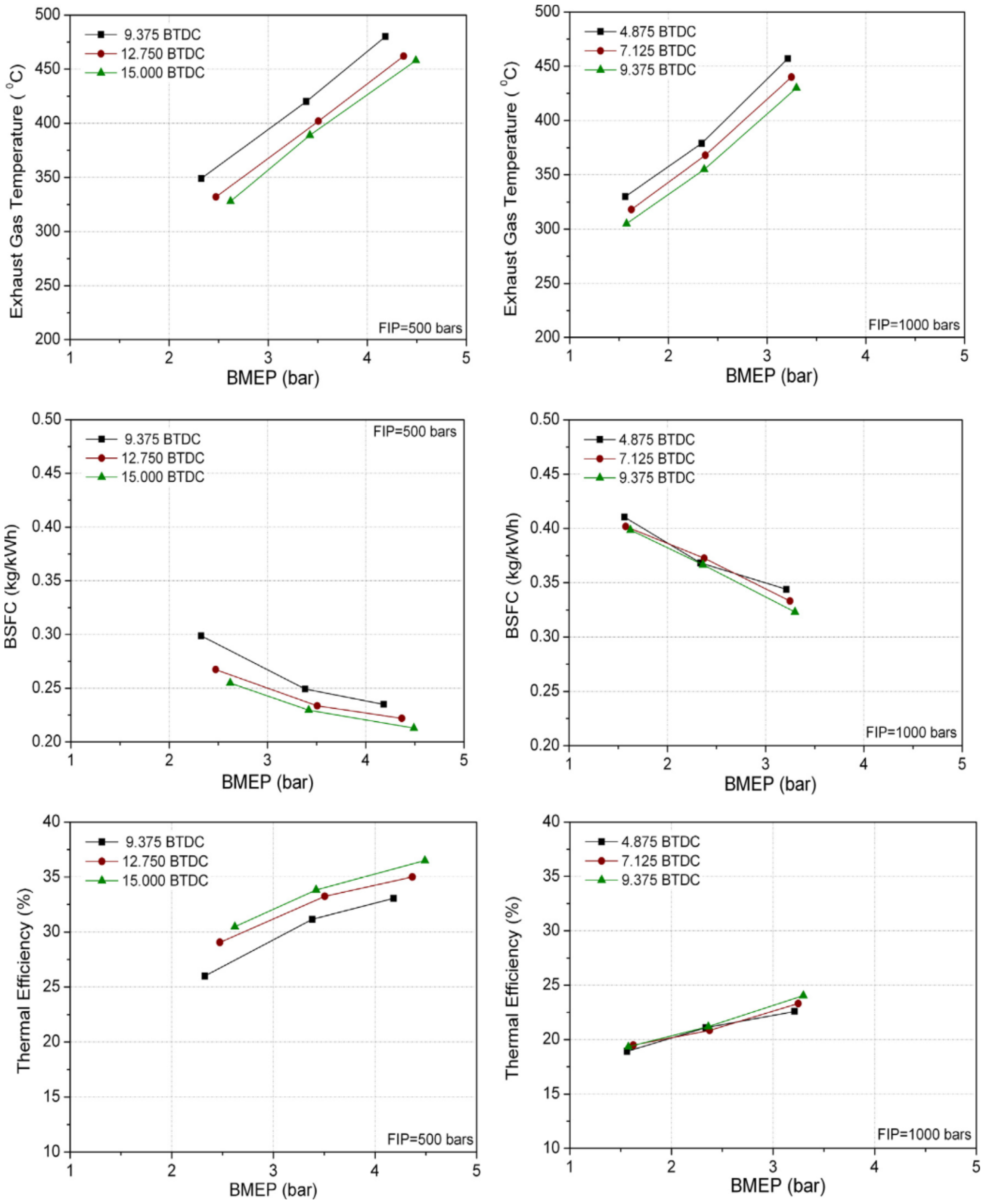

Figure 6: Effect of fuel injection parameters on performance [6]

while for higher FIP, finer droplet size distribution resulted in relatively better fuel–air mixing and smoother combustion. EGT was found to be lower for advanced injection timings due to main heat release occurring closer to TDC in the expansion stroke, which provided enough time for hot gases to expand and cool down before the exhaust valves were opened. Similarly, brake specific fuel consumption (BSFC) decreased with increasing engine load. This reduction in BSFC can be explained by the fact that as the engine load increases, there was continuous improvement in combustion quality and efficiency. Cylinder pressure increased with increasing engine load and increasing injected fuel quantity, which burned more efficiently therefore fuel consumption per unit brake power produced i.e. BSFC decreased. At very high FIP (1000 bars), BSFC was higher than lower FIP (500 bar) due to relatively inferior combustion characteristics, which led to lower power output at 1000 bars FIP. Brake thermal efficiency (BTE) followed exactly reverse trend as that of BSFC and increased with increasing engine load. BTE was seen to be significantly lower at very high fuel injection pressures. It happened because the combustion got into knocking regime. Shorter ignition delay led to knocking and fluctuations in cylinder pressure and temperature and shorter combustion duration. The results also showed that BTE increased with advanced SOI due to similar reasons as BSFC. **[6]**

Another experimental investigations by other researchers **[7], [12]–[14]** for studying the effect of fuel injection pressure on performance of diesel engine also demonstrated similar results.

Figure 7 demonstrates the brake specific emission characteristics of exhaust gases CO, HC and NOx for different engine loads at varying injection parameters.

BSCO and BSHC emissions both follow the same trend. BSCO and BSHC emissions also decreased with increasing engine load. It can be explained as, higher cylinder gas temperatures at higher engine loads, which led to more efficient combustion of fuel at higher temperature, producing lower quantities of these

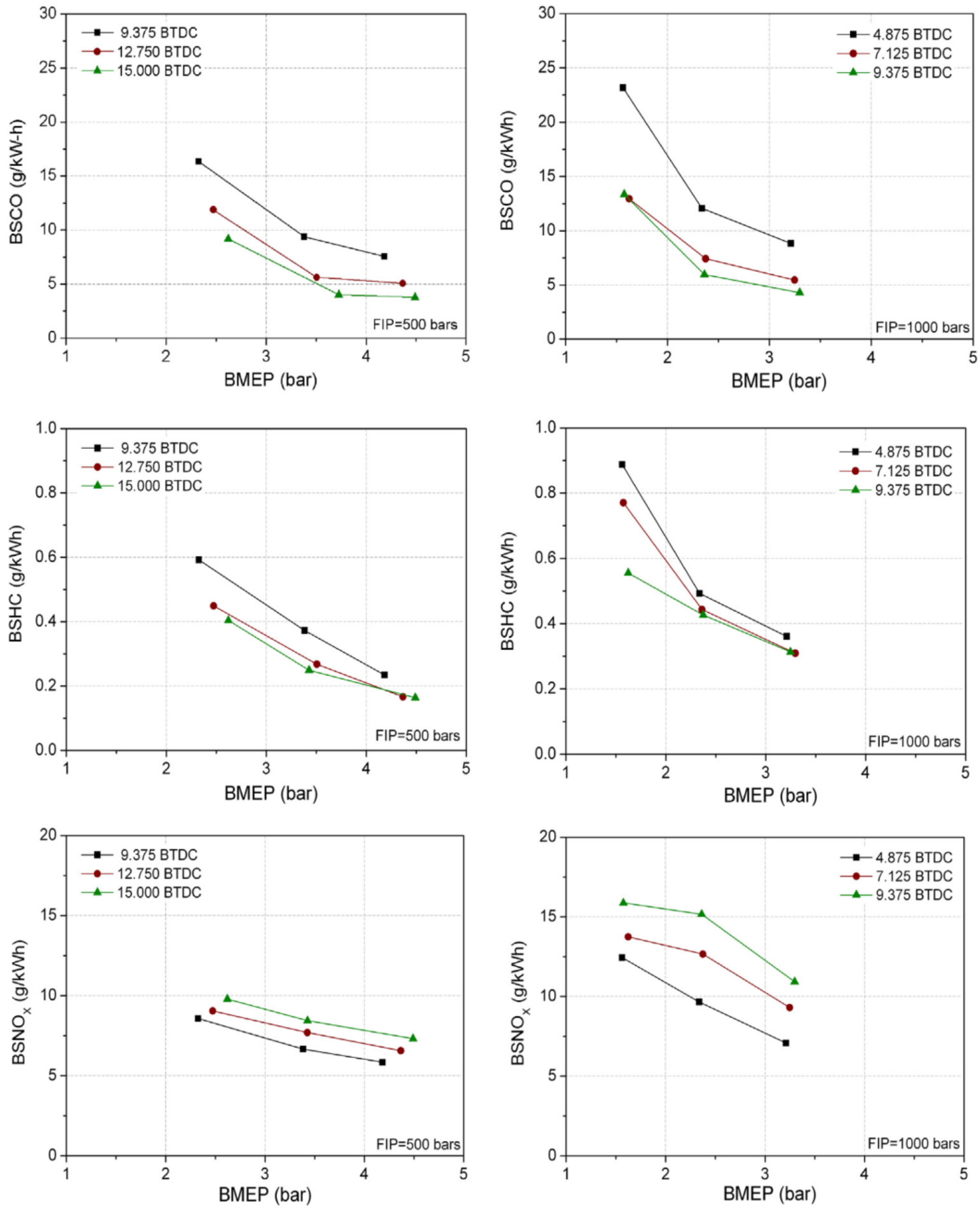

Figure 7: Effect of fuel injection parameters on emission [6]

emissions. At similar BMEP, CO and HC mass emissions decreased with increasing FIP due to superior fuel–air mixing in the combustion chamber. Advanced ignition timing improved air–fuel mixing due to availability of more time for mixing process; therefore, this led to lower CO and HC mass emissions as well. Formation of NOx is highly dependent on the maximum temperature of the burning gases, oxygen content, and residence time available for the reactions to take place at these extreme conditions.

Figure 7 shows that mass emission of NOx decreased with increasing engine load due to relatively higher increase in power output. BSNOx emissions increased significantly with increasing FIP due to higher ROHR during premixed combustion phase. Similarly, advanced ignition timing also increased BSNOx emissions due to higher ignition delay and more time available for NOx chemistry to take place, which ultimately increased the BSNOx emissions.

Similar findings for effect of fuel injection pressure on exhaust emissions of diesel engine were shown by other researchers **[7], [11], [12], [14]** in their investigations.

2.5.2 Effect of compression ratio

Jindal et al. [8] studied the effect of compression ratio and fuel injection pressure in a direct injection diesel engine running Jatropha methyl ester (JME) and diesel as fuels each at three different fuel injection pressures (150, 200 and 250 bars) and three different compression ratios (16, 17 and 18).

Figure 8 shows the effect of compression ratio as well as fuel injection pressure on the performance of the engine in terms of BSFC and BTE operated on diesel as well as JME. The standard rated parameters for diesel fuel are 210 bar injection pressure and 17.5 compression ratio. It can be observed that, BSFC decreases with load significantly for both fuels. However, BSFC is greater for JME than diesel by about 25-35% at standard rated parameters. This is due to lower heat value of esters of vegetable oils than diesel. Hence, more biodiesel is needed to maintain the same power output. BSFC decreases as compression ratio increases. Compression ratio 18 gives lowest BSFC as at higher compression ratios, brake power increases.

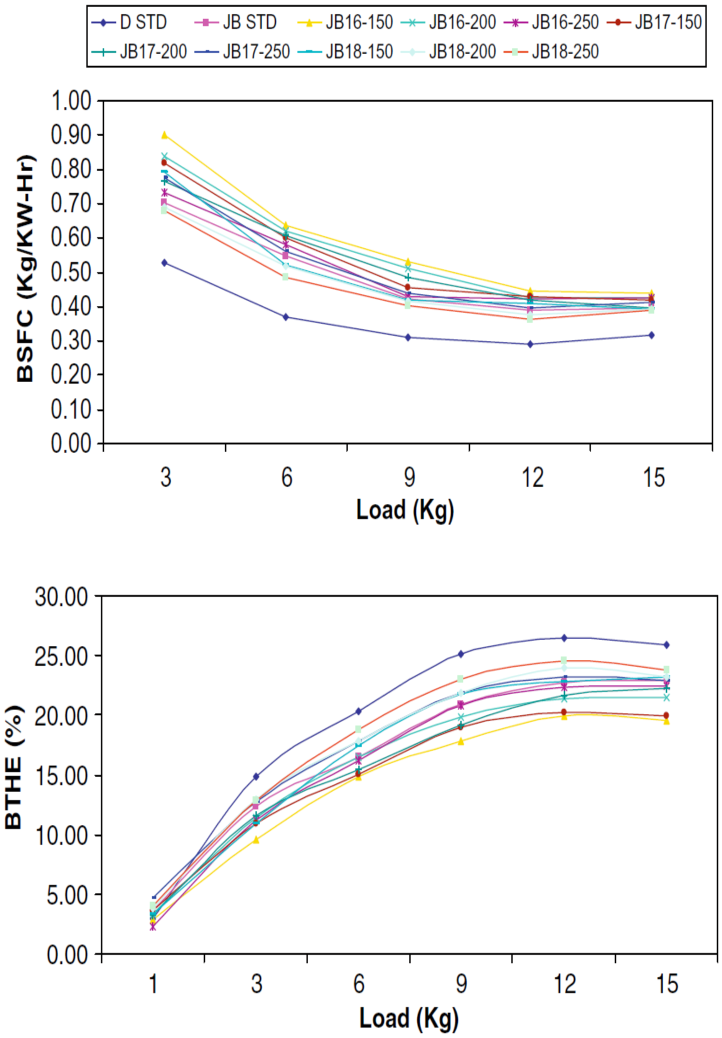

Figure 8: Effect of CR and FIP on performance of diesel and JME [8]

The effect of change in injection pressure on BSFC is also found significant with lowest BSFC at 200 bar up to 50% load and at 250 bar for higher loads. The decrease in BSFC can be attributed to the more efficient utilization of the fuel at higher injection pressures because of better atomization. The BSFC was lowest at 250 IP while maintaining the compression ratio as 18 i.e. high FIP and high CR.

Similar effects can be observed in case of brake thermal efficiency (BTE), since for higher injection pressure of 250 bar because of better combustion due to finer breakup of fuel droplets providing more surface area and better mixing with air.

It was observed that, HC (ppm), oxides of nitrogen NOx (ppm) and exhaust temperature of the exhaust are found lower for JME as compared to Diesel at standard settings of the engine whereas carbon dioxide CO_2 (%) and carbon monoxide CO (%) emissions are higher. HC emissions tend to increase with increase in compression ratio and also on reduction in injection pressure. At lower compression ratio, insufficient heat of compression delays ignition whereas at high compression ratio, dilution by residual gases affects the combustion. With increase in compression ratio, CO emission decreases whereas with increase in injection pressure for a given compression ratio, CO increases. Higher levels of HC emission explain the reduction in CO with higher compression ratio. With higher injection pressure, increase in CO is due to poor diffusion flame combustion. At lower compression ratio, the temperature reached is also low and thus more CO is exhausted from engine. The emission of NOx is more sensitive to compression ratio at lower injection pressures as compared to higher compression ratios. With low injection pressure, increase in compression ratio facilitates the combustion of larger droplets because of high temperature of compression, whereas, high injection pressure breaks the fuel into smaller droplets with good combustion suppressing the effect of compression pressure. The highest NOx emissions were observed for high compression ratio and low injection pressure which is due to greater availability of fuel inside the cylinder and higher peak pressure leading to high temperatures.

Similar findings were reported by Raheman and Ghadge **[9]** and Sayin and Gumus **[10]** in their investigations for studying effect of compression ratio and injection parameters on the performance and emission characteristics of a direct injection diesel engine fueled with biodiesel-blended diesel fuel.

Chapter 3

Experimentation

3.1 Experimental setup

A twin cylinder optical access research engine (Legion brothers') shown in Figure 9 is used for this study. Out of two cylinders, one is working cylinder in which combustion occurs continuously, called thermodynamic cylinder and other cylinder is can be accessed optically in which combustion occurs whenever triggered. This study does not consist of any optical diagnosis of the combustion process, so the optical accessed cylinder is cut off throughout the experimentation. The combustion chamber geometry is a toroidal bowl in a piston top, ensuring fast burning and compact combustion chamber. The engine is equipped with two overhead camshafts driving 4 inlet valves (2 for each cylinder) and 4 exhaust valves.

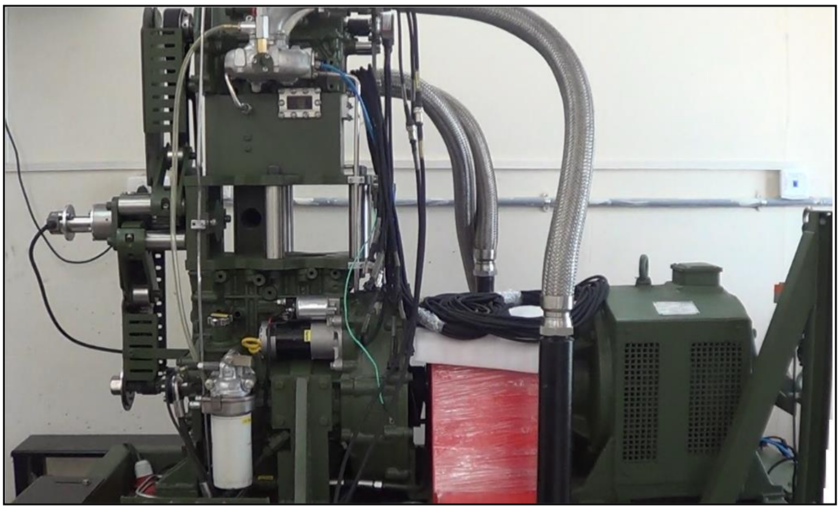

Figure 9: Optical access research engine

The specifications and valve timings of the engine are given in Table 1.

No of running cylinders	1 out of 2
Stroke (mm)	100
Bore (mm)	94
Connecting rod length (mm)	235
Compression ratio	6.5:1-18:1
Speed range (rpm)	1000-1200
Inlet open (degree)	5 ATDC
Inlet close (degree)	21 ABDC
Exhaust open (degree)	25 BBDC
Exhaust close (degree)	9 BTDC
Injection system	CRDi
Injection pressure (bar)	300, 500 and 800 bars
Injection timing (degree)	9 BTDC

Table 1: Specifications of research engine

An electronically controlled throttle body is used to control flow of air to the engine. It is mounted on the inlet manifold. The air flow rate is measured by an air box instrument, wherein, air from a large volume box passes through the orifice plate and the pressure drop across the orifice is measured. This pressure drop signal is fed to the ECU to calculate the accurate air flow rate. Rotameters are provided for cooling water and calorimeter water flow measurement.

The fuel flow rate is measured by an automatic volumetric fuel flow meter. It consists of two sensors, one at the bottom and another at the top of a 100 ml measuring burette. The fuel is made to pass through this burette and time required for emptying the burette is recorded and fed to the ECU. The ECU then calculates the mass flow of fuel based on density of fuel fed to it manually. The fuel properties are mentioned in Table 2.

Fuel property	**Value**
Name	Standard diesel
Cetane rating	CN 48
Density (kg/m^3)	860
Calorific value (kJ/kg)	42500

Table 2: Properties of a fuel

The in-cylinder pressure is measured using a piezoelectric pressure transducer. It is fitted to the cylinder head, receiving gas pressure through a passage drilled in the head, up to the center of the cylinder head, opened to the combustion chamber. An eddy current dynamometer is directly coupled to the engine's crankshaft to apply and measure the load on the engine. Its load range varies from 0 kg to 10 kg.

The engine is equipped with CRDi injection system. It consists of CRDi driver module and CRDi kit, which control injection pressure, injection timing and duration. The schematic diagram of CRDi system is shown in Figure 10. The CRDi kit and driver are shown in Figure 11.

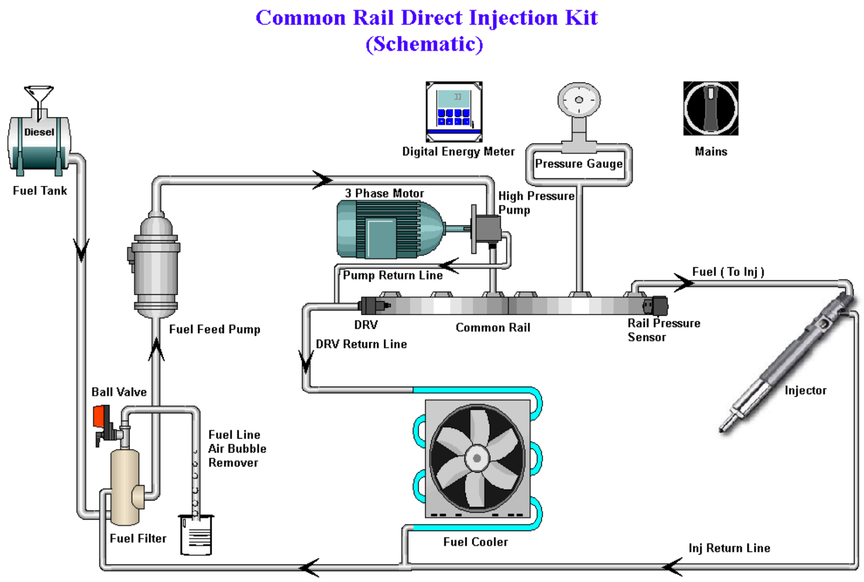

Figure 10: Schematic diagram for CRDi kit

Figure 11: CRDi kit and CRDi driver

The acquisition software is Legion brother's software. This system allows real-time, on screen display of recorded parameters such as in-cylinder pressure, exhaust gas temperature, temperatures of cooling water to the engine and the calorimeter. It also displays calculated parameters such as air-fuel ratio, volumetric and brake thermal efficiency. For every test point, pressure data is recorded for 100 consecutive cycles and averaged. The parameters like heat release rate, peak pressure etc. are calculated for each cycle and then averaged. A screenshot of the software user interface is shown in Figure 12.

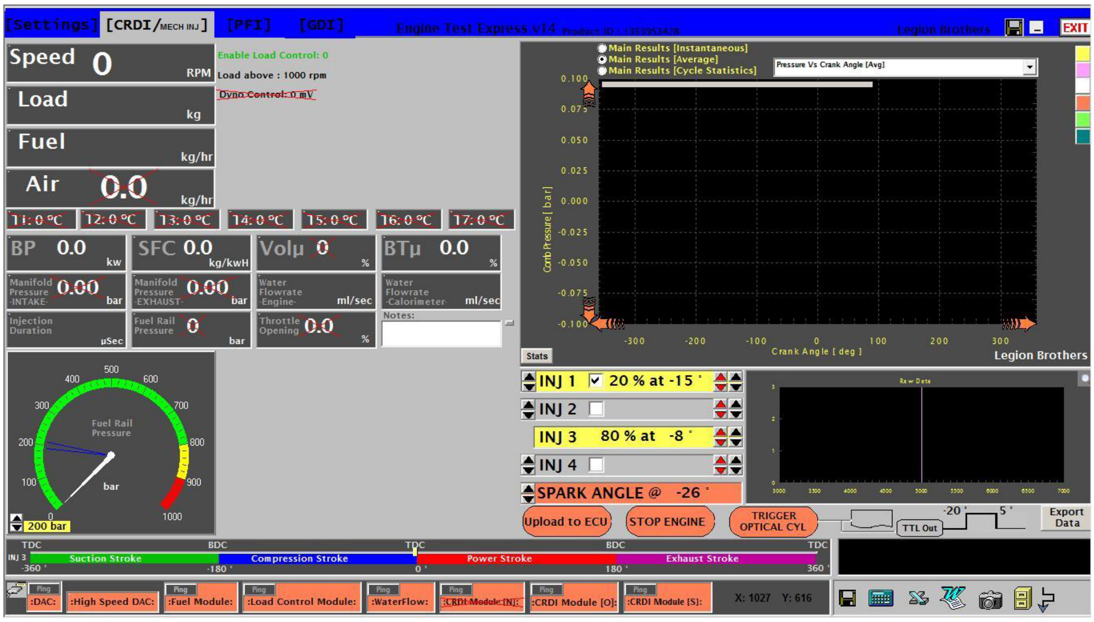

Figure 12: Screenshot of software user interface

An exhaust gas analyzer (AVL DiGas 444G) is used for exhaust emissions measurement. The exhaust from the engine is fed to the analyzer through a series of filters. The analyzer gives the emission values of CO (% vol), HC (ppm hex), CO_2 (%vol), O_2 (% vol) and NOx (ppm vol).

3.2 Experimental procedure

The focus of this investigation is to find out the operation envelope of the research diesel engine equipped with CRDi injection system; and to study the effect of compression ratio and fuel injection pressure on overall engine performance.

For this study, the compression ratio of the engine is varied between 16 and 18. The fuel injection pressure (FIP) is varied as 300, 500 and 800 bars. The engine speed is maintained constant at 1050±30 RPM at different loads i.e. 4, 6 and 8 kg. Where 4 kg is considered to be low load, 6 kg as part load and 8 kg as full load condition. The engine pressure data was recorded for each of the combination of above parameters using high speed data acquisition system. The engine is highly susceptible to unstable operation at 2 kg and 10 kg load. So experiments are conducted on loads varying from 4 kg to 8 kg only.

For each test condition, the engine was run for at least 3 minutes after which data were collected. The pressure data is recorded for 100 consecutive engine cycles and then averaged. Each experiment was carried out three times for repeatability purpose.

The in-cylinder pressure measurement device used does not measure pressures precisely and accurately during intake and exhaust processes. The effect of this error in pressure measurement on each test result is considered identical for each test, since the study is being carried out relatively.

For all test conditions, the emission values were recorded thrice and a mean of these was taken for comparison. The performance of the engine at different loads and settings was evaluated in terms of brake specific fuel consumption (BSFC), brake thermal efficiency (BTE) and emissions of carbon monoxide (CO), carbon dioxide (CO_2), un-burnt hydrocarbon (HC) and oxides of nitrogen (NOx) with exhaust gas temperature.

The BSFC is evaluated by the software on the basis of fuel flow and brake power developed by the engine using the expression,

$$BSFC = \frac{Volumetric\ fuel\ flow\ rate * Fuel\ density}{Brake\ power}\ kg/kWhr$$

Similarly, BTE is also evaluated by software using the expression,

$$BTHE = \frac{Brake\ power * 3600 * 100}{Volumetric\ fuel\ flow\ rate * Fuel\ density * Calorific\ value\ of\ fuel}\%$$

The load values are represented as corresponding BMEP values in graphs. BMEP values are evaluated using the expression,

$$BMEP = \frac{Brake\ power * 60000}{L * A * N/2}\ bar$$

Load (Dynamometer)	BMEP (bar)
4 kg	1.72
6 kg	2.65
8 kg	3..28

Table 3: Load conversion in BMEP

Chapter 4

Results and discussion

4.1 Experimental results

Experiments are carried out at different loads (4, 6 and 8 kg) for a variable compression ratio (16 and 18) and variable fuel injection pressures (300, 500 and 800 bars). Cylinder pressure data is collected for 100 consecutive cycles and averaged. Each experiment is performed three times for repeatability. For plotting p- θ and HRR- θ graphs, average of the three readings is taken for each point. Similarly, emission data (CO, CO_2, HC, O_2 and NOx) as well as performance data (BSFC, BTE and EGT) are recorded thrice for every test condition; and for plotting graphs, average of the three readings is taken for each point and the standard deviation for each point are plotted as error bars.

The relationships between independent variables (compression ratio and injection pressure, load) and dependent variables (BSFC, BTHE and emissions) are shown in the figures and the results are discussed in the following sections.

4.2 Combustion analysis

Cylinder pressure data analysis is the most effective tool to analyze engine combustion behavior because cylinder pressure history directly influences power output, combustion characteristics and engine emissions. The effect of compression ratio and fuel injection pressure on the peak cylinder pressure as well as peak rate of heat release are investigated at variable loading conditions.

4.2.1 Effect of compression ratio

The effect of compression ratio on peak cylinder pressure and heat release rate (HRR) is depicted in Figure 13: Effect of CR on combustionFigure 13 with FIP kept constant at 500 bars.

It can be observed from the curves that, peak cylinder pressure increases with increasing load. It happens due to with increasing load, quantity of fuel injected per cycle also increases for maintaining same engine speed. So, richer mixture is made within the burning chamber, which burns more rapidly in premixed combustion stage and increases peak cylinder pressure. HRR curves also follow similar trend, i.e. peak HRR increases with increasing load.

As the compression ratio is increased from 16 to 18, peak cylinder pressure also increases in every case. It can be explained as, when the compression ratio is increased, cylinder pressure and temperature rises to a higher value and helps in shortening ignition delay. This promotes premixed combustion, and more amount of fuel the is burned in early stages of combustion. Hence, peak cylinder pressure increases with increasing compression ratio and shifts towards TDC in expansion stroke. Similar conclusion can be made for HRR curves as well. As the compression ratio is increased, the ignition delay is lowered and the combustion process is started earlier than the lower compression ratio. Since more amount of fuel is burned in premixed combustion stage than diffusion combustion stage, the peak HRR in case of higher CR is lesser than that in lower CR condition. The peak HRR in case of lower CR is more and shifted away from TDC in expansion stroke.

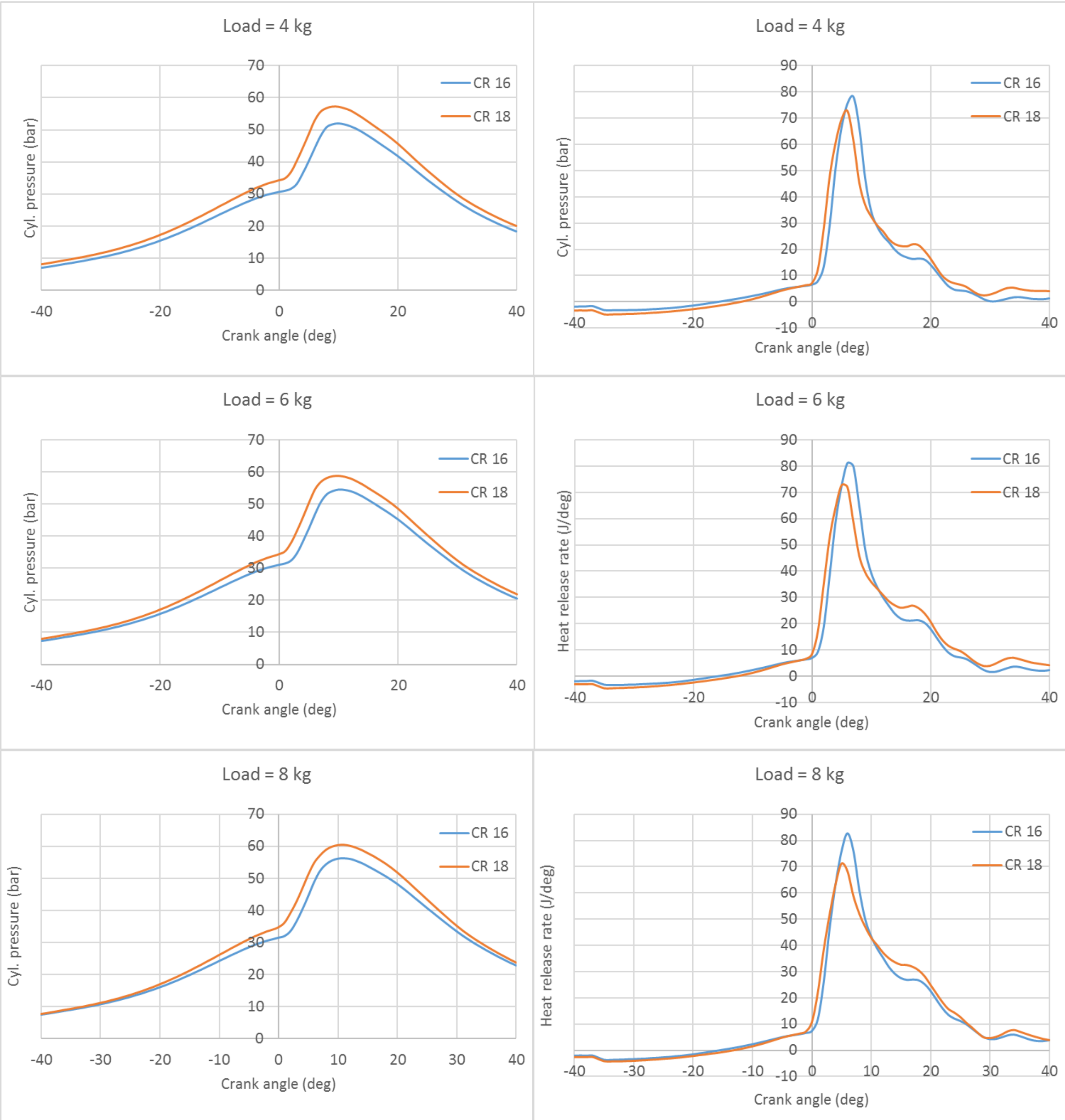

Figure 13: Effect of CR on combustion

4.2.2 Effect of injection pressure

Effect of fuel injection pressure on peak cylinder pressure and rate of heat release (HRR) is shown in Figure 14 with CR kept constant at 18.

As stated earlier, peak cylinder pressure as well as peak HRR increases with increasing load. It can also be observed that, peak cylinder pressure and peak HRR are increased as fuel injection pressure is increases from 300 bars to 500 bars and so on. Maximum cylinder pressure and HRR are attained for 800 bars FIP.

It can be explained as; as the fuel injection pressure is increased, the combustion quality is improved because of finer breakup of fuel droplets due to high pressure which provides more surface area resulting in better fuel-air mixing and fuel evaporation. More the injection pressure, finer are the fuel droplets and fuel-air mixing is better. This promotes premixed combustion. Since, the fuel droplets are smaller, the ignition delay gets shorter for higher FIPs. Thus, the peak HRR as well as peak cylinder pressure is higher and shifts towards TDC in expansion stroke for high injection pressures.

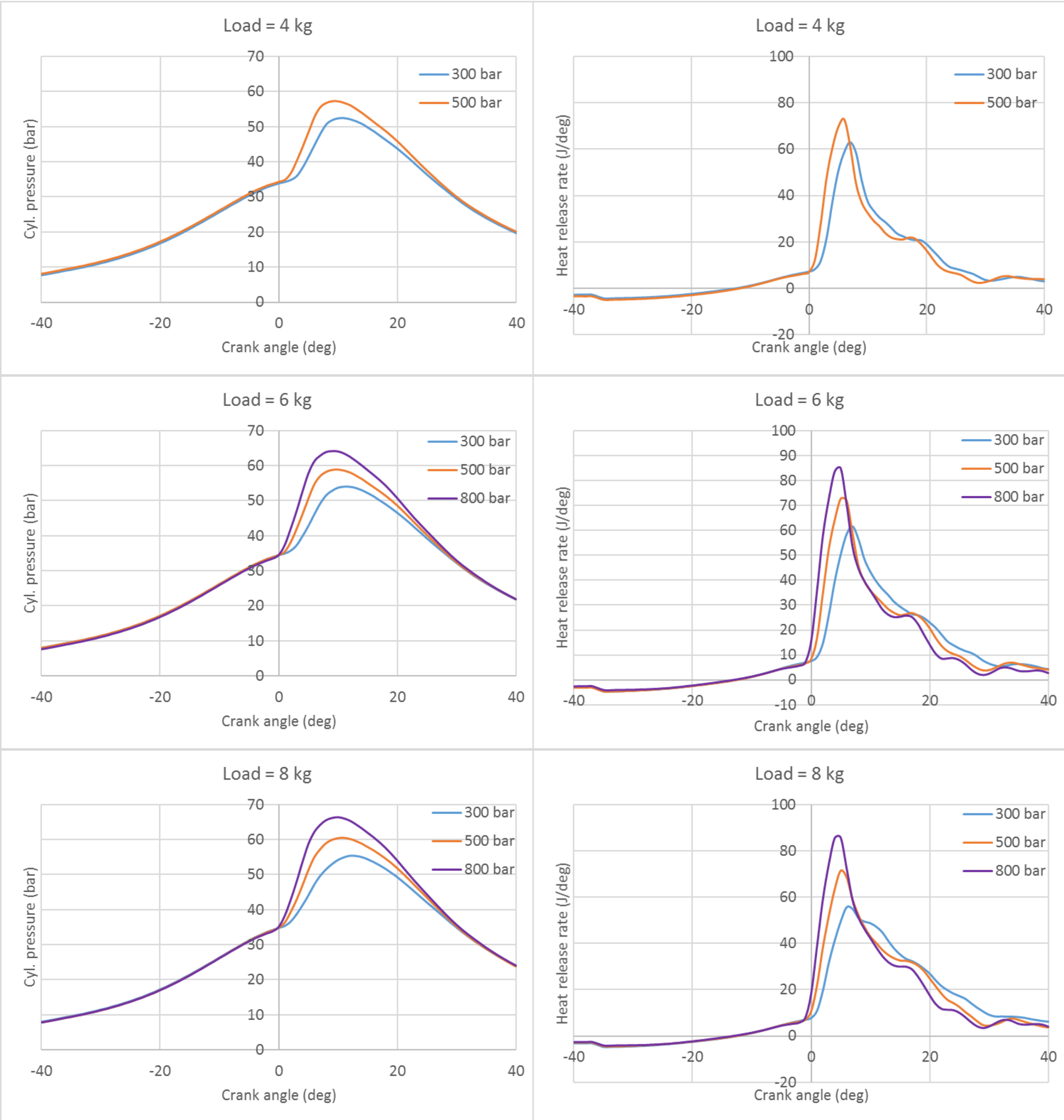

Figure 14: Effect of FIP on combustion

4.3 Performance analysis

The performance characteristics of the engine are represented in terms of brake specific fuel consumption (BSFC), brake thermal efficiency (BTE) and exhaust gas temperature (EGT). Following figures demonstrate the simultaneous effect of compression ratio (16 and 18) and fuel injection pressure (300, 500 and 800 bars) on performance characteristics.

Figure 15 shows effect of CR and FIP on BSFC at variable loads represented as BMEP. It can be observed that, BSFC decreases with increase in load for all cases. It can be explained as; with increasing load, there was improvement in combustion quality and efficiency. Cylinder pressure increases with increasing load and increasing injected fuel quantity. This results in more power output from the engine. Therefore, fuel consumption per unit brake power produced, i.e. BSFC decreases. Similar conclusion can be made for BTE from Figure 16. As load increases, the fuel burns more efficiently. As a result of this, brake power also increases and results in better brake thermal efficiency. However, the effect of increasing FIP was not much significant at higher CR.

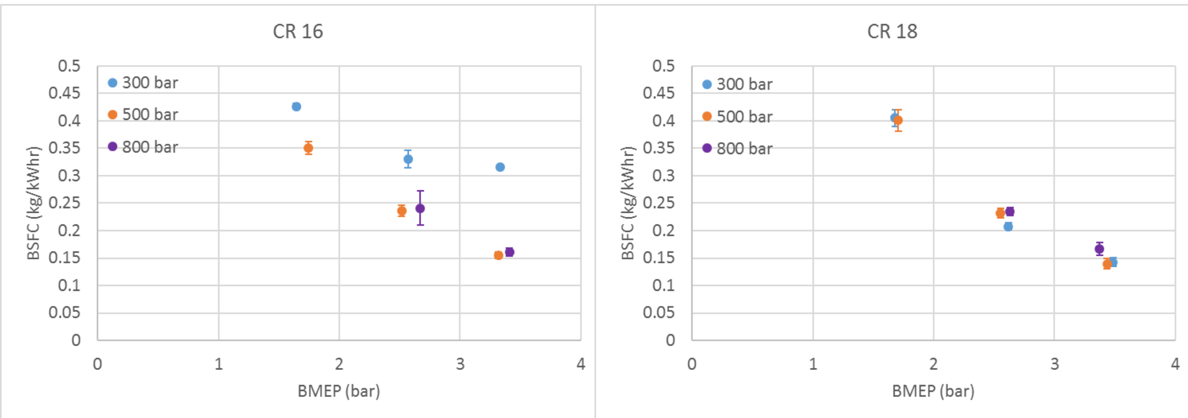

Figure 15: Effect of CR and FIP on BSFC

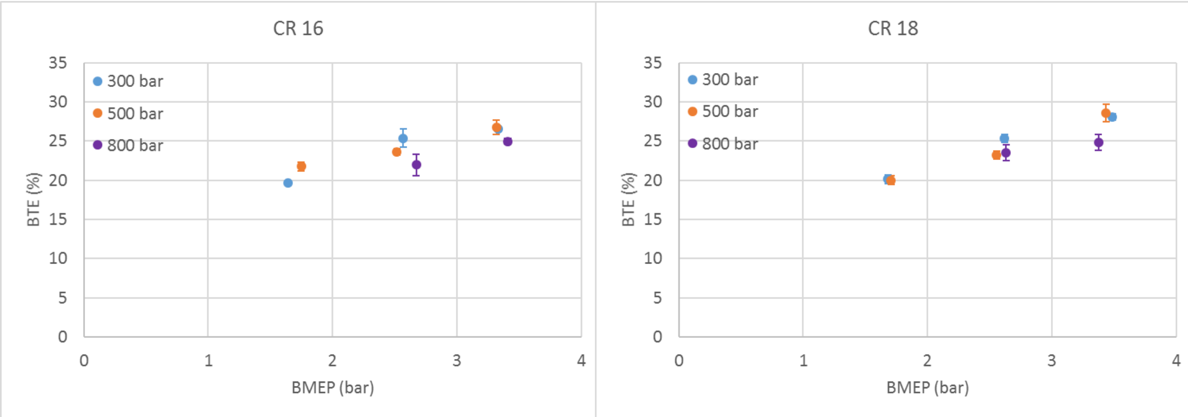

Figure 16: Effect of CR and FIP on BTE

BSFC decreases while BTE increases significantly with increase in CR at similar BMEP. At higher compression ratio, brake power increases because of higher cylinder pressures. Hence, BSFC decreases as well as BTE increases with increase in CR. The effect of increasing FIP is also found significant on BSFC and BTE at similar BMEP. BSFC decreases while BTE increases with increase in FIP. This can be attributed to the more efficient utilization of fuel at higher injection pressure because of better atomization which enhanced fuel-air mixing and resulted in higher cylinder pressures and consequently higher brake power output.

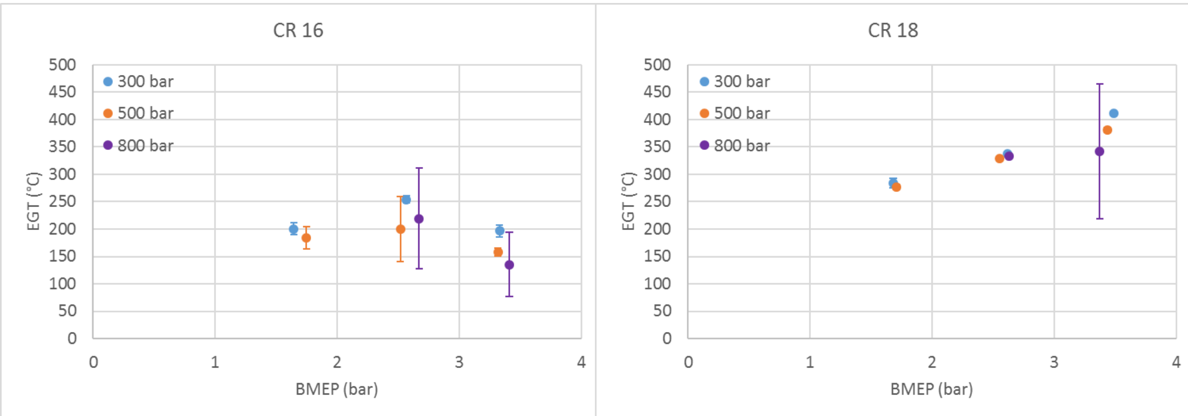

Figure 17: Effect of CR and FIP on EGT

Figure 17 shows that the EGT increases with increasing load (BMEP) because of the increased fuel quantity injected. Also, EGT increases with increase in CR at similar BMEP because, the cylinder pressure and temperature are already higher prior to combustion for high compression ratios. However, EGT decreased with increasing FIP at similar BMEP. It happened due to larger droplet size distribution inside the combustion chamber, which promoted heterogeneous combustion, while finer droplet size distribution at higher FIP gave relatively better fuel–air mixing and smoother combustion. Also, for higher FIP, peak heat release rate (HRR) was shifted towards TDC in expansion stroke. Thus, it allowed for the exhaust gases to cool down significantly during expansion stroke.

4.4 Emission analysis

The exhaust emission characteristics of the engine are represented in terms of emissions of carbon monoxide (CO), carbon dioxide (CO_2), unburnt hydrocarbons (HC), oxygen (O_2) and oxides of nitrogen (NOx). Following figures demonstrate the simultaneous effect of compression ratio (16 and 18) and fuel injection pressure (300, 500 and 800 bars) on these emission characteristics.

CO and HC emissions both follow the same trend in Figure 18 and Figure 19. CO and HC emissions decreased with increasing engine load. It happened due to higher cylinder gas temperatures at higher engine loads, which led to more efficient combustion of fuel at higher temperature, producing lower quantities of these emissions. At similar BMEP, CO and HC emissions decreased with increasing CR. It can be attributed to higher cylinder temperatures for higher CR leading to more efficient combustion of fuel. As well as, increasing FIP at similar BMEP also reduces CO and HC emissions. This is due to superior fuel–air mixing in the combustion chamber at high injection pressures which leads to more complete combustion of the fuel.

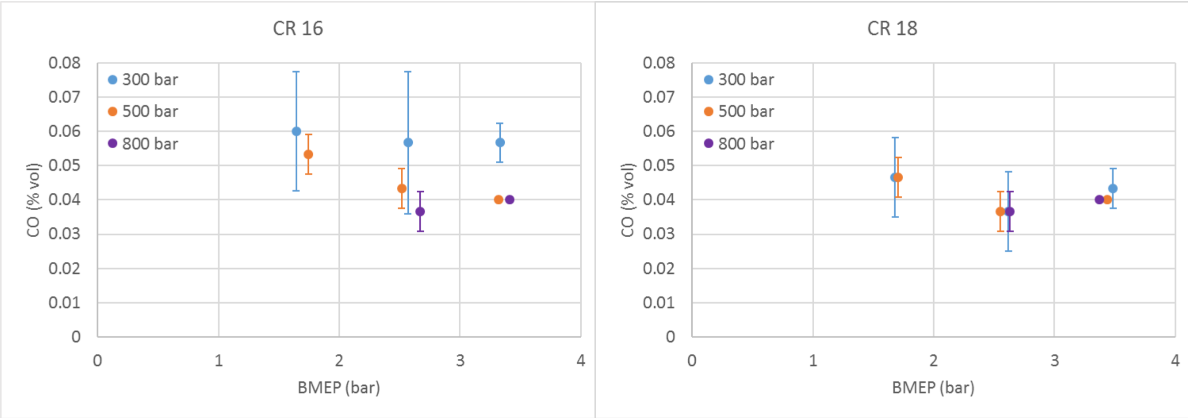

Figure 18: Effect of CR and FIP on CO emission

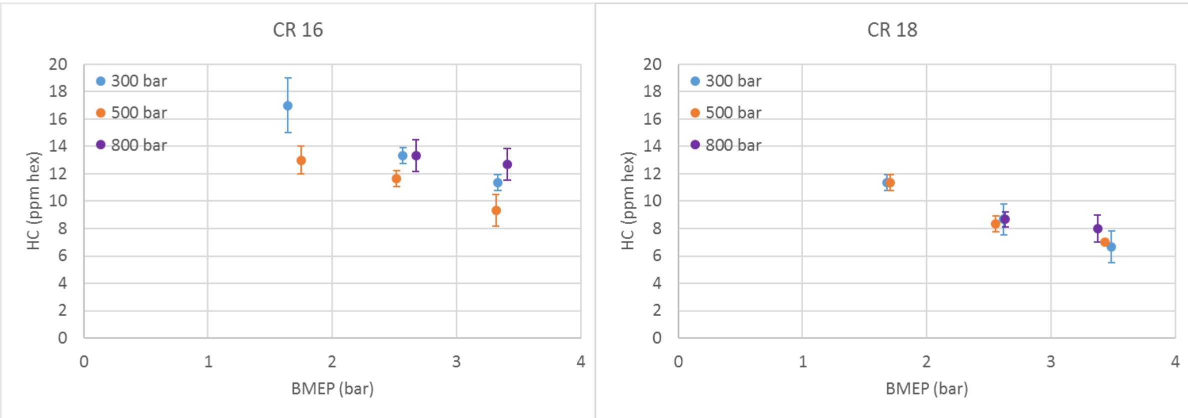

Figure 19: Effect of CR and FIP on HC emission

Figure 20 and Figure 21 show emissions of CO_2 and O_2 w.r.t. load (BMEP) respectively. CO_2 emission decreased whereas O_2 emission increased with increasing loads which indicated towards more complete combustion of the fuel. Because, CO_2 gas emission is a product of complete combustion when hydrocarbons react with O_2 molecules in the air. Similarly, CO_2 emission increased while O_2 emission increased with increase in compression ratio as well as increase in FIP. This can be attributed to better utilization of fuel and efficient combustion at higher loads as well as higher CR and FIP as stated earlier.

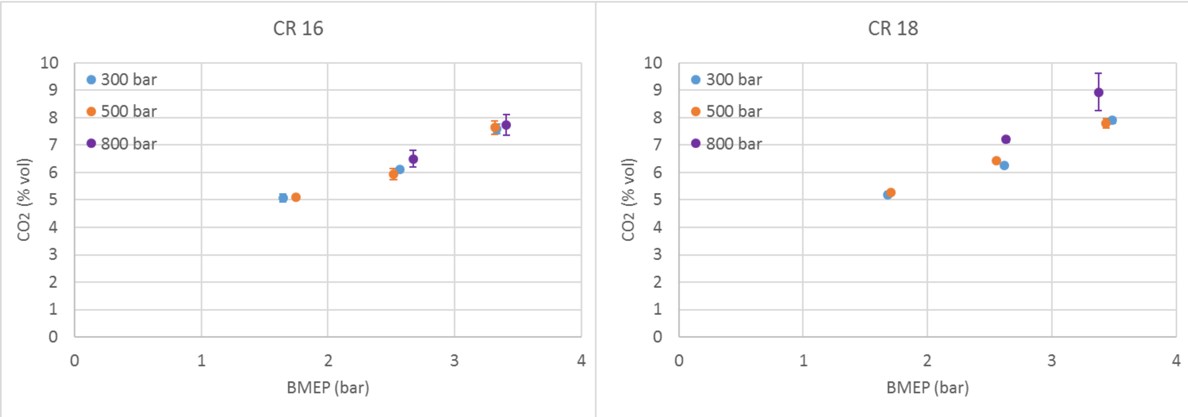

Figure 20: Effect of CR and FIP on CO_2 emission

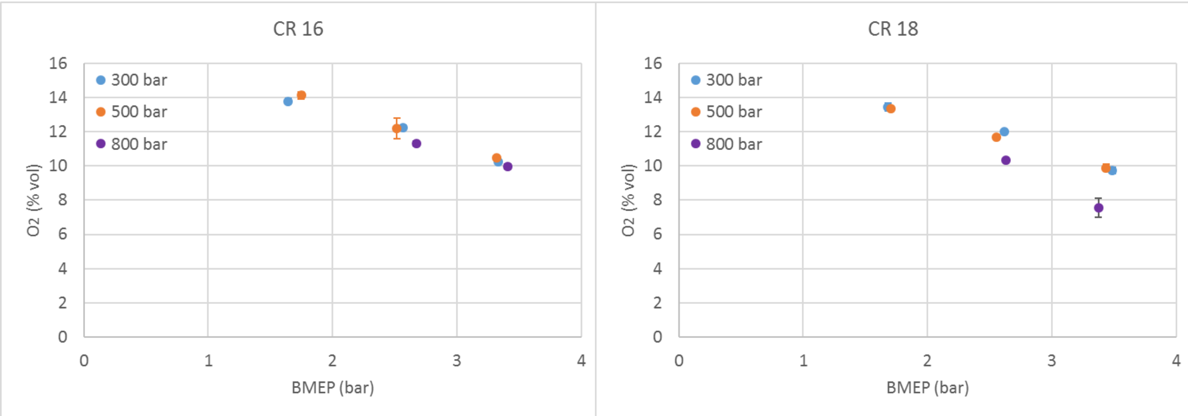

Figure 21: Effect of CR and FIP on O_2 emission

Figure 22 demonstrates the effect of compression ratio and FIP on NOx emission w.r.t load (BMEP). Formation of NOx is highly dependent on the maximum temperature of the cylinder gases, oxygen content, and residence time available for the reactions to take place at these extreme conditions. It can be observed that, that emission of NOx increased with increasing engine load due to relatively higher cylinder temperatures because of more fuel quantity injected. NOx emissions also increased significantly with increasing FIP due to higher peak HRR during premixed combustion stage for higher injection pressures. Increase in CR also caused increase in NOx emission due to higher cylinder temperatures at higher compression ratios.

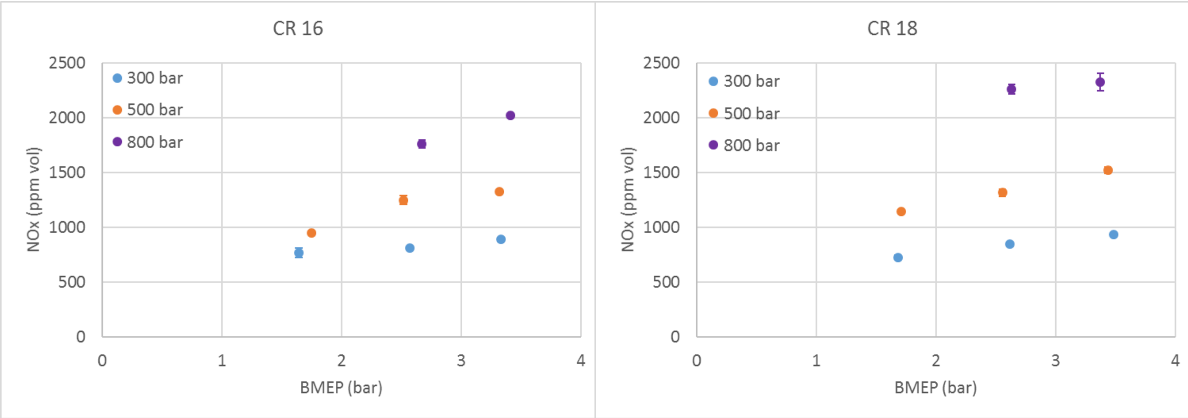

Figure 22: Effect of CR and FIP on NOx emission

Chapter 5

Conclusion

Experiments have been carried out on the research diesel engine equipped with CRDi injection system to find its operational envelope. For investigating effect of compression ratio and fuel injection parameters on the combustion, performance and emission characteristics of the engine, experiments were conducted at constant engine speed with two compression ratios (16 and 18) and three injection pressures (300, 500 and 800 bars) at variable engine loads.

The experimental results were studied and post-processing was done. The obtained result trends were verified with the literature. The combustion characteristics i.e. peak cylinder pressure and peak HRR were found to be increasing with increasing load as well as increasing injection pressure, whereas peak cylinder pressure increased and peak HRR decreased for increasing compression ratio. The performance characteristics BSFC and BTE improved with increasing FIP as well as CR. However, effect of increasing FIP was not significant at higher CR. The emissions of CO and HC were reduced whereas the NOx emission increased with increasing CR as well as FIP.

Thus, with increase in compression ratio and fuel injection pressure, combustion and performance characteristics improved significantly while simultaneously reducing emissions up to some extent.

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