“STUDIES ON SINGLE CYLINDER CRDI DIESEL ENGINE USING RETARDED AND SPLIT INJECTION STRATEGY”

A project report submitted in partial fulfilment of the requirement for the award of the degree of

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ABSTRACT

Good fuel economy and high thermal efficiency of direct injection diesel engines are truly welcome characteristics from the viewpoints of preserving energy sources and suppressing global warming. Despite the attractive fuel economy, high emissions of oxides of nitrogen (NOx) and particulate matter (PM) with their unresolved trade-off are a major challenge to be addressed through researchers. Downsizing of the engine, dilution using exhaust gases, retardation of injection timing, etc. are widely adopted techniques to lower NOx emissions. In the present days multiple or split injection strategy is gathering much attention from the researchers for its potential to effectively address NOx, soot. For changing the Fuel Injection Pressure, Fuel Injection Timing, Injection pulses existing single cylinder diesel engine fitted with conventional mechanical fuel injection system was suitably modified to operate on Common Rail Direct Injection (CRDI).The purpose of this study is to decrease the NO_X and smoke by applying retarded injections and various split-injection strategies. Experiments were carried out in a single cylinder CRDI research engine using diesel fuel at 1500 rpm, under 50% to 100% of full load brake power in multiple injection mode at 400 bar as constant fuel injection pressure (FIP) under varying start of pilot injection (SOPI) and start of main injection (SOMI) timings. First the experiment was conducted for retarded the fuel injection timing from 19^0bTDC to 7^0bTDC. Experimental results revealed that retarding the fuel injection timing enhanced the fuel economy by reducing the brake-specific fuel consumption (BSFC) mainly the BSFC is lower at high engine load and smoke and NO_X was reduced. The results at 13^0bTDC offered significant improvement in NOx and smoke emissions with considerable reduction in BTE. Second the experiment was conducted by constant SOPI at 30^0bTDC and retarded the SOMI from 19^0bTDC to 13^0bTDC.The best results was obtained at the 30^0bTDC (SOPI) and 13^0bTDC(SOMI) are considerable reduction in the NO_X and BTE is improved as compared with the single injection of 13^0bTDC.after that the experiment was done at advancing the SOPI from 30^0bTDC to 36^0bTDC and SOMI as constant at 13^0 bTDC was revealed no improvement.

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1. INTRODUCTION

1.1. INVENTION OF DIESEL ENGINE

The diesel engine, named after Rudolf diesel, is an internal combustion engine in which ignition of the fuel is caused due to mechanical compression, thus the diesel engine is also called as compression ignition engine. The invention of diesel engines goes way back all the way to the 1890's.

Fig 1.1. Diesel engine

In the year 1892, A German engineer Rudolf Diesel invented and patented an internal combustion engine known as Diesel engine. In the 1870's, steam was the main supplier of power for factories and trains. Rudolf diesel was a student learning about thermodynamics at that time, and he got the idea for creating an engine that would be highly efficient and convert heat energy generated into work. The first diesel engine prototype was built in 1893, though the first engine test was unsuccessful, he made necessary effects to build a diesel engine. However, in 1897, Diesel produced successful results after many improvements and tests. In February 1897, he was able to show an efficiency of 26.2% with the engine. Compared with the steam engine popular at that time, the engine Diesel had developed was more efficient by 16.2%. Rudolf diesel, who is best known for the invention of the diesel engine was born in

Paris, France in 1858. Diesel engine built by Langen & Wolf under licence in 1898 as shown in Fig1.1.

1.2. HISTORY OF DIESEL ENGINES

- **October 29th, 1897** Rudolf obtains a patent on supercharging the Diesel engine.
- **1899** First two-stroke Diesel engine was built by Hugo Guldner.
- **1903** Two first 'Diesel-powered' ships are launched.
- **1912** The first locomotive with a Diesel engine is used in Switzerland.
- **1913** Commercial ships and US Navy Submarines began to use the diesel engines.
- **1986** Bosch designs electronic diesel control for BMW 524tD.
- **1995** First common rail injection system in production truck, in Japan.
- **1998** BMW wins the 24 Hour Race at Nurburgring using a four cylinder diesel. Better fuel efficiency allowed the BMW to take less pit stops in winning the race.
- **2004** Piezo electric fuel injection technology introduced by Bosch.
- **2011** Chevrolet increases torque to 765 lb/ft and horsepower to 397 in its newest version of the Duramax Diesel 6600 by converting to piezo electric fuel injection and refining other features of the engine.

1.3. FOUR STROKE DIESELENGINE

Four stroke was first demonstrated by Nikolaus Otto in 1876, hence it is also known as Otto cycle. It consists of 4 stroke, one cycle operation is completed in 4 stroke of the piston, that is one cycle is completed in every 2 revolutions of the crankshaft. Each stroke consists of 180, of crankshaft rotation and hence a cycle consists of 720, of crankshaft rotation. Otto built his first gasoline-powered engine in 1861. Three years he formed a partnership with the German industrialist Eugen Langen, and together they developed an improved engine that won a gold medal at the Paris Exposition of 1867.

In 1876 Otto built an internal-combustion engine utilizing the four-stroke cycle was patented in 1862 by the French engineer Alphonse Beau de Rochas, but since Otto was the first to build an engine based upon this principle, it is commonly known as the Otto cycle. Components of four stroke Diesel Engine as shown in Fig1.2. Because of its reliability, its efficiency, and its relative quietness, Otto's engine was an immediate success.

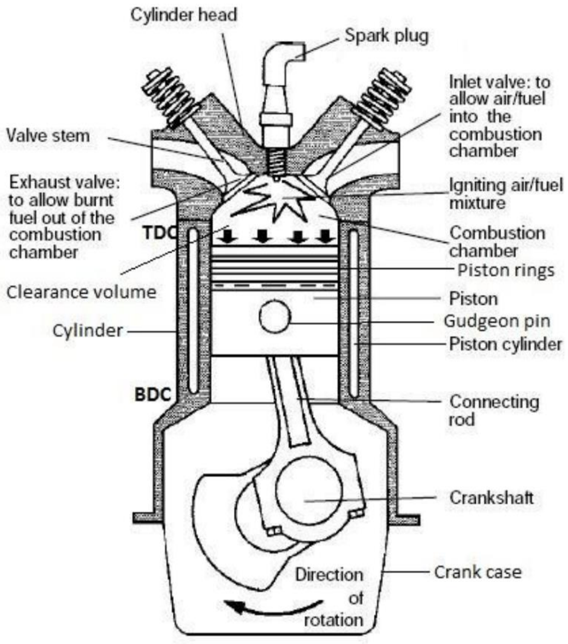


Fig 1.2. Components of four stroke Diesel Engine.

1.3.1. Working of Four Stroke Diesel Engine

Diesel engines, like gasoline engines, are considered internal combustion engines. This means fuel is burned inside the main part of the engine where the power is produced. This made diesel engines more efficient than the steam-powered engines at the time, which were external combustion engines that burned fuel outside the cylinders of the engine. Diesel engines use four-stroke combustion cycles to operate as shown in Fig 1.3.

1.3.1.1. *Intake Stroke:*

In suction stroke piston starts at Top Dead Centre (TDC) of the cylinder and moves to the Bottom Dead Centre (BDC). Outlet valve will be closed and inlet valve will be open to allow the fresh charge of mixed fuel and air into the cylinder.

1.3.1.2. *Compression Stroke:*

In compression stroke, once piston reaches BDC and moves back TDC, inlet valve will be closed. As the piston moves towards TDC. It compresses air inside the cylinder and compression takes place. Hence it is called compression stroke.

1.3.1.3. *Power Stroke:*

In expansion stroke, both the valves are closed. When piston reaches top of its stroke. The fuel is sprinkled by the Fuel Injector and the fuel mixture is ignited due to high temperature and pressure generated inside the cylinder and push down the piston to BDC. Hence it is known as power or expansion stroke. The power generated in this stroke is stored in the flywheel for its further utilization in the other strokes**.**

1.3.1.4. *Exhaust Stroke:*

In this stroke exhaust valve is opened when piston reaches to BDC and moves to upward. Piston pushes out the burnt gases to the atmosphere through the exhaust valve. Hence called exhaust stroke and the engine is ready to begin the cycle again.

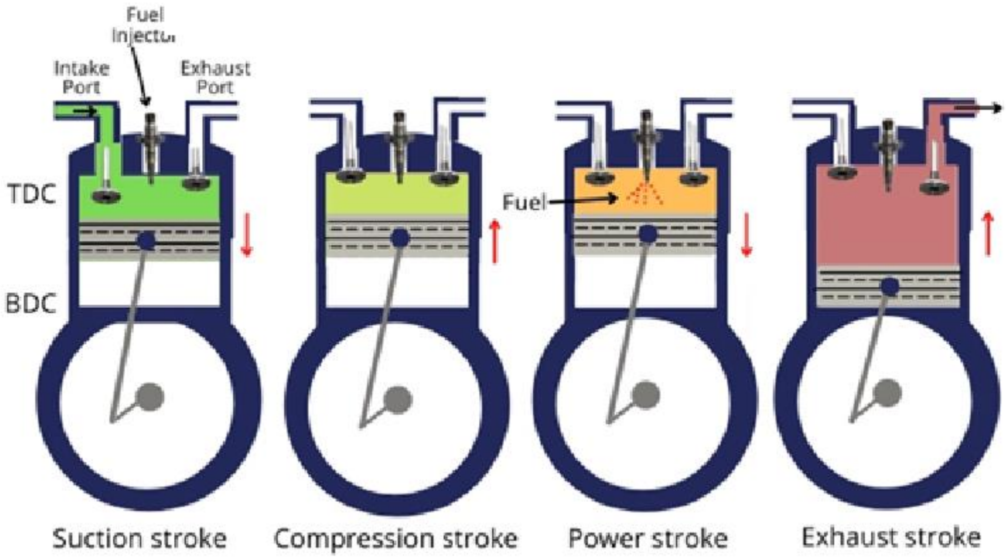


Fig 1.3. Working of four stroke of diesel engine

1.3.2. Applications of a Four Stroke Engine

1) Four stroke engine widely used in automobile industries.
2) They are used in bus, trucks and other transportation vehicles.
3) They are used in pumping system.
4) These engines find application in mobile electric generators.

1.3.3. Advantages of Four Stroke Diesel Engine

1) Four stroke engines give higher efficiency.
2) It creates less pollution.
3) Less wear and tear due to good lubrication system
4) It is quitter in operation.
5) It runs cleaner due to no extra oil added in fuel.
6) They give high rpm at low power.

1.3.4. Disadvantages of Four Stroke Diesel Engine

1) ***Complicated design:*** A 4 stroke engine has complex valve mechanisms operated & controlled by gears & chain. Also there are many parts to worry about which makes it harder to troubleshoot.
2) ***Less powerful:*** As power gets delivered once every 2 rotations of crankshaft (4 strokes), hence 4 stroke is less powerful.
3) ***Expensive:*** A four stroke engine has much more parts than 2 stroke engine. Sothey often require repairs which leads to greater expense.

1.3.5. Comparison Between 2-Stroke Engine and a 4-Stroke Engine

1) A 4-stroke engine weighs 50% heavier than a 2-stroke engine.
2) A 4-stroke engine is more efficient than a 2-stroke engine because fuel is consumed once every 4 strokes.
3) A 2-stroke engine creates more torque at a higher RPM, while a 4-stroke engine creates a higher torque at a lower RPM.
4) A 4-stroke engine is quieter than a 2-stroke engine.

5) 2-stroke engines tend to wear out fast because they are designed to run at a higher RPM.
6) 2-stroke engines are easier to fix because of simple construction.

1.4. IMPROVING THE ENGINE PERFORMANCE

Diesel engines produce more harmful emissions when compared with their counterpart spark ignition (SI) engines, particularly oxides of nitrogen (NOx) and particulate matter (PM). From past decade, increasing ecological concerns have tightened the diesel engine pollutant emission norms. This spurred the diesel industry to produce more efficient and cleaner engines. Various research works have been carried out on cylinder design, nozzle diameters, biodiesel and diesel fuels, injection timing and pressure can be varied to improve the engine performance.

By varying the injection timing, injection pressure, CR and nozzle holes leads to enhance the performance parameters like brake thermal efficiency, meanwhile reduced the emission characteristics in biodiesel fuelled diesel engine but however oxides of nitrogen emission has been increases as number of holes increase in injector nozzle.

Several strategies are available to lower the harmful emissions from a diesel engine and to increase engine performance such as:

a) Alternation of fuels that have similar fuel properties of diesel fuel.
b) Controlling the combustion by modifying injection timing, turbo charging and changing the combustion strategy.
c) After treatment of exhaust gases by oxidation catalyst and a particle filter.

1.5. FUEL INJECTION

Fuel injection in an internal-combustion engine, introduction of fuel into the cylinders by means of a pump rather than by the suction created by the movement of the pistons. Diesel engines do not use spark plugs to ignite the fuel that is sprayed, or injected, directly into the cylinders, instead relying on the heat created by compressing air in the cylinders to ignite the fuel. In engines with spark ignition, fuel-injection pumps are often used instead of conventional carburettors. Fuel injection into a chamber upstream from the cylinders distributes the fuel more evenly to the individual cylinders than does a carburettor system; more power can be developed and undesirable emissions are reduced. In engines with continuous combustion, such as gas turbines and liquid-fuelled rockets, which have no pistons to create a pumping action, fuel-injection systems are necessary. Fuel injection system works by atomizing the fuel at high pressure, mixing it with clean air as it passes the inlet manifold, before entering the combustion chamber of each cylinder. Fuel injection is usually electronically-controlled system for injecting a precise amount of atomized fuel into the cylinders or the intake airstream of an internal combustion engine.

1.5.1. Components of Fuel Injection System

The fuel injection system is more important for diesel engine. By pressurising and injecting the fuel, the system forces it into air that has been compressed to high pressure in the combustion chamber. Components of fuel injection system as shown in Fig 1.4 and discussed below.

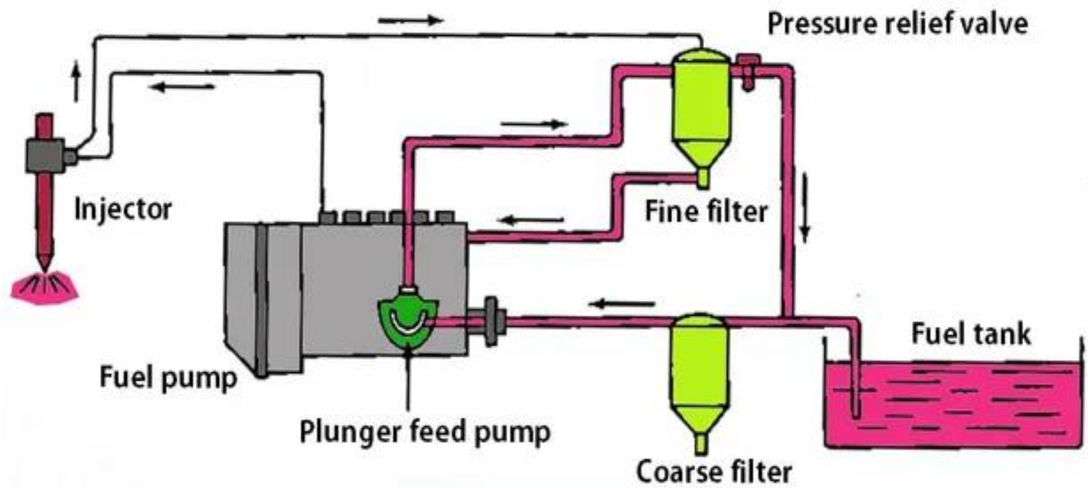


Fig 1.4. Components of fuel injector

1.5.1.1. *Fuel Tank:*

The Fuel tank is the fuel reservoir for the engine which holds the fuel and it will help to maintain the temperature of the fuel below the flash point. It also a corrosion-resistant and leak proof to pressures of at least 30 kPa. The fuel tank will be provided with a safety valve to relieve the excessive pressure. It will be able to dissipate the heat from the fuel coming from the engine.

1.5.1.2. *Fuel Feed Pump:*

The Fuel feed pump is used to feed the fuel from the fuel tank to the injection pump. It has the spring loaded plunger actuated through the push rod from a camshaft. When the push rod is at minimum position, the spring force on the plunger will create suction in the pump to flow the fuel from the fuel tank into the injection pump. When the cam is turned to its maximum lift position, the plunger is lifted upwards. The inlet valve will be closed and the fuel will be forced through the outlet valve. See the following schematic sketch of the fuel feed pump.

1.5.1.3. *Injection Pump:*

The main function of the fuel Injection pump is to deliver the right amount of the fuel into the injector under high pressure. (usually, the range will be 120 to 200 bar) at the correct instance to the injector fitted in each cylinder head.

1.5.2. Functions of the System

The function of a fuel injection system is to meter the appropriate quantity of fuel for the given engine speed and load to each cylinder, each cycle, and inject that fuel at the appropriate time in the cycle at the desired rate with the spray configuration required for the particular combustion chamber employed. It is important that injection begin and end cleanly, and avoid any secondary injections. To accomplish this function, fuel is usually drawn from the fuel tank by a supply pump, and forced through a filter to the injection pump. The diesel fuel injection system has four main functions:

1.5.2.1. *Feeding fuel:*

Pump elements such as the cylinder and plunger are built into the injection pump body. The fuel is compressed to high pressure when the cam lifts the plunger, and is then sent to the injector.

1.5.2.2. *Adjusting Fuel Quantity:*

In diesel engines the intake of air is almost constant, irrespective of the rotating speed and load. If the injection quantity is changed with the engine speed and the injection timing is constant, the output and fuel consumption change. Since the engine output is almost proportional to the injection quantity, this is adjusted by the accelerator pedal.

1.5.2.3. *Adjusting Injection Timing:*

Ignition delay is the period of time between the point when the fuel is injected, ignited and combusted and when maximum combustion pressure is reached. As this period of time is almost constant, irrespective of engine speed, a timer is used to adjust and change injection timing – enabling optimum combustion to be achieved.

1.5.2.4. *Atomising Fuel:*

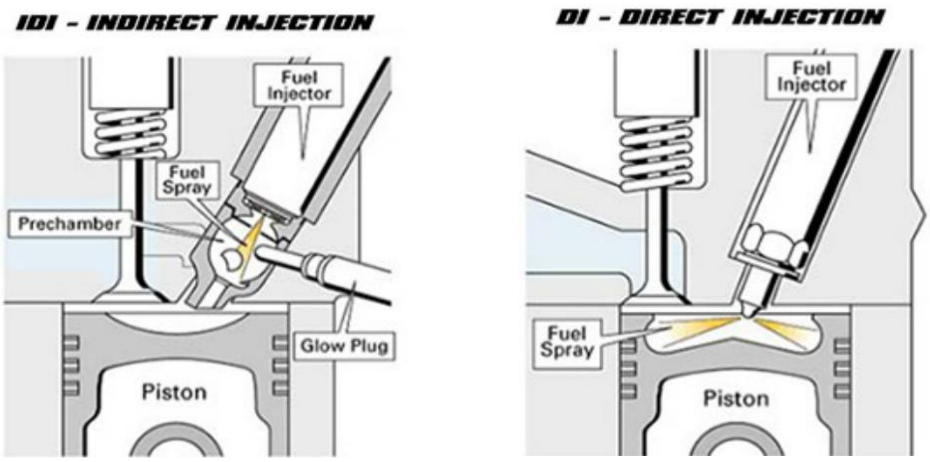
When fuel is pressurized by the injection pump and then atomised from the injection nozzle, it mixes thoroughly with air, thus improving ignition. The result is complete combustion.

1.5.3. Types of Fuel Injection Systems

1.5.3.1. *Indirect Injection:*

IDI stands for Indirect Injection. This technology is used for achieving higher engine speeds in the diesel engines. It is mainly used in light-duty diesel engines fitted in the earlier generation passenger cars, sedans & Multi-Utility vehicles. In the Indirect Injection, the injector does not spray the diesel directly over the piston as shown in Fig1.5 (a). Instead, it injects the diesel in an auxiliary or a ‘Pre-combustion chamber' located inside the cylinder head.

The Pre-combustion chamber creates swirling effect in the compressed air. This helps in mixing of diesel with the air uniformly. In this system, the air moves fast while mixing the fuel and air. During the compression stroke, air from the engine cylinder first enters the pre-combustion chamber. As the air is compressed, its temperature rises and thus, the air becomes hot. At this moment, an injector injects fuel into the pre-combustion chamber. The combustion begins in the pre-chamber and then, it spreads to the entire cylinder.



(a) (b)

Fig 1.5. Types of injection systems (a) Indirect injection (b) Direct injection

1.5.3.2. *Direct Injection:*

DI stands for direct injection. It is a basic type of fuel injection system. This Direct injection system inject the fuel into combustion chamber above the piston directly as shown in Fig1.5 (b).In the modern technology, the direct injection that is used in many vehicles in this current world. By using this DI we can better control over the fuel delivery inside the engine. Because of this, it is only the air that enters the chamber through the intake valve and permits a better air-fuel mixture. And in the end, this well-mixed charge allows for better and more efficient combustion. In the present CRDI system is mainly used in diesel engines because of advantages of that system.

1.5.4. Split Injection

Split injection has been shown to be a powerful tool to simultaneously reduce soot and NOx emissions for DI and IDI diesel engines when the injection timing is optimized. It is defined as splitting the main single injection profile in two or more injection pulses with definite delay dwell between the injections.In modern diesel engines, the fuel is not always injected at one time into the combustion chamber, but depending on the engine speed and load, the injection is divided into as many as three individual injections.

Split injection is divided into three injections:

1. ***Pilot injection:*** is an effective way to reduce the ignition delay and high rates of pressure rise at the start of injection. Pilot injection enhances ignition and combustion, and reduces NOx and soot emissions.
2. ***Main injection:*** During the main injection, as the name suggests, the main part of the fuel is fed into the cylinder.
3. ***Post injection:*** The post injection terminates the injection process.

Split injection combined with the exhaust gas recirculation (EGR) and exhaust gas turbo charging (EGT) reduces emissions and improves fuel economy. As opposed to single injection, pilot injection will form a higher-temperature and fuel-lean region. When the main injection enters the high-temperature region, the main injection fuel burns more rapidly, soot formation rates decreases, and the net soot production decreases dramatically. The negative influence of the pilot injection combustion is that it restricts the combustion performance of the split injection. So to reduce emissions and to increase the brake thermal efficiency of the diesel engine studying on injection timing and injection pressures is essential.

1.6. INJECTION TIMING

Injection timing is the timing of when fuel is injected into the cylinder, which alters when the combustion takes place. The time of when fuel is injected can be altered to be injected at different points in time. The manufacturer of an engine does recommend certain timing, which is the timing they set it at when the engine is first made. This timing is usually balanced to get as much power as possible, while still remaining in legal limits for emissions. Adjusting injection timing is also often referred to as spill injection. Fuel injection timing of engine as shown in Fig 1.6.

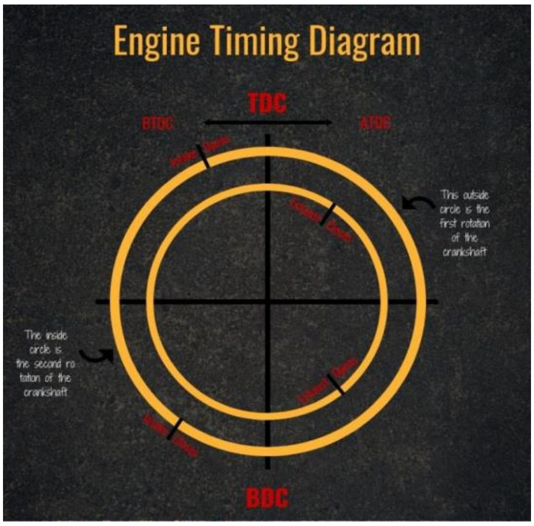


Fig 1.6. Fuel injection timing diagram of engine

1.6.1. Diesel Engine Injection Timing Adjustments

Advancing the timing of an engine means that you are moving the combustion up in time. In other words, you are adjusting the timing so that ignition happens earlier than when the manufacturer originally set it to occur. When talking about timing of any kind, but especially advancing, you'll often hear or see the term BTDC, or Before Top Dead Centre. Top Dead Centre, or TDC, for a particular piston is when that piston is at the very top of the cylinder, or furthest from the crankshaft. The opposite, Bottom Dead Centre, or BDC, is when the piston is at its lowest point in the cylinder, closest to the crankshaft. So, BTDC would be the point before the piston is at its uppermost point in the engine. A timing advance is the number of degrees BTDC that ignition occurs.

Retarding the timing of an engine is essentially the opposite of advancing. It is when you adjust timing so that ignition occurs after the manufacturers original specified time. People will retard the ignition timing of their engines for various reasons, although it is less common. Some of these reasons are fuel economy and performance.

1.6.2. Advantages and Disadvantages of Advancing the Timing

1.6.2.1. *Advantages:*

Advancing the timing will usually increase the amount of power your engine produces. It will also usually increase fuel efficiency. The original engine manufacturers set the timing to balance power and emissions, so that the engines they produce get as much power as possible while staying within legal emissions regulations. This means that they aren't originally set to produce the most power that the engine is capable of. And if your engine is older, or it's had some work done. Any little thing could affect your timing, so it may just need a bit of an advance to up the power.

1.6.2.2. *Disadvantages:*

Advancing the timing can lead to more smoke. It can cause a lot more vibration in the engine, making it noisier. It will also increase NOx emissions, which is the reason manufacturers usually retard the engines slightly in the first place. And if you don't care about any of those things, it will actually affect the performance of the engine in other ways; advancing the timing will often decrease and delay boost.

However, It is well known that advanced fuel injection timing leads to increase in ignition delay period, earlier and rapid combustion. So to reduce ignition delay, high combustion temperatures and pressures and also more importantly to reduce NOx emissions retarded fuel injection is used in our study. Nowadays research woks were carried for CI engine by adopting CRDI technology. CRDI technology has very suitable for CI engine to improve the performance and reduce the emissions of engine.

1.7. COMMON RAIL DIRECT INJECTION SYSTEM

The term CRDI stands for Common Rail Direct Injection. The technology directly injects fuel into the cylinders of a diesel engine through a single, common line, known as the common rail. The common rail is connected to all the fuel injectors.

Regular diesel direct fuel-injection systems have to build up pressure for every new injection cycle. The CRDI set up was shown in Fig1.7. Engines featuring the new common rail maintains a constant pressure regardless of the injection sequence. This pressure is said to be permanently available throughout the fuel line. Instant atomization takes place and this spray is very fine and evenly distributed aiding efficiency and power delivery. Also the injectors can inject up to 5 times per combustion cycle which gives a more uniform and controlled combustion and helps extract maximum energy from the combustion cycle. Technologically the engine's electronic timing regulates injection pressure according to engine speed and load. The electronic control unit (ECU) modifies the injection pressure with precision which is in relation to the data obtained from sensors on the cam and crankshafts.

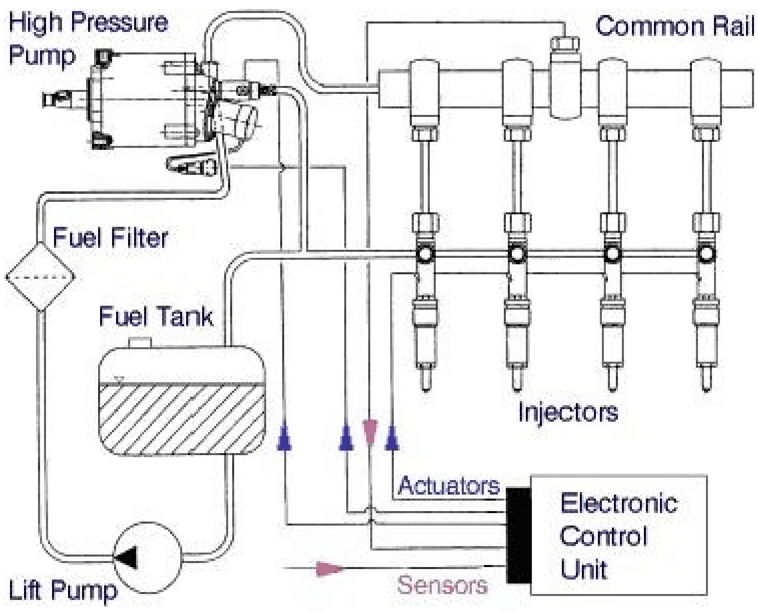


Fig 1.7. Common Rail Direct Injection (CRDI)

1.7.1. Working Principle

A high-pressure pump supplies pressurized fuel. The pump compresses the fuel at the pressures of about 1,000 bar. It then supplies the pressurized fuel via a high-pressure pipe to the inlet of the fuel rail. The fuel-rail distributes the fuel to individual injectors which then inject it into the combustion chamber.

Moreover, most modern CRDI engines use the Unit-Injector system with a Turbocharger, which increases power output and meets stringent emission norms. Additionally, it improves engine power, throttle response, fuel efficiency and controls emissions. Barring some design changes, the basic principle & working of the CRDI technology remains primarily the same across the board. However, its performance depends mainly on the combustion chamber design, fuel pressures, and the type of injectors used.

1.7.2. Advantages of CRDI System

1. CRDI system can control the flow of fuel in accordance with the load and speed of the engine.
2. This system requires only one fuel pump for multiple cylinders.
3. CRDI system is beneficial for the environment as it reduces noise, smoke and particulate matter.
4. It gives high power output at low rpm.
5. The main advantage of the CRDI system is fuel economy.

1.7.3. Applications of CRDI

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 car models ranging from city cars to premium executive cars.

Some of the Indian car manufacturers who have widely accepted the use of common rail diesel engine in their respective car models are the Hyundai Motors, Maruti Suzuki, Fiat, General Motors, Honda Motors, and the Skoda. In the list of luxury car manufacturers, the Mercedes-Benz and BMW have also adopted this advanced engine technology. All the car manufacturers have given their own unique names to the common CRDI engine system.

However, most of the car manufacturers have started using the new engine concept and are appreciating the long term benefits of the same. The technology that has revolutionized the diesel engine market is now gaining prominence in the global car industry.

CRDI technology revolutionized diesel engines and also petrol engines (by introduction of GDI technology). By introduction of CRDI a lot of advantages are obtained, some of them are, more power is developed, increased fuel efficiency, reduced noise, more stability, pollutants are reduced, particulates of exhaust are reduced, exhaust gas recirculation is enhanced, precise injection timing is obtained, pilot and post injection increase the combustion quality, more pulverization of fuel is obtained, very high injection pressure can be achieved, the powerful microcomputer make the whole system more perfect, it doubles the torque at lower engine speeds. The main disadvantage is that this technology increases the cost of the engine. Also this technology can't be employed to ordinary engines.

1.8. DIESEL EMISSIONS

Diesel engines convert the chemical energy contained in the fuel into mechanical power. Diesel fuel is injected under pressure into the engine cylinder where it mixes with air and where the combustion occurs. The exhaust gases which are discharged from the engine contain several constituents that are harmful to human health and to the environment. Table 1.1 lists typical output ranges of the basic toxic material in diesel fumes.

Table 1.1. Emissions from Diesel Engine

CO (ppm)	HC (ppm)	DPM (g/m^3)	NO_X (ppm)	SO_2 (ppm)
5 - 1500	20-400	0.1-0.25	50-2500	10-150

Carbon monoxide (CO), Hydrocarbons (HC), and Aldehydes are generated in the exhaust as the result of incomplete combustion of fuel. A significant portion of exhaust hydrocarbons is also derived from the engine lube oil. When engines operate in enclosed spaces, such as underground mines, buildings under construction, tunnels or warehouses, carbon monoxide can accumulate in the ambient atmosphere and cause headaches, dizziness and lethargy. Under the same conditions, hydrocarbons and aldehydes cause eye irritation and choking sensations. Hydrocarbons and aldehydes are major contributors to the characteristic diesel smell. Hydrocarbons also have a negative environmental effect, being an important component of smog.

Nitrogen oxides (NOx) are generated from nitrogen and oxygen under the high pressure and temperature conditions in the engine cylinder. NOx consist mostly of nitric oxide (NO) and a small fraction of nitrogen dioxide (NO_2). Nitrogen dioxide is very toxic. NOx emissions are also a serious environmental concern because of their role in the smog formation.

Sulphur dioxide (SO_2) is generated from the sulphur present in diesel fuel. The concentration of SO_2 in the exhaust gas depends on the sulphur content of the fuel. Low sulphur fuels of less than 0.05% sulphur are being introduced for most diesel engine applications throughout the USA and Canada. Sulphur dioxide is a colourless toxic gas with a characteristic, irritating odour. Oxidation of sulphur dioxide produces sulphur trioxide which is the precursor of sulphuric acid which, in turn, is responsible for the sulphate particulate matter emissions. Sulphur oxides have a profound impact on environment being the major cause of acid rains.

Diesel particulate matter (DPM) as defined by the EPA regulations and sampling procedures, is a complex aggregate of solid and liquid material. Its origin is carbonaceous particles generated in the engine cylinder during combustion. The primary carbon particles form larger agglomerates and combine with several other, both organic and inorganic, components of diesel exhaust. Generally, DPM is divided into three basic fractions.

i. Solids - dry carbon particles, commonly known as soot.

ii. SOF - heavy hydrocarbons adsorbed and condensed on the carbon particles, called Soluble Organic Fraction.

iii. SO4 - sulphate fraction, hydrated sulphuric acid.

The actual composition of DPM will depend on the particular engine and its load and speed conditions. "Wet" particulates can contain up to 60% of the hydrocarbon fraction (SOF), while "dry" particulates are comprised mostly of dry carbon. The amount of sulphates is directly related to the sulphur contents of the diesel fuel.

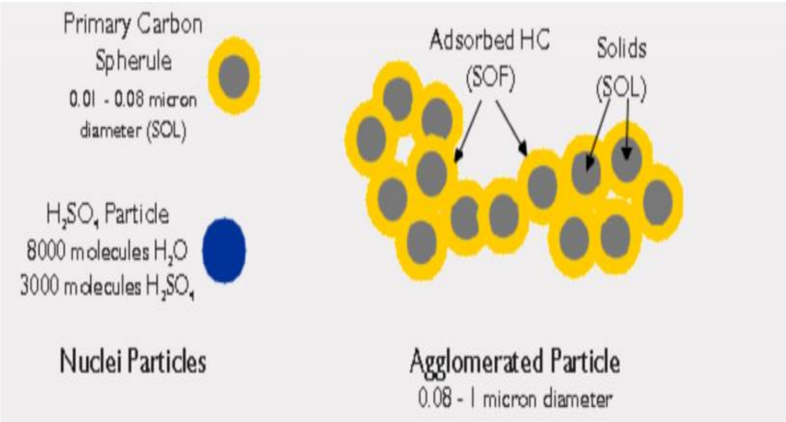


Fig 1.8. Schematic Composition of Diesel Particulate Matter

Diesel particulates are very fine. The primary (nuclei) carbon particles have a diameter of 0.01 - 0.08 micron, while the agglomerated particles diameter is in the 0.08 to 1 micron range. As such, diesel particulate matter is almost totally respirable and has a significant health impact on humans. It has been classified by several government agencies as either "human carcinogen" or "probable human carcinogen". It is also known to increase the risk of heart and respiratory diseases.

Polynuclear Aromatic Hydrocarbons (PAH) are hydrocarbons containing two or more benzene rings. Many compounds in this class are known human carcinogens. PAHs in the exhaust gas are split between gas and particulate phase. The most harmful compounds of four and five rings are present in the organic fraction of DPM (SOF).

1.8.1. Impacts of Diesel Emissions

Human health, our environment, global climate and environmental justice are all affected by diesel emissions.

1.8.1.1. *Human Health*

Exposure to diesel exhaust can lead to serious health conditions like asthma and respiratory illnesses and can worsen existing heart and lung disease, especially in children and the elderly. These conditions can result in increased numbers of emergency room visits, hospital admissions, absences from work and school, and premature deaths.

1.8.1.2. *Environment*

Emissions from diesel engines contribute to the production of ground-level ozone which damages crops, trees and other vegetation. Also produced is acid rain, which affects soil, lakes and streams and enters the human food chain via water, produce, meat and fish. These emissions also contribute to property damage and reduced visibility.

1.8.1.3. *Global Climate*

Climate change affects air and water quality, weather patterns, sea levels, ecosystems, and agriculture. Reducing greenhouse gas (GHG) emissions from diesel engines through improved fuel economy or idle reduction strategies can help address climate change, improve our nation's energy security, and strengthen our economy.

1.8.1.4. *Environmental Justice*

EPA seeks to provide all people the same degree of protection from environmental and health hazards and equal access to decision-making to maintain a healthy environment in which to live, learn, and work. DERA activities further EPA’s commitment to reduce health and environmental harm from diesel emissions in all communities throughout the country.

1.9. EMISSION CONTROL SYSTEM

The emission control system includes a series of functions that the vehicle performs to keep the emissions as low as possible. Harmful emissions like carbon monoxide (CO), hydrocarbons (HC) and nitrogen oxide (NOx) are minimised with the help of an emission control system. We can generally classify the emission control methods into two categories, prevention/active and destruction/passive. Automotive engines have become a lot more advanced and the new technologies have reduced the emission output. Some small improvements like using the correct air/fuel ratio, enhanced combustion techniques, and variable fuel ratio are some of the advancements. However, we explain more impactful and necessary emission control methods that are needed to comply with new stringent emission norms.

1.9.1. Catalytic Converters

Catalytic Converter is popular equipment used in all vehicles to destruct tailpipe emissions. Even though catalytic converters have been used since 1970, there have been multiple advancements in the working of catalytic converters. The two-way catalytic converter could only control CO and HC only whereas the three-way setup also controls oxides of Nitrogen (NOx) and is hence used in all modern cars. Modern catalytic converters convert harmful gases and pollutants into carbon dioxide (CO_2) and water (H_2O) It holds some precious metals like platinum (Pt), Palladium (Pd) and Rhodium (Rh) that perform oxidation and convert the harmful gases into CO_2 and water.

Fig 1.9. Catalytic converter

The catalytic converter works under the effect of heat and lack of heat can degrade the overall efficiency. Hence, when the engine is cold, the catalytic converter can not work to its optimum efficiency, and to control emissions. The catalytic converter is the most important emission control device that destructs harmful emissions.

1.9.2. Evaporative Emission Control

Evaporative emission control not only helps to reduce the emissions but also saves fuel and increases the overall efficiency of the vehicles. In technical terms, an evaporative emission control system eliminates the evaporation of hydrocarbons from the fuel tank and circulates them into the combustion chamber. The key mechanical component of this emission control system is the carbon canister that stores the hydrocarbons. The carbon canister absorbs the fuel vapours via loose chemical bonds and releases them via the purge solenoid that is controlled via the onboard computer module.

1.9.3. Exhaust Gas Recirculation (EGR)

Exhaust Gas Recirculation is very useful in lowering emissions and keeping the engine temperatures low as possible. EGR is mostly available with turbocharged petrol and diesel engines and petrol engines adopted this technology much earlier than diesel engines. Talking about the construction, the exhaust manifold channels some of the exhaust gases into the intake manifold and that helps to decrease the engine temperature and overall emissions. EGR is used in diesel engines to reduce NOx emissions whereas it comes in handy to increase efficiency in petrol engines.

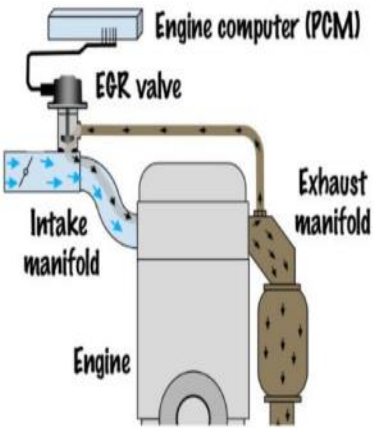


Fig 1.10. Exhaust Gas Recirculation (EGR)

1.9.4. Diesel Particulate Filter (DPF)

Diesel Particulate Filter (DPF) is a honeycomb filter that traps the soot post-combustion from the exhaust manifold. It traps all the solid particles and collects them to a certain capacity post which the substances are burnt. The burning of soot is called regeneration and it happens when the car is driving in a controlled environment at certain engine RPMs.

Fig 1.11. Diesel Particulate Filter (DPF)

1.9.5. Selective Catalyst Reduction (SCR)

Selective Catalyst Reduction (SCR) is also an advanced emission control method that is mostly used in higher-capacity diesel engines. SCR technology which is also known as Adblue has become important for high-capacity diesel engines to comply with stringent BS6 norms. The fluid reacts with NOx and converts it into nitrogen, water and CO2. The converted gases are far less harmful when compared to NOx and go out from the exhaust pipe. SCR system can reduce NOx emissions by up to 90% and helps to comply with stringent BS6 norms.

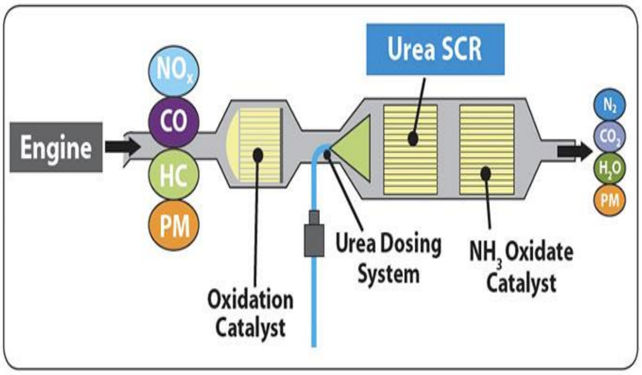


Fig 1.12. Selective Catalyst Reduction (SCR)

CHAPTER 2

2. LITERATURE SURVEY

2.1. STUDIES ON INJECTION TIMING

Atuldhar *et al.* [1] Used single cylinder Common rail direct injection (CRDI) diesel engine which is coupled with an AC dynamometer. Used 10%, 20%, 50% Karanja (vegetable oils) biodiesel blends with varying pilot injection timing and Pilot injection pressure in CRDI diesel engine at 1500 RPM at 500 bar and 1000 bar fuel injection pressure(FIP). At 500 bar FIP for all test fuels, BSFC was lowest at 15° CA and12° CA SOMI timings at 21° CA and 18° CA SOPI timings respectively. Among all combinations of injection timings investigated at 500 bar FIP with multiple injections, BSFC was lowest at 12° CA SOMI timings at 18° CA SOPI timing for all test fuels. At 21° CA and 18° CA SOPI timings for all test fuel sat 1000 bar FIP, BSFC was lowest at 9° CA and 6° CA SOMI timings respectively. Their report that BSFC of test fuels increased with increasing concentration of Karanja biodiesels in the test fuels. BTE of Karanja biodiesel blends was light higher with increasing the concentration. lowerKaranja biodiesel showed lower BSCO and BSCHC emissions as comparison to Mineral diesel .But KOME10 has higher BSHC emission compared to mineral diesel at same operation conditions. BSNOx emissions from KOME20 and KOME 10 higher than mineral diesel. BSNOX emissions are higher for 1000 bar pressure compared to 500 bar pressure at same SOPI and SOMI.

Avinash Kumar Agarwal *et al.* [2] used a single cylinder research engine to investigate the effects of fuel injection strategies and start of fuel injection timing on particulate size–number, surface area, and volume concentration distributions by using engine exhaust particulate sizer (EEPS) spectrometer. Investigations have been conducted at three different fuel injection pressures (300, 500, 750 bar) and four different start of injection timings. Particulate size–number concentration increases with increasing engine load (BMEP) and it reduces with increasing fuel injection pressure.

Babu *et al.* [3] used biodiesel fuel on CRDI diesel engine and investigated the effect of single and split injection strategy on combustion, performance and emissions characteristics. In single injection strategy, Nozzle opening pressure and fuel injection timing was varied from 200 to 600 bar and 19° to 27° CA bTDC respectively. In split injection strategy, start of main injection timing and post injection timing was varied from 19° to 25° CA bTDC and -5° CA bTDC to 5° CA aTDC respectively. The reported that the B100-90%-10% has the maximum brake thermal efficiency of 34.43%. Minimum unburned hydrocarbon and smoke emissions were obtained in B100-75%-25%. Maximum nitric oxide emission was obtained in B100-90%-10%.Biodiesel yield rises with rise in time and temperature as higher temperature reduces the viscosity of the oil and decreases the kinetic energy required for the reaction. The maximum yield of 93.751 obtained at a time. 120min and temperature 55°C. BTE of the engine increased with increase in NOP and FIT. This is because of increasing NOP reduce the fuel droplet diameter and improve the fuel evaporation rate which causes complete combustion.

CenkSayin *et al.* [4] 1/3 of the Petroleum fuels are consumed in the IC engines which has less power than 185 kW. The experiments conducted on a four stroke naturally The original injection timing of engine the is 27 CA bTDC thickness of advance shim located in connection place between engine and fuel is 0.25mm and adding one shim advances the injection timing 3°CA. Experiments were carried out in five different injection timing (21°. 24°. 27°, 30 and 33° CA bTDC) Values. CO_2 Emissions increased because of the improved combustion.

Pranab*et al.* [5] used a single cylinder direct injection diesel engine by homogeneous charge compression ignition combustion of diesel fuel using a novel dual injection strategy. They reported that reduction in smoke and NOX emissions is observed over a range of 20°–11° bTDC MIT when compared to baseline emissions. For 20- bTDC MIT, 56% and 32% reduction in NOX emissions are observed for 45% and 60% loads respectively. At this injection timing, the corresponding reduction in smoke emissions is 60% and 56%. Based on considerations of simultaneous reduction in smoke and NOX, and 20- bTDC MIT is identified as the best to run the engine in HCCI–DI mode. They also stated that by studying different combustion parameters, it is observed that there is an improvement in performance and emissions with marginal loss in thermal efficiency when the main injection timing is 20^0 before top dead centre.

J. Ramachander *et al.* [6] Used biodiesel as fuel in CRDI Diesel engine. They investigated the emission and combustion characteristics of diesel engines operating under the reactivity-controlled compression ignition mode. The primary objective of this research is to examine the effect of fuel injection timings (7.5, 12.5 and 17.5 bTDC) and injection pressure (500 and 1000 bar), experimental testing is carried out on single cylinder water-cooled testing engines at constant speed of 1500 rpm with variable engine load (16, 20 and 24 Nm). In case of a fuel injection pressure of 1000 bar, the maximum brake specific fuel consumption of 0.42 kg/kWh is registered with a brake mean effective pressure of 3.2 bar. In this experimental study, Box-Behnken based response surface methodology was used to predict the optimal input parameters, resulting in the optimal combination of output and emission parameters. In addition, a statistically relevant test analysis of variance has been developed to obtain a regression model. Results have shown that the proposed 'Regression Model' is ideally suited to 0.095 standard deviation,0.972 modified R2 and 18.482 acceptable accuracy. This analysis also attempts to describe the application of the response surface methodology analysis to optimize the emission and performance parameters.

2.2. STUDIES ON INJECTION PRESSURE

Agarwal *et al.* [7] investigated the effects of FIP on CRDI engine with karanja biodiesel (KOME) blends at a constant engine speed. From experimental studies it is reported that BTE of biodiesel blends is higher compared to neat diesel. Lower BSCO and BSHC emissions with KOME10, KOME20 were noticed compared to diesel and KOME50. However higher BSNOx trend is observed with all blends compared to diesel, General trend is that increasing FIP improves the performance.

Gumus *et al.*[8] used Lombardini single cylinder engine fuelled with biodiesel blends, from emission analysis it was observed that NOx value increased, whereas smoke opacity, CO and UBHC decreased. Varying FIP results showed that increased injection pressure lowers HC CO and smoke opacity.

Labecki and Ganippa, [9] experimented with multi-cylinder turbo-charged diesel engine at 1500 rpm for different FIP with rapeseed biodiesel. They reports that higher FIP reduced soot emission but increased NOx for biodiesel blends compared to diesel. It is concluded that combination of EGR and the injection timing can lower NOx emissions at the cost of CO, UBHC.

Saravanan *et al.* [10] used common rail direct injection (CRDI) diesel engine fueled by 30% pine oil biodiesel blend (P30) and studies the impact of high fuel injection pressure on the engine and the engine characteristics were analyzed. They reported that P30 fuel exhibits maximum BTE of 26.1% at 350 bar which is 6.9% higher than BTE of diesel at 200 bar. They also added that at 100% load condition, the P30 blend shows a maximum decrease in CO and HC emission at 300bar, which is 16.6% and 13.1% less respectively compared to sole diesel at 200 bar injection pressure. The NOx is increased by about 9.4% for P30 blend at 350 bar when compared to that of Diesel at 200 bar. It was found that the BSFC of P30 decreases when fuel injection pressure increases from 200 bar to 350 bar, but when compared with diesel at 200 bar, the P30 blend shows slightly higher BSFC.

2.3. STUDIES ON CHARGE DILUTION (EGR)

Agarwal *et al.* [11] evaluated the drawback of more NOx emissions while employing biodiesel which can be expelled by using EGR. Simultaneous application of EGR with biodiesel blends resulted in lower NOx emissions with minimum loss in fuel economy.

Krishna *et al.* [12] investigated the performance emissions and combustion characterises of common rail direct injection (CRDI) engine fuelled with waste plastic oil and diesel blends. The experiments are carried out with 10, 20 and 30% of diesel blended with waste plastic oil on volume basis. The investigations are carried out at a constant speed of 2000 rpm with load varying from 20 to 80%. Results illustrate that 9.23 % decrement of brake thermal efficiency and increment of 16.35% in NOx emissions are perceived when contrasted with diesel.

Santhosh *et al.* [13] carried out experiments on the CRDI engine by using 1-Pentanol-diesel blends with Exhaust Gas Recirculation (EGR) technique and estimated the performance and emission characteristics. Results depict that the influence of EGR was having a pessimistic effect on BTE and maximum diminishment was noticed for the blend P30D70 (30% of 1-pentanol+ 70% of diesel). HC and CO emissions were increased as the 1-pentanol proportion increased in the blend; this may be attributed to a lower cetane number of alcohol fuels. However effect of EGR was having a positive effect on NOx emissions, it got drastically decreased for all the 1-Pentanol blends when correlated to neat diesel. Summary of works on Performance and Emissions of CRDI Engine.

Radheshyam *et al.* [14] have witnessed the effects of 1- pentanol addition and Exhaust Gas Analyser(EGR) in CDRI engine, from results he has concluded that in-cylinder pressure, Mean Gas Temperature (MGT), Net Heat release rate (NHR) got declined at lower loads and NOx emissions came down by the addition of 1-pentanol additive in CRDI Diesel engine.

Parashuram *et al.* [15] investigated the influence of Exhaust gas recirculation (EGR) and injection pressure on the performance and emissions of CRDI engine using Jatrophacurcas biodiesel blends of 10% and 20% (B10 and B20). Experiments were carried out for three fuel injection pressures (FIP) of 300, 400 and 500 bar with 15% and 20% EGR rate at constant speed of 2000 rpm and standard injection timing of 150 BTDC. Parameters like brake thermal efficiency and emission characteristics such as smoke opacity, oxides of nitrogen (NOx), hydrocarbon (HC) and carbon mono-oxide (CO) were measured and analysed. The results showed improvement of performance in terms of brake thermal efficiency for blends B10, B20 and with 15%EGR rate. Smoke, HC and CO decreased while slightly increasing NOx emissions when working with biodiesel. In summary, it is optimized that engine running with combination of B20 blend and 15% EGR rate culminates into NOx reductions without affecting engine efficiency and other emissions like smoke opacity, hydrocarbon and carbon mono-oxide.

Parashuram *et al.* [16] investigated the influence of EGR, biodiesel blends for variable speed CRDI diesel engine for performance, emissions and combustion characteristics. Biodiesel blends of B10, B20 and B30 with EGR rates of 10%, 20% and 30% at constant load were tested. Performance parameters such as brake thermal efficiency (BTE) and emission characteristics like smoke opacity, oxides of nitrogen (NOx), Hydrocarbon (HC) and carbon mono-oxide (CO) were measured and analyzed. The results revealed about the improvement in performance with minimal effect emissions among the different fuels operation. The reduction of smoke, HC, CO with slight increase of NOx emissions was observed with usage of biodiesel blends. In summary, it is optimized that engine running on biodiesel blend of B20 with 20% EGR rate culminates into NOx reductions without affecting engine efficiency and other emissions like smoke opacity, hydrocarbon and carbon mono-oxide.

2.4. STUDIES ON SPLIT OR MULTIPLE INJECTION

Mahantesh *et al.* [17] used Palm Oil Methyl Ester on common rail direct injection (CRDI) single cylinder four stroke diesel engine has made to modify in terms of toroidalreentrant combustion chamber (TRCC) shape and 7 holes CRDI nozzle injector. In the first phase of work, experiment results showed that slightly improved in brake thermal efficiency (BTE) and reduced emissions except oxides of nitrogen (NOx) for POME fuelled engine operates under optimized MIS, fuel injection timing (IT) of -10° before top dead center (BTDC) and 600 bar injection pressure (IOP) in modified CRDI diesel engine. In the second phase of work, the performance of modified CRDI diesel engine is improved by increasing IOP from 600 bar to 900 bar at same MIS and fuel IT. The second phase of experiment results showed that percentage of increase in BTE by 2.47%, peak pressure (PP) by 13.69%, heat release rate (HRR) by 17.64%, NOX by 11.70% and percentage of decreased in ignition delay (ID) by 29.62%, combustion duration (CD) by 13.79%, unburnt hydrocarbon (UBHC) by 19.04%, carbon monoxide (CO) by 14.28%, smoke level by 20.93% as compared to first phase of work in modified CRDI diesel engine fuelled with POME.

Anil Bhaurao Wakale *et al.* [18] studied the effects of different injection strategies of a CRDI engine by using n-butanol and diesel blends at various proportions like 5%, 10%, and 20% by volume; from his baseline experiments, he has noticed that little drop of peak cylinder pressure by using the n-butanol blend. With the aid of spilt-injection strategy, the ignition delay got reduced and this lead to a 35% diminishment of NOx emissions at 20% blend concentration. Chemical sensitivity scrutiny determines that n-butanol acts as an important sink of OH radicals, hence it delays OH inducted ignition peak by 2 0CA, which leads to a reduction offlame temperatures, and that helps in the suppression of NOx emissions.

Ramesh Babu *et al.* [19] Investigated emission analysis of CRDI Diesel Engine fuelled with cotton seed oil biodiesel with multiple injection strategy. In this work an attempt is made to study the effect of various multiple injection strategies with different injection timings and dwell periods. In this multiple injection strategy of three injection pulses the pilot fuel quantity is fixed as 10% of total fuel injected, post injection fuel quantity is fixed as 0.5 mg. The dwell between pilot and main was varied at different main injection timing. The post injection is closely coupled with main injection with a dwell of 3 CAD. The main injection timing along with pilot and post was retarded from the recommended 23o bTDC in steps of 3 degrees. At all main injection timing the dwell of 10 CAD observed to be the best for smoke reduction, where as 20 CAD is better for NOx reduction. HC and CO emission observed to be reducing with multiple injection strategies.

Sanjoy*et al.* [20] studied on the effect of Quadruple injection (epMa) strategy on performance, emissions and noise with different Pilot injection timing and post injection dwell combinations(8 Nos) on a CRDI diesel engine with high EGR and fixed main injection schedule at4 operating loads and 5 speeds. The study shows that Quadruple injection strategy with retarded early and advanced pilot (early 35° and pilot 19°CA BTDC) and advanced post injection dwell timing1100 ms is superior in providing optimum results in emissions(NOx, PM, THC, CO) and combustion noise (CN)[1.55–2 dBA @Rig level and 0.7 dBA @ Vehicle level Pass by Noise (PBN)]. This gives the best results in brake specific fuel consumption (BSFC)[0.2 to 1.31%], Torque and brake thermal efficiency (BTE) performance w.r.t other combinations and base triple injections(pMa). In contrary, the quadruple injection strategy having advanced double pilots with delayed post injection dwell shows the best CN reduction. Best smoke results found with the combination of retarded pilots and advanced post injection dwell. This study shows the importance of injection timing specially the twin pilots(early, pilot) along with post injection dwell. Furthermore, it indicates the potentiality of newest Quadruple injection strategy over triple injection.

Tanaka *et al.* [21] have examined the influence of pilot injection and its parameters(fuel quantity and timing) on combustion noise and emissions on an automotive CRDI diesel engine equipped with EGR. The research work concluded that simultaneous reduction of CN combustion noise and emissions are feasible by reducing the influence of pilot burned gas by means of minimization of quantity and advancement the timing ofpilot injection.

Pavan *et al.* [22] used B20 blend (Palm Oil Methyl Ester 20% + neat diesel 80%) with varying pilot injection timing and pilot injection pressure in a common rail compression ignition diesel engine at part load condition. They reported that a reduction of 40% in NO emission was observed with the introduction of 10% pilot fuel and higher injection pressure at 32° bTDC compared with single injection mode at 23° bTDC. With the same strategy the smoke, HC and CO emission were found be lower at 34° bTDC by 26.2%, 19.2% and 21.5%. A slight increase in CO2 emission by 1.62% was noticed at 500 bar injection pressure at this injection timing compared to single injection mode. An increase in carbon dioxide emission was found to have increased when the timing was advanced to 34° bTDC. They have observed a reduction in peak pressure from 72 bar in single injection mode to 63 bar at a pilot injection timing of 34° bTDC and at a pressure of 500 bar.

J. M. Babu *et al.* [23] used Palm-munja biodiesel/diesel blend on CRDI diesel engine at constant speed of 1500rpm and investigated the performance, combustion and emission characteristics of Palm-(B20). Pilot injection timing (−30° bTDC to −50° bTDC). They found out that NOx emissions were lower for the multiple injection strategy as compared to single injection strategy. It was observed that CO emissions decrease by advancing of pilot injection timing, whereas they increases with increasing of pilot injection fuel quantity. They also stated that advancing of injection timing reduces the NOx, soot and CO emissions but it increases the HC emissions. Also as the increase of pilot injection quantity increases the NOx, soot and CO emissions, it also increase the HC emissions.

CHAPTER 3

3. EXPERIMENTAL SET-UP AND METHODOLOGY

3.1. TEST SET-UP

All the experiments were conducted on the premises of Apex Innovations Pvt. Ltd., Sangli (MS),India. A single cylinder, naturally aspirated, water cooled, VCR diesel engine coupled with eddy current dynamometer and data acquisition system was used for this investigation.

Fig 3.1. Single cylinder, CRDI with EGR, open ECU and Emission Analyzer

The engine was downsized to develop a maximum power of 3.5 kW by modifying the engine head by specially designed tilting cylinder block arrangement. The set-up was equipped with a jerk type fuel injection pump and a three hole injector. Initial tests were conducted on this set-up to obtain the reference data. The cylinder head was then modified to incorporate a six-hole injector nozzle without altering the combustion chamber geometry to investigate the effects of various FIPs and SITs on engine performance, emission and combustion parameters. The engine of the test facility is shown in Fig 3.1. The engine was equipped with a CRDI system (Bosch, E099GF231) to control FIP and SIT. This CRDI engine works with programmable Open ECU (Nirai7r, Sweden) for diesel injection; the engine is equipped with fuel injector, common rail with rail pressure sensor and pressure regulating valve, crank and cam position sensors, fuel pump and wiring harness. The technical specifications of the modified test engine for this study are given in Table 3.1

Table 3.1. Specifications of Test Engine

Name of the description	Details/values
Product	Enginetestsetup1cylinder,4stroke,Diesel(Computerized)
Productcode	224
Engine	MakeKirloskar,ModelTV1,Type1cylinder,4strokeDiesel,waterco oled,power 5.2kW at 1500rpm,stroke110mm,bore 87.5mm. 661cc, CR17.5
Dynamometer	Type eddy current, water cooled
Propeller shaft	With universal joints
Airbox	MS fabricated with orifice meter and manometer
Fuel tank	Capacity15litwithglassfuelmeteringcolumn
Calorimeter	Type Pipe in pipe
Piezo sensor	Range5000PSI,withlownoisecable
Crank angle sensor	Resolution1Deg,Speed 5500RPM with TDC pulse.
Data acquisition device	NIUSB-6210,16-bit,250kS/s.
Piezo powering unit	ModelAX-409.
Temperature sensor	TypeRTD,PT100andThermocouple,TypeK
Temperature transmitter	Typetwowire,InputRTDPT100,Range0–100DegC,I/PThermocouple, Range0–1200DegC,O/P4–20mA
Load indicator	Digital,Range0-50Kg,Supply230VAC
Load sensor	Loadcell,typestraingauge,range0-50Kg
Fuel flow transmitter	DPtransmitter,Range0-500mmWC
Air flow transmitter	Pressure transmitter,Range(-)250mmWC
Software	“Enginesoft” Engine performance analysis software
Rotameter	Enginecooling40-400LPH;Calorimeter25-250LPH
Pump	Type Mono block
Overall dimensions	W2000xD2500xH1500mm
Optional	Computerized Diesel injection pressure measurement

The test facility was equipped with essential instruments for online measurement of CP, FIP, crank angle, load on the engine, and temperature of –inlet air and exhaust gas, -coolant at inlet and outlet, -lubricating oil. Provision was also made to measure the flow rate of –cooling water, -air and –fuel. The entire signalling system was interfaced to laptop through data acquisition system to record all observation parameters using Windows based engine performance software "ICEngineSoft". This software serves the purposes like monitoring, reporting, data entry, data logging. Necessary signals are scanned and stored through online testing of the engine in RUN mode which can be used for further analysis. By providing the input values of density, heating value of fuel and the ambient temperature of air, the software gives the complete summary of combustion and performance of the engine. The exhaust gases were diverted to a sampling line for the measurement of emissions without increasing the back pressure in the exhaust pipe. Five gas emission analyzer (AVL DIGAS 444) and a smoke meter (AVL 437C) were used to measure vital emissions from the engine. Fig 3.2 shows the different components of CRDI diesel engine.

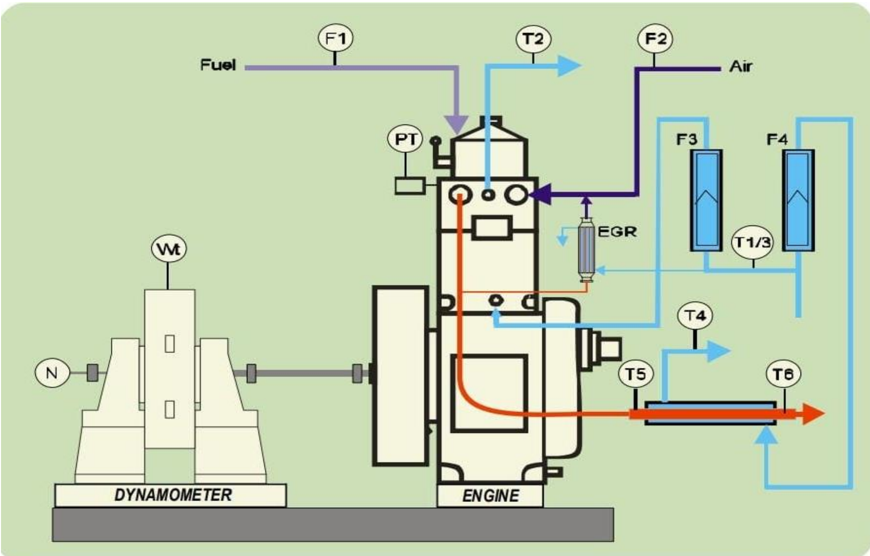


Fig 3.2. Components of CRDI Diesel Engine

3.2. SPECIFICATIONS OF DATA LOGGING INSTRUMENTS

The engine was built with essential instruments like Crank angle encoder to capture crank angle data. The Piezoelectric pressure transducers to track pressure data for corresponding crank angle from combustion chamber and at fuel line. The Airflow transmitter at the entry of Air box and Fuel flow transmitter at burette to log Air, fuel consumption. The temperature sensors/ transmitters such as resistance temperature detectors (RTDs)/thermocouples to capture the temperature data at each required location of engine setup. The instruments like Eddy current dynamometer, Load cell and loading wheel to apply required load and to log load data from engine setup. To interface above all instruments, the US based National instruments (NI) make USB-6210 model 16-bit Data acquisition device used to capture 250-kilo samples data per second (kS/s). The ICEngineSoft software used to analyze engine Performance and Combustion characteristics based on captured observations data with some input values of fuel physicochemical properties.

Table 3.2. Data logging instruments and their specifications

Sl. No.	Name (Model)	Details/value
1	Crank angle encoder (8.KIS40.1361.0360)	German based Kubler company make, 1 degree Resolution, 5500rpm speed with TDC pulse
2	Piezo sensors (S111A22)	USA based PCB Piezotronics company make 5000psi (344.75 bar) pressure range Piezo sensors at Combustion chamber and Fuel inline.
3	Data acquisition device	United States based NI make USB-6210, 16bit, 250kS/s
4	Software	National Instruments developed Lab VIEW based ICEngineSoft software
5	ECU	Make PE USA, Model PE3
6	Temperature sensor	Make Radix, Type RTD, PT100 and Thermocouple, Type K
7	Temperature transmitter	Make ABUSTEK USA, Type 2 wire, Input RTD/Thermocouple, Output 4 - 20mA
8	Load cell	Make VPG Sensotronics, Load cell, S Type strain gauge
9	Fuel flow transmitter	Make Yokogawa Japan, DP transmitter, Range 0-500mmWC
10	Air flow transmitter	Make Wika Germany, Pressure transmitter, Range 0-250mmWC

3.2.1. Crank Angle Encoder(8.KIS40.1361.0360)

	.	**1**	**3**	**6**	**1**	.	**0**	**3**	**6**	**0**
Type		**a**	**b**	**c**	**d**		**e**			

TheabovetablerepresentstheshaftordercodeforkublermakeIncrementaltype Rotary Encoder (8.KIS40):

- **a** – The number **1** denotes that 40mm diameter synchronous Flange for clamping
- **b** – The number **3** denotes that 6mm diameter 12.5mm length flat shaft
- **c** – The number **6** denotes that 5 Volts DC input supply RS422 with inverted signal
- **d** – The number **1** denotes that 2 meters PVC axial cable
- **e** – The number **0360** denotes pulse rate

The IP64, Logic level: RS422; Supply= 5VDC Incr/turn: 360 PPR

3.2.2. Piezo Sensors(S111A22)

The 6gram weight Pizeo sensor with Stainless Steel housing with 344.75 bar maximum pressure measurement range, 0.00145 sensitivity, 0.001 Hz low frequency response and >=400kHz Resonant frequency and 10-32 Coaxial jack.

3.2.3. Technical Specifications of PE Make Electronic Control Unit PE3

Size	:	11 x 12 x 3 centimetres
Weight	:	B0.4 kgf of aluminium &potted waterproof enclosures

Operational voltage : 6 – 22v DC supply

Operational temperature : -30°C to 75°C depends on loading

Active voltage : 3.25v (High) & 2.0v(Low)

The Maximum continuous supplied voltage for Digital system : 22v

3.2.4. Specifications of Data Acquisition Device

This model USB6210 made in United States of America (USA) by National instruments (NI). The ADC resolution is 16bits with sample rate 250-kilo samples per second (kS/s). The range of Operational current -16mA (high) & 16mA (low). The lowest & highest digital input voltages are 0to5.25. The DC input coupling with timing accuracy 50ppm of sample rate and resolution 50ns.

3.2.5. The Specifications of Load Cell

The VPG Sensotronics Company make 60001 Model S beam type 50kg Capacity load cell manufactured with high quality alloy steel with nickel coated plate presented above.

3.2.6. Differential Pressure Fuel Transmitter (Model-EJA110E-JMS5J)

Model EJA110E – JMS5J suffix codes and description:

J-stands for **Output signal** range**-** 4 to 20mA Direct Current (DC) with digital communication (HART 5/HART 7 protocols)

M-stands for **Measurement span** (capsule) range**-** 1 to 100kPa (4 to 400 inH_2O)

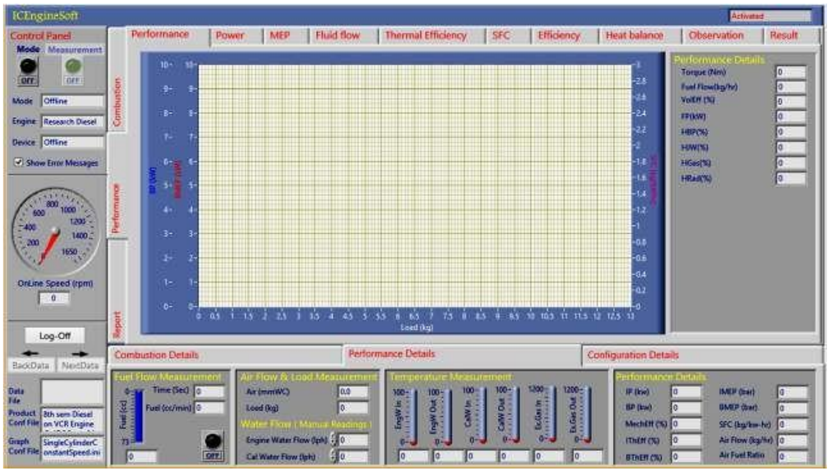
S-standsfor**Wettedpartsmaterial**ofCoverflange,processconnector:ASTMCF- 8M Capsule: Hastelloy DiaphragmC-276[4]; F316L SST, 316 L SST Capsule gaskets: Teflon-coated316L SSTVent/Drain plug 316SST

5-standsfor**Processconnections**withoutprocessconnector(1/4NPTfemaleonthe cover flanges)

J-stands for **bolts, nuts materials** made with B7carbon steel.

3.2.7. ICEngineSoft

The National instruments developed Lab VIEW based ICEngineSoft software to analyze Performance, Combustion characteristics using logged observation data from test setup and by some input fuel properties.



3.2.8. Temperature Transmitter

The USA based ABUSTEK company make 2 wire, Input Resistance Temperature Detector (RTD) or Thermocouple with Output of 4 -20mA.

3.2.9. Airflow Transmitter(SL-1-A-MQA)

The WIKA Company makes Airflow transmitterSL-1-A-MQA model 4-20mA output current and zero to 10bar pressure range.

3.3. SPECIFICATIONS OF EMISSION ANALYZERS

In the present work, the CRDI engine tailpipe emissions Carbon monoxide (CO), Carbon dioxide (CO_2), Unburned Hydrocarbons (HC), Oxygen (O_2) and NOx measured by using AVL DIGAS 444 model 5gasanalyzer.TheSmokeemissions measured by using AVL 437C model Smoke Meter. The detailed specifications of above 5 gas analyzer and Smoke meter given in Table 3.3 and their photographic views are presented in Figure 3.3.

Table 3.3. Specifications of emissions analysers

AVL 5 gas analyzer, Model: DI GAS 444 N				
Emission	**Unit**	**Range**	**Resolution**	**Accuracy**
CO	% vol	0 - 10	0.01	±0.02
HC	ppm	0-20000	1	±4
CO_2	% vol	0 - 20	0.1	±0.5
O_2	% vol	0 - 22	0.01	±0.02
NO_x	ppm	0-5000	1	±5

AVL Smoke meter Model:437C			
Emissions	Unit	Range	Resolution
Smoke density(K)	m^{-1}	0 - 9.99	0.01
Smoke opacity(%/HSU)	%	0-99.99	0.01

Fig 3.3. Photographic view of the emission analyzer

3.4. EXPERIMENTAL METHODOLOGY

Studies on single cylinder CRDI diesel engine by using retarded and split injection strategy were evaluated in this study. The engine was first run on no load at a rated speed for about 30 minutes allowing it to reach thermal equilibrium conditions. Coolant temperature at engine outlet was maintained in the range of 75 ± 2 ^{0}C, and the lubricating oil temperature was maintained in the range of 85 ± 2 ^{0}C throughout the testing.

Engine testing was carried out at constant speed at varying loads, Start of injection timing and fuel injection pressure was const analysis ant for analyzing its performance, combustion and emission characteristics. Eddy current dynamometer was connected to the engine to apply load on the engine.

Three phases of testing conditions are considered to analyse the performance, combustion and emissions characteristics of the engine. At phase 1 the experiment was conducted on retarded single injection at 19^0, 16^0, 13^0, 10^0, 7^0 bTDC. The best performance, combustion and emissions characteristics are obtained at 13^0 and 16^0 bTDC. To obtain a broader view on reducing the emission and increasing the performance spilt injection is introduced in our study. In phase 2, the experiment was conducted on advanced pilot injection timings at intervals of 3 degrees from 30^0 bTDC to 36^0 bTDC, to analyze the combustion and emission characteristics in a single-cylinder diesel engine. The best result was obtained at 30^0 bTDC pilot injection timing. The next set of testing conditions were carried out at constant pilot injection timing and main injection was retarded at intervals of 3 degrees from 13^0 bTDC to 19^0 bTDC. The averaged combustion data was then used to compute variation of CP, HRR, PRR with crank angle, occurrence of HRR and PRR, combustion duration, etc. Emissions of NOx, CO, UHC, and smoke opacity were measured.

CHAPTER 4

4. RESULTS AND DISCUSSION

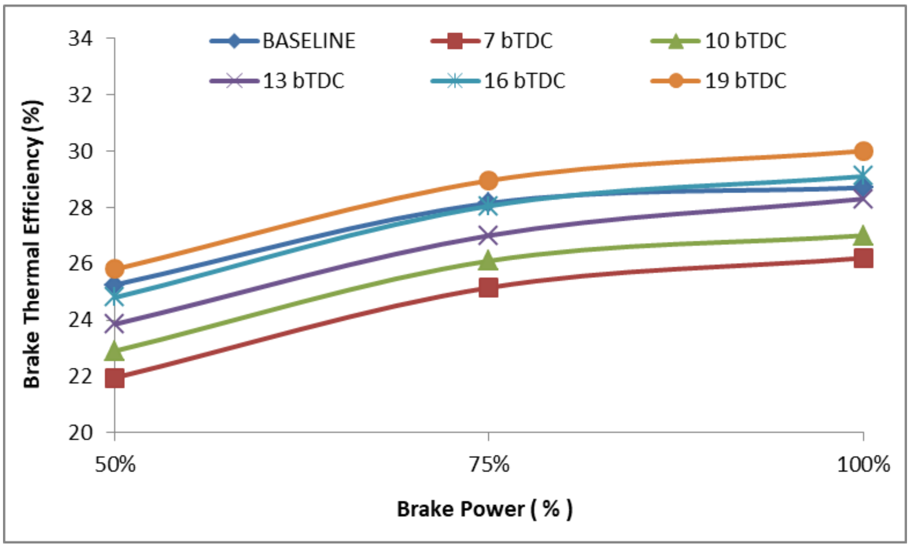
4.1. PERFORMANCE CHARACTERISTICS

4.1.1. Brake Thermal Efficiency

BTE of the engine is the fraction of energy supplied by the fuel that is converted into useful brake power. Fig.4.1 (a) shows the variation in Brake Thermal Efficiency (BTE) with MIT at different loads. With the Increase in load, BTE increases for all injection timings. BTE at 16^0 bTDC is slightly more when compared to baseline. This could be due to better atomization, enhanced air fuel mixing quality at 400 bar and reduce in wall wetting leads to better burning of fuel. It is observed that BTE is decreased for different loads as the MIT (Main Injection Timing) is retarded. This is because as the fuel injection timing is retarded, the power output reduces and main fuel quantity automatically increases to maintain the same power output. Thus total fuel consumption increases as the MIT is retarded which results in a drop in thermal efficiency of the engine. The maximum BTE obtained at 19^0 bTDC.

Fig. 4.1 (b) shows the variation in Brake Thermal Efficiency (BTE) with constant pilot and MIT at different loads. Retarding fuel injection timing 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. Pilot injection strategy is used to increase BTE hence complete combustion can be obtained. When compared to baseline and MIT, the MIT with pilot injection strategy i.e., at 30^0 PIT + 19^0 MIT showed maximum BTE at part load (75%) and slightly decreased at full load condition. This is due to split injection strategy. The pilot injection decreases the ignition delay and increases the cylinder temperature this helps in complete combustion of fuel and increased efficiency.

Fig. 4.1 (c) shows the variation in Brake Thermal Efficiency (BTE) with constant MIT and different pilot injection at different loads. It was observed that the BTE reaches maximum at 75% of load and then slightly decreased with further increased the engine load due to the effect of shorter ignition delay period. However, the shorter ignition delay reduced the timing in air fuel mixture process that led to partial combustion, resulting in reduction of engine performance at 100% load condition. The BTE of the engine has increased with an increase in SOMI (Start of Main Injection) timing decreased with an increase in SOPI (Start of Pilot Injection) timing and increased with an increase in fuel injection quantity due to longer ignition delay period and better air-fuel mixture which improved the burning efficiency and resulted in higher BTE.



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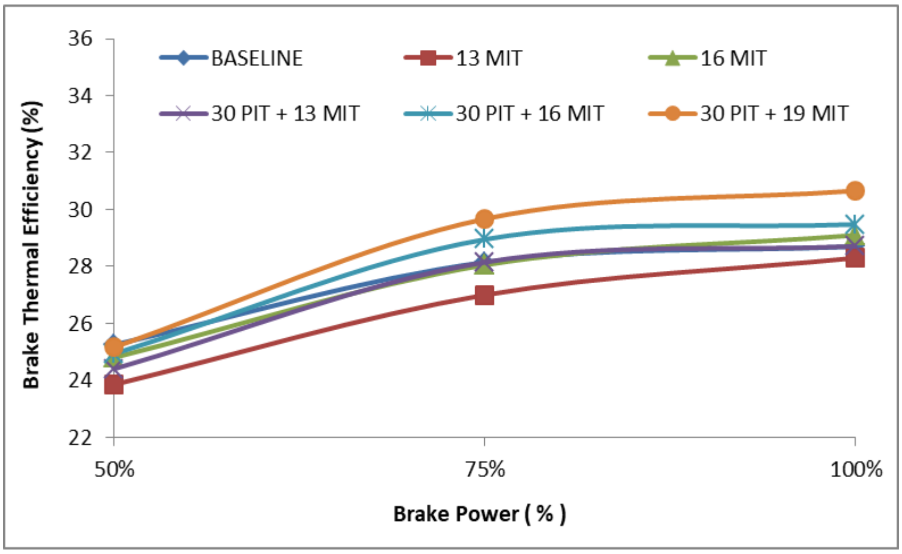

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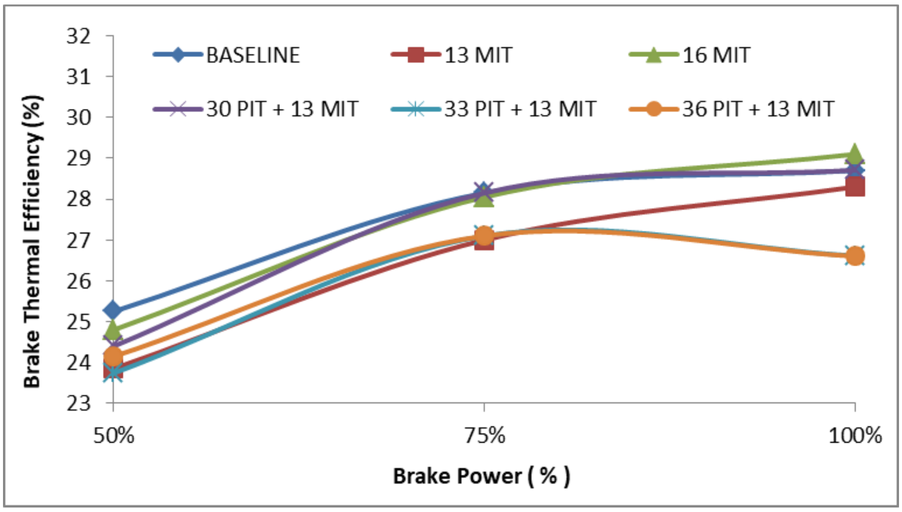


(c)

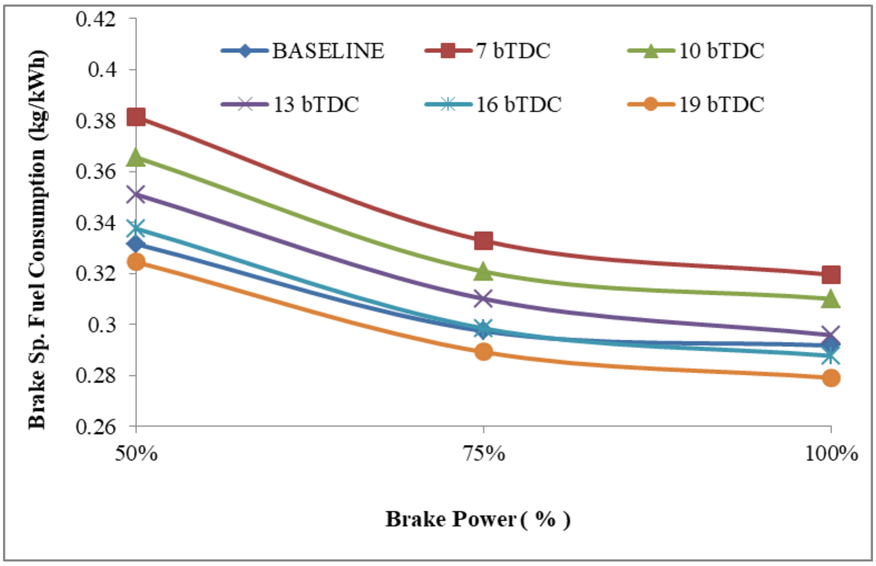
Fig 4.1. Variation of BTE with Brake power at different operating conditions.

4.1.2. Brake Specific Fuel Consumption

BSFC of the engine is the amount of fuel consumed for developing unit brake power. Fig.4.2 (a) shows the variation in brake specific fuel consumption (BSFC) with MIT at different loads. The figure shows that BSFC increases as the MIT was retarded and decreases as the load increases. This reduction in BSFC may be explained by the fact that as the engine load increases, there has been 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 is less. Hence BSFC is reduced if load increases. The best fuel economy can be obtained at 16^0 bTDC (400 bar) when compared to baseline (23^0 bTDC, 210 bar). This is due to the fact that higher injection pressure generates smaller sized fuel droplets which mix better with air, improving the combustion process.

Fig. 4.2 (b) shows the variation in brake specific fuel consumption (BSFC) with constant pilot and retarded MIT at different loads. Best performance of the engine at baseline operation may be credited to the higher fraction of premixed combustion and the occurrence of peak PRR just before TDC. The reduction in the engine performance at retarded injection timings may be attributed to the decrease in the fraction of heat release in premixed combustion, late combustion phase and the extension of combustion into the expansion stroke. This phenomenon worsens with the increase in retardation of SIT. The improved performance of the engine at higher FIP of 400 bar and slightly retarded SIT of 19^0 bTDC may be on account of the improved combustion and lower ID due to fine atomization of fuel.

Fig.4.2 (c) shows the variation in Brake Thermal Efficiency (BTE) with constant MIT and different pilot injection at different loads. In general, the BSFC was found to decrease with increase in the engine load for all test fuels. This was because of improved combustion in the cylinder at higher loads. As the applied load increases the in-cylinder temperature rises this increases the fuel burning rate.



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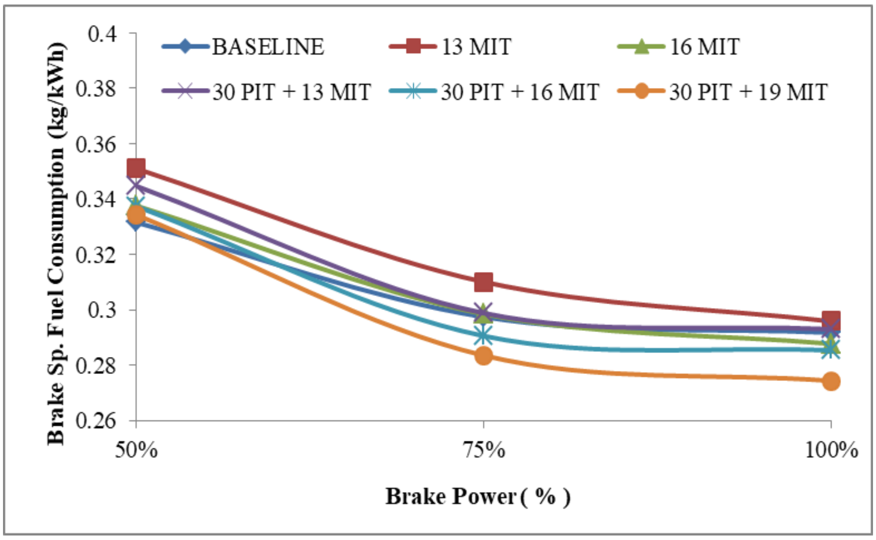

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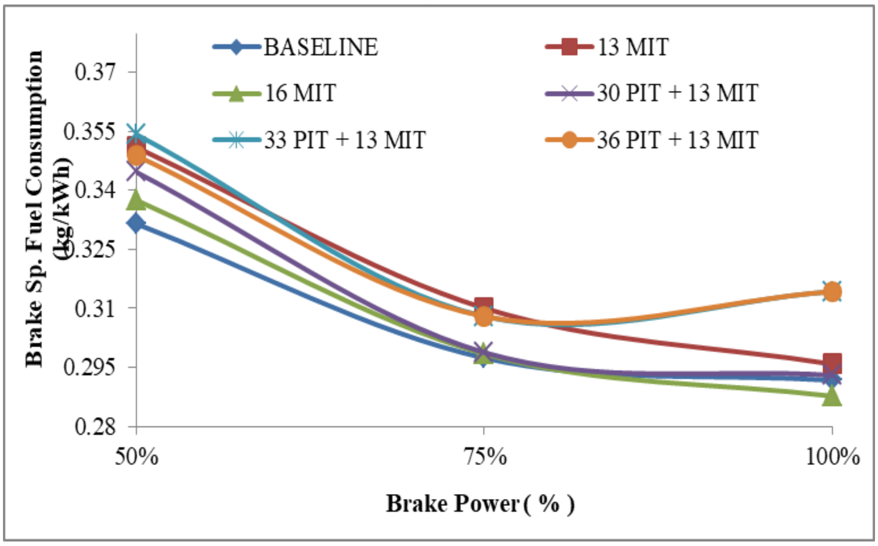


(c)

Fig 4.2. Variation of BSFC with Brake power at different operating conditions.

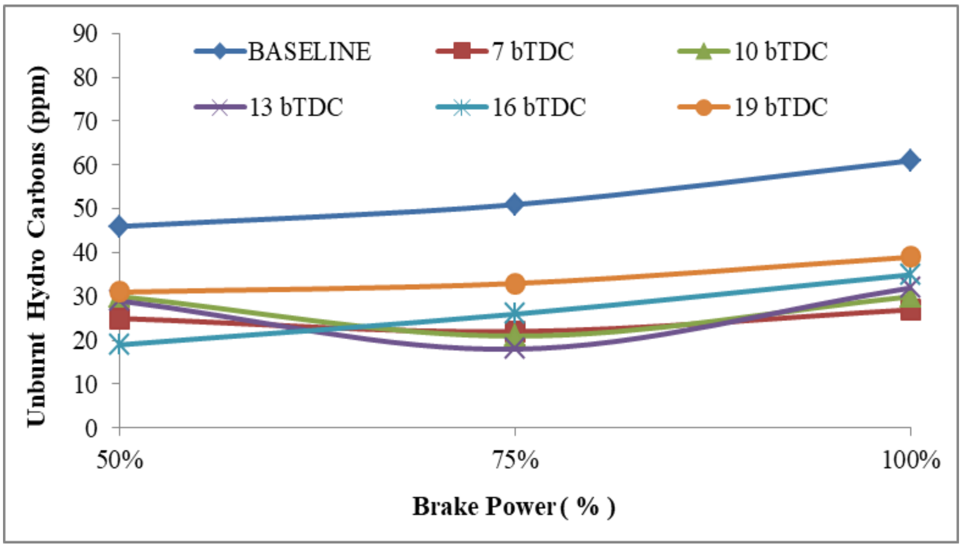
4.2. EMISSION CHARACTERISTICS

4.2.1. Emissions of Unburnt Hydrocarbons

Unburnt hydrocarbon emissions from internal combustion engines are because of incomplete combustion of fuel. Lower excess air–fuel ratio, the oxygen content of the fuel and heat of vaporization are the important factors which affects hydrocarbon (HC) emissions. Fig.4.3 (a) shows the variation of hydrocarbon emission at various injection timings and load condition. Generally, at higher loads HC emissions are higher because of lower excess air–fuel ratio which makes combustion difficult at fuel- rich zones resulting in higher HC emission. HC emissions increased with retarded Injection timing. It lowers the in-cylinder pressure and temperature during combustion, which in-turn increases engine-out HC emissions. Compared with baseline, retarded injection timing at 13° bTDC is found to be best optimum value.

Fig.4.3 (b) shows the variation in smoke emissions with constant pilot and MIT at different loads. It is observed that, as the MIT is retarded HC emissions show an increasing trend. This could be because retarded main injection retards the combustion phasing and subsequently ignition delay increases which increases the mixing time of the pilot fuel and results in a sharp increase in HC emissions. Another reason could be that as the MIT is retarded, in-cylinder gas pressure and temperature also reduces and in such a low temperature environment, un-burnt hydrocarbons do not oxidize any further and end up with higher HC concentrations in the exhaust. By introducing pilot injection, the HC emissions were reduced. The HC emissions were minimum at 30^0 PIT and 19^0 MIT.

Fig.4.3(c) shows the variation in HC emissions with constant MIT and different pilot injection at different loads. The HC emission reduced at 30^0 PIT and 13^0 MIT. Advancing the injection timing causes earlier start of combustion relative to the TDC. Because of this, the cylinder charge, being compressed as the piston moves to the TDC, had relatively higher temperatures, and thus lowered the unburned HC emissions. When compared with baseline, both the pilot injection and retarded main injection showed greater reduction in HC emissions.



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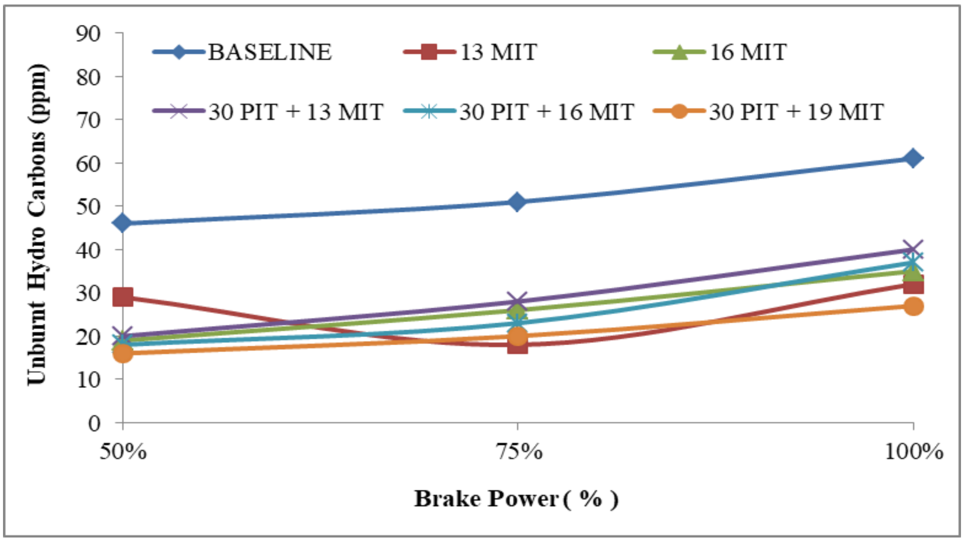

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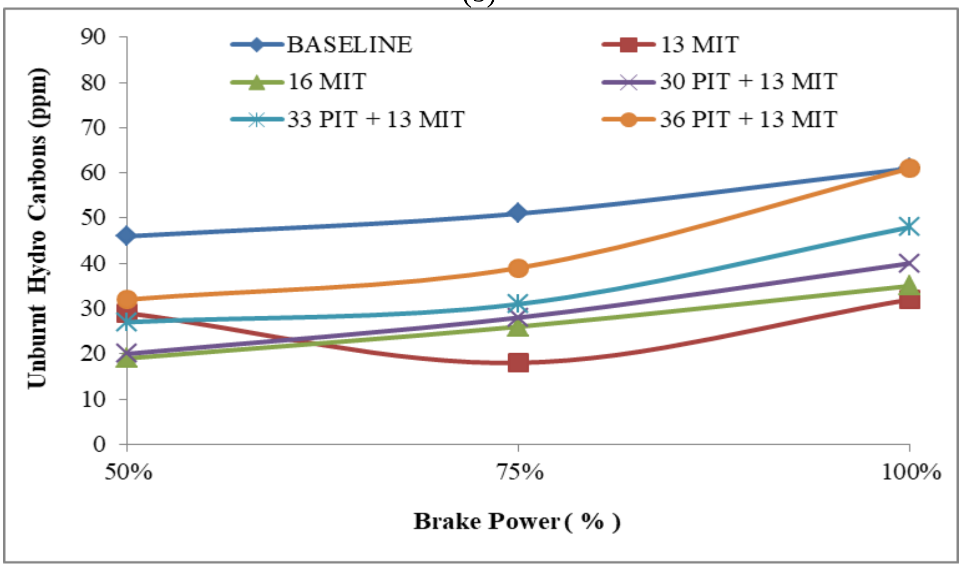


(c)

Fig 4.3. Variation of UHC emissions with Brake power at different operating conditions.

4.2.2. Emissions of Carbon Monoxide

CO is an intermediate combustion product formed because of incomplete combustion of hydrocarbon fuels. Emissions of CO at various engine loads for different IT are shown in Fig.4.4 (a) CO emissions increase with increasing engine load. As engine load will increase, relative fuel–air ratio also increases, leading to richer heterogeneous combustion, which results in inefficient mixing of fuel and air, resulting in higher CO emissions under high engine load conditions. Retarding the injection timing led to increased CO because it pushed majority of combustion into the expansion stroke, which decreased the temperature and pressure during the later a part of the combustion in the expansion stroke, which successively increases CO formation. Compared with baseline, the retarded injection at 13°bTDC showed optimum reduction in CO emissions.

Fig. 4.4 (b) shows the variation in CO emissions with constant pilot and MIT at different loads. CO emissions increase with increasing engine load. The results have showed that the CO emissions were minimum at 30^0 PIT and 19^0 MIT. It can be noted from the figure that as the brake power increases, the CO_2 emissions increases linearly and are found to be highest at full load condition for all the test samples. This can be attributed to the fact that at as the engine load increases, greater amount of fuel is injected inside the engine cylinders resulting in the increase of incylinder temperature which is assisting complete combustion.

Fig.4.4 (c) shows the variation in NOx emissions with constant MIT and different pilot injection at different loads. Here split injection strategy is utilized with the pilot injection advancing but the main injection being constant at 13°bTDC. With the advanced injection timings the oxygen concentration in the cylinder increases. This increased concentration attracts the free carbon molecules and as the free oxygen molecules are also more the tend to form carbon dioxides and leave the exhaust. Comparatively CO_2 is less harmful that CO. Therefore this results in less carbon monoxide emissions in the cylinder. The CO emissions showed a reducing trend at 30^0 PIT and 13^0 MIT.

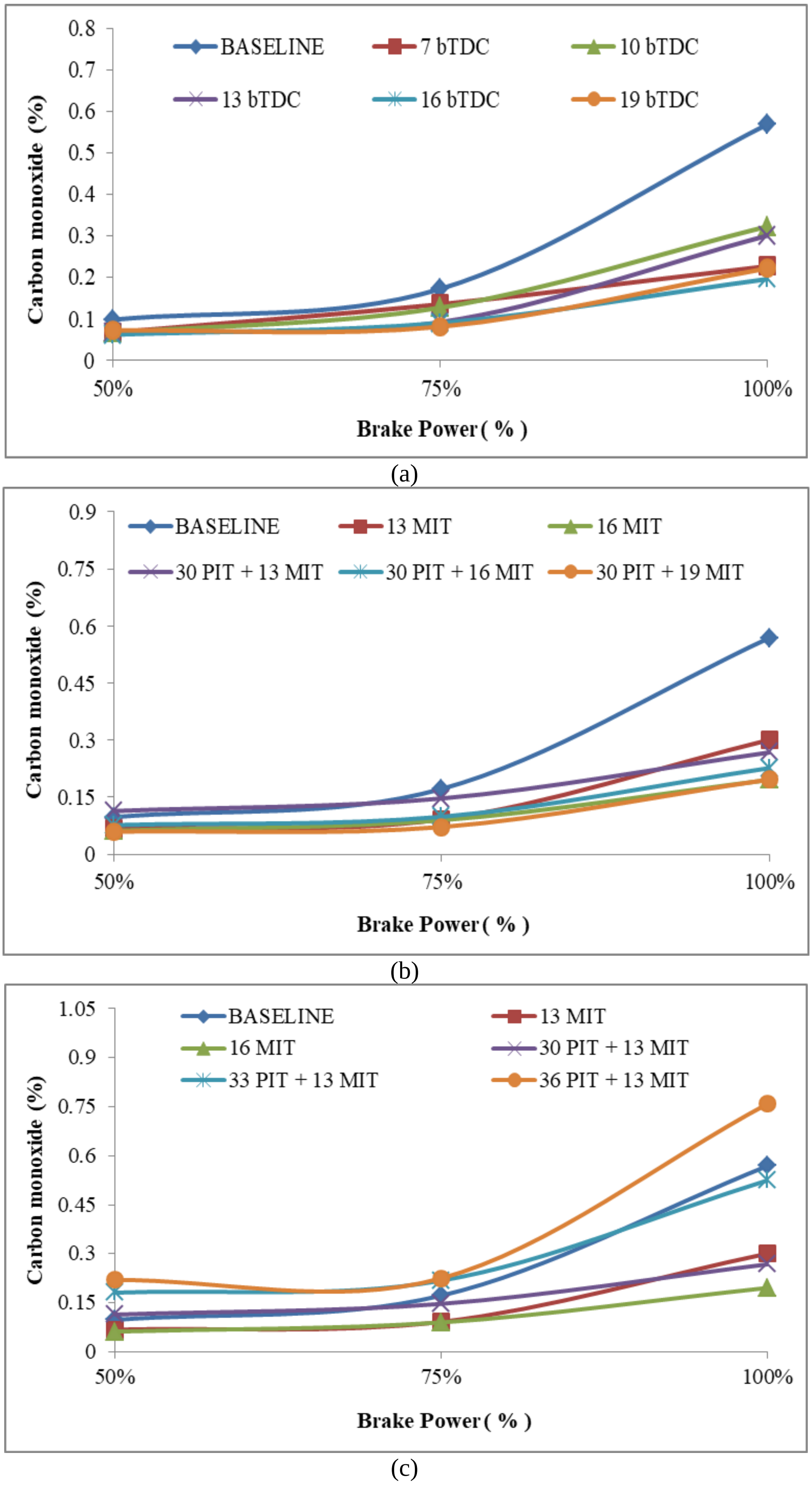


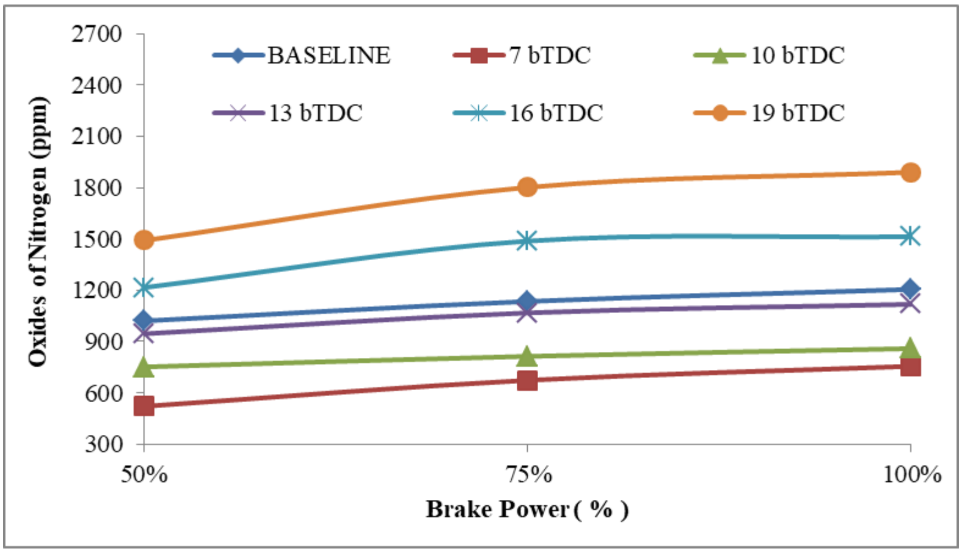
Fig 4.4. Variation of CO emissions with Brake power at different operating conditions.

4.2.3. Emissions of Oxides of Nitrogen

The formation of the NOx emission mainly depends on the in-cylinder temperature, oxygen concentration, and also the time available for the reactions to take place. Dissociation of the diatomic nitrogen (N_2) and oxygen (O_2) molecules into their atomic states at very high temperatures undergo a series of reactions in the combustion chamber to produce thermal NOx. Fig.4.5 (a) displays the variation of NOx emission at various injection timings and load conditions. In general, as the BP increases, NOx emission tends to increase at all working conditions. This may be attributed to the rise in fuel–air ratio at higher loads which successively increases the gas temperature within the combustion chamber leading to higher NOx. The NOx emission significantly reduced when retarding injection timing from 19° bTDC to 7° bTDC. This is often because of the time available for reaction is a smaller amount and decrease in combustion gas temperature. When compared with baseline 13° bTDC showed optimum reduction in NOx emission.

Fig.4.5 (b) shows the variation in NOx emissions with constant pilot and retarded MIT at different loads. It has observed that with an increase in load, NOx emission increases, the higher reduction is observed at low/medium load however at higher load reduction in NOx emission is very less. It was observed that NOx emissions were lower for the multiple injection strategy as compared to single injection strategy. When compared with baseline and retarding single injection timing, pilot injection showed much reduction in NOx at 30^0 PIT and 13^0 MIT. The NOx emission is decreased at retarded multiple fuel injection timing due to the shorter ignition delay duration.

Fig.4.5 (c) shows the variation in NOx emissions with constant Main Injection Timing and different pilot injection at different loads. It is observed that NOx emission was found to be lower at a pilot injection angle of 30^0 bTDC for 400 bar. This can be attributed due to two reasons; firstly injecting certain quantity of fuel in the form of pilot injection reduces the amount of fuel burning when the main injection happens closer to TDC leading to reduction in the peak temperature. Second, the heat produced by the burning of pilot fuel reduces the ignition delay period of the main fuel which lead to its quick combustion and reduced heat release rate. However, it is also seen that NOx emission has increased to a decreased when the pilot injection was advanced further. When compared to baseline and retarded single injection timing, 30° PIT and 13° MIT showed optimum reduction in NOx emissions.



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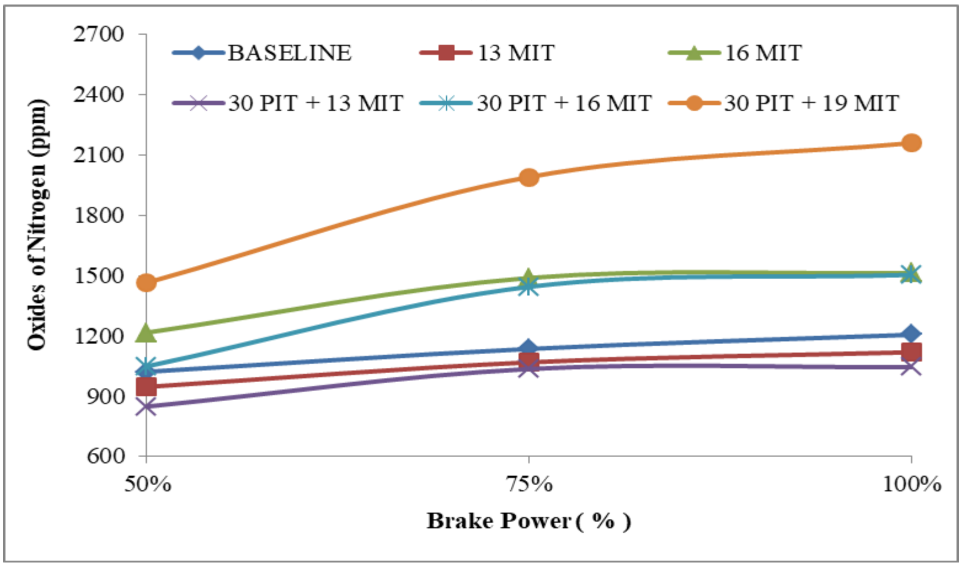

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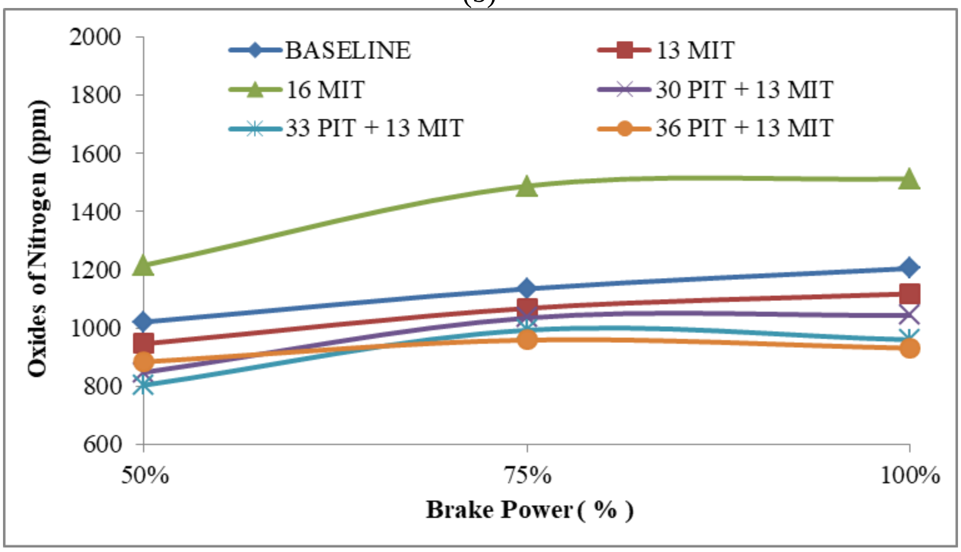


(c)

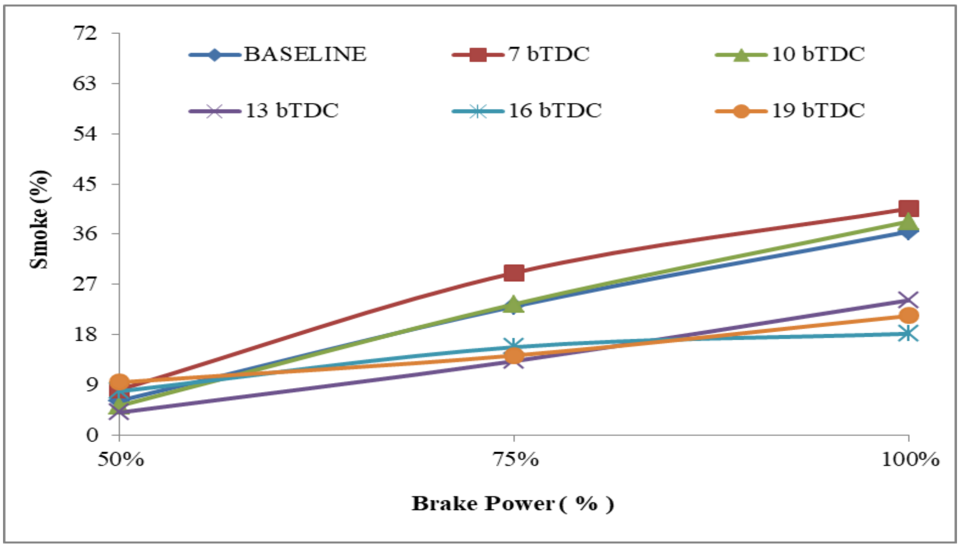
Fig 4.5. Variation of NOx emissions with Brake Power at different operating conditions.

4.2.4. Emissions of Smoke

Fig. 4.6 (a) shows the variation of smoke emission with respect to brake power at different injection timings. The formation of smoke commonly effects from the unfinished burning of the hydrocarbon fuel and the partly reacted carbon content material in the liquid fuel. The smoke emission generally increases with the increase in the brake power for all experiment conditions, which indicates that the exhaust stream is having higher particulate emissions. Increasing engine load consequences in an increase in fuel–air equivalence ratio and longer mixing controlled combustion phase, which leads to higher combustion temperatures in addition to lower oxygen concentration in the engine combustion chamber, consequently the smoke opacity increases. Retarded SOI timings increase smoke opacity because of decrease in-cylinder combustion temperatures and reduction in the time available for oxidation and re-burning of soot already formed for the duration of expansion stroke. Compared with baseline, at retarded timing of 13° bTDC showed optimum reduction in smoke emissions.

Fig.4.6 (b) shows the variation in smoke emissions with constant pilot and MIT at different loads. Here the injection is done twice first the pilot injection is done at 30^0 bTDC and latter the main injection is done at varying retarded injection timings. The in-cylinder temperature decreased with retardation in he injection timings. This is due to lower HRR with retarded injection. This low temperature results in low burning speed. With this resulted in reduction of oxygen concentration which ultimately forms rich mixtures. These rich mixtures increases the soot formation in the engine. Thus the soot emissions increases. At 30^0 PIT and 13^0 MIT, optimal timing between pilot and main injection supports better combustion and any shortened or prolonged strategy of split fuel injection may cause incomplete combustion thereby increasing the HC emissions.

Fig.4.6 (c) shows the variation in smoke emissions with constant MIT and different pilot injection at different loads. Here split injection strategy is utilized with the pilot injection advancing but the main injection being constant at 13^0 bTDC. As the injection timings are advanced that results in increasing of in cylinder temperature which in turn increases the oxygen concentration in the cylinder. As the oxygen concentrations are increased the mixture is no longer a rich mixture there burn perfectly and the smoke emissions are decreased. When compared to baseline and single injection timing, 30^0 bTDC PIT and 13^0 bTDC MIT showed optimum reduction in all emissions including smoke emissions.



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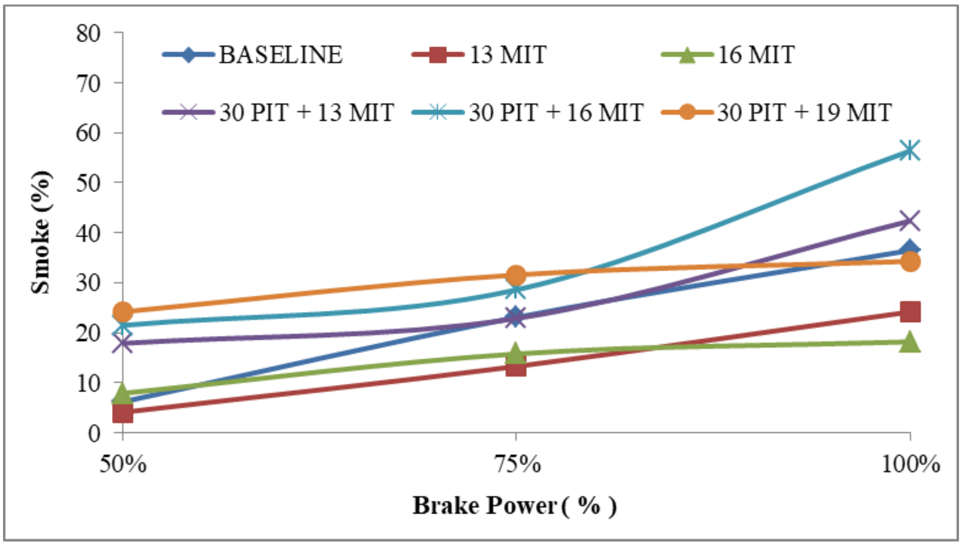

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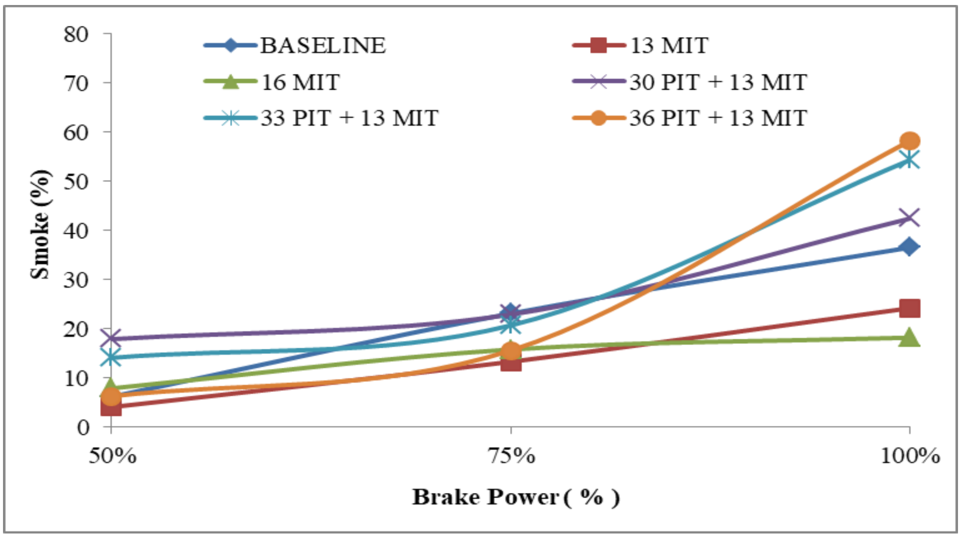

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Fig 4.6. Variation of smoke emissions with Brake power at different operating conditions.

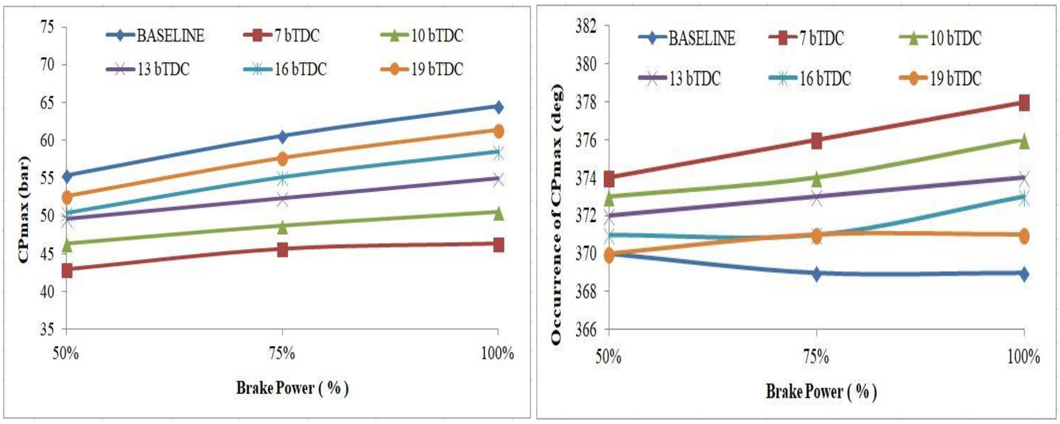
4.3. COMBUSTION CHARACTERISTICS

4.3.1. Cylinder Pressure

The measurement of in-cylinder pressure is an vital parameter for understanding engine combustion. The evaluation of in-cylinder pressure is used in finding various engine combustion parameters which includes heat release rate and pressure rise rate. The cylinder pressure and occurrence of CP max vs. crank angle for various SOI timings are shown in Fig.4.7 (a) for distinct engine loads. With increasing engine load, peak cylinder pressure increased and its position shifted away from top dead center (TDC) because of higher fuel quantity being burnt, which leads to longer combustion period consequently the pressure peak seems relatively later in the expansion stroke. When SOI became retarded and it came closer to TDC in compression stroke, ignition delay became shorter, which caused higher fuel fraction burning in diffusion combustion thereby lowering maximum cylinder pressure. Due to this shorter ignition delay, pressure peak become smaller and it also shifted away from TDC position in expansion stroke. Compared to baseline operation, retarded injection timings have resulted in lower peak cylinder pressure at all engine loads. The occurrence of peak pressures is also delayed i.e. it moves away from TDC with retarded injection timings.

cylinder pressure and occurrence of CP Max for different retarded Injection timings with pilot fuel injection as constant at various loads as shown in Fig 4.7 (b). It is seen that retarded the injection timing by pilot injection as constant maximum pressure was decreased because dwell period was increased. As we increased the load, cylinder pressure was increased because more fuel is accumulated in cylinder chamber. Compare to the baseline operation, retarded the injection timing with constant pilot injection have resulted lower peak cylinder pressure at all engine loads. Occurrence of peak pressure is moved towards the TDC.

Fig. 4.7 (c) clearly shows the effect of split injection compared to base line single injection. It is seen that there is a decrease in peak pressure and widening of the p-θ curve with the introduction of pilot fuel compared to single injection. With increasing the engine load cylinder pressure was increased because more amount of fuel is being burnt. Increasing separation between pilot and main injection pulses resulted in cooling of cylinder charge before the main injection. Advancing the pilot injection timing doesn’t change the pressure value its constant in all pilot injection conditions. Compared to baseline operation, advancing the pilot injection timings maintain main injection timing as constant have resulted in lower peak cylinder pressures at all engine loads. Occurrence of peak pressure is moves away from TDC with varying pilot injection timing.



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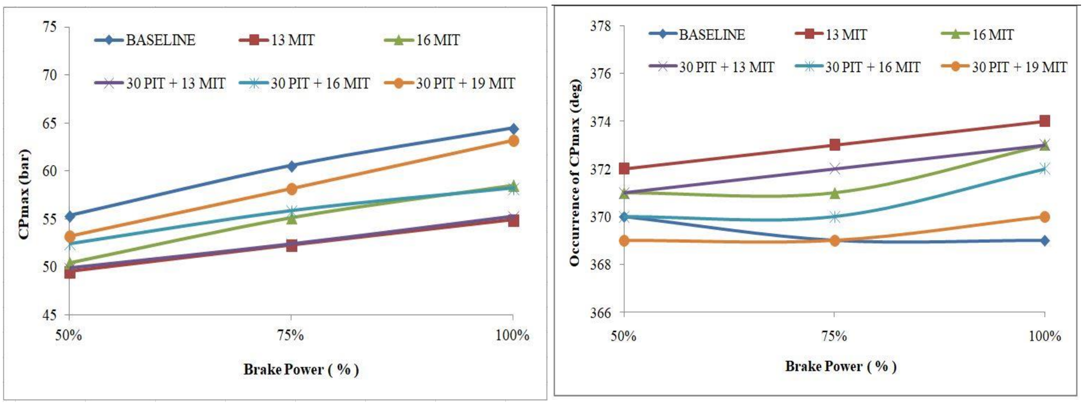

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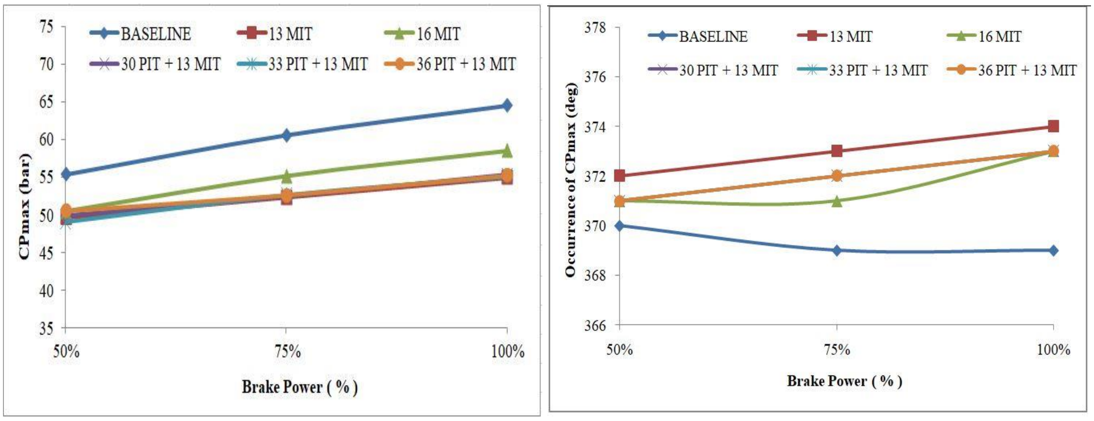


(c)

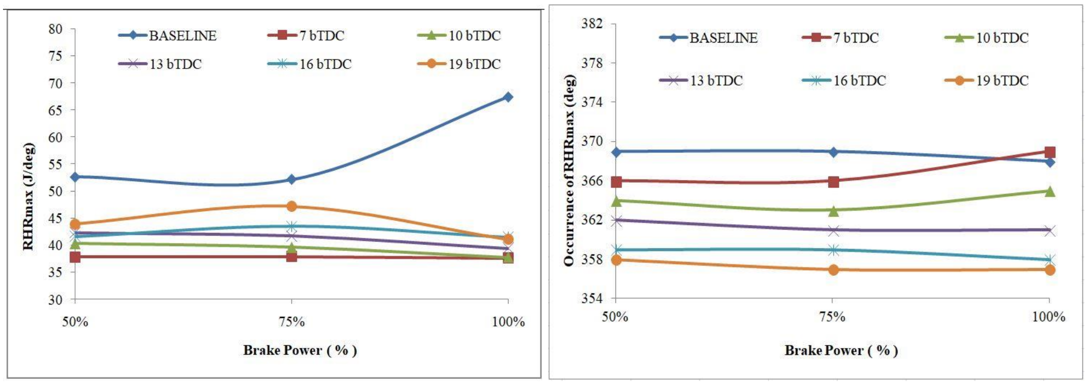
Fig 4.7. Variation of cylinder pressure and occurrence of peak pressure with Brake power at different operating conditions.

4.3.2. Heat Release Rate

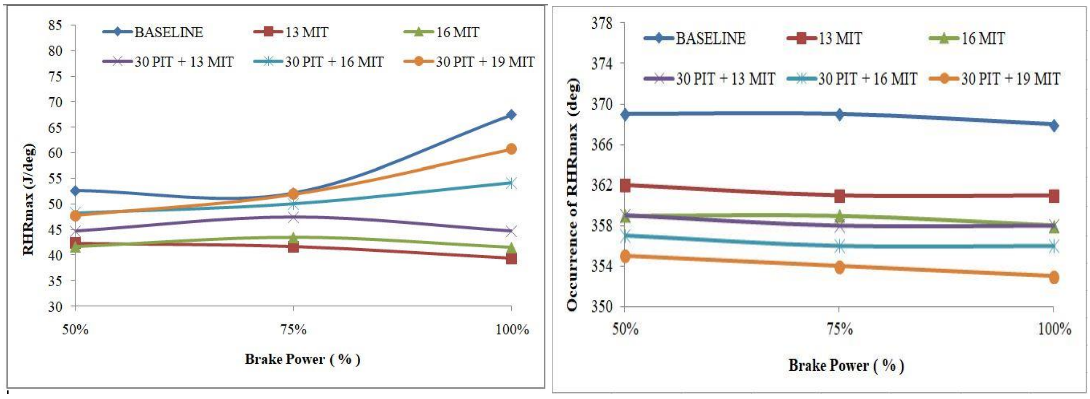
Fig.4.8 (a) shows the heat release rate at various SOI timings for various loads. The graph indicates two distinct levels of heat release. The primary is immediately after the SOI to a point, in which the heat release rate sharply drops. This is often because of combustion primarily within the premixed combustion phase. The second phase starts from the end of first phase (Premixed combustion) to the end of combustion and this can be called ‘mixing-controlled combustion phase’. This is usually a slower heat release phase among the two, therefore, it spreads over an extended combustion period and is basically controlled by the rate, at which, the fuel - air can blend together in the combustion chamber. At higher engine loads, the fraction of heat release occurring in the mixing controlled combustion phase is higher due to the fact the ignition delay is shorter for higher engine load. Therefore smaller fuel quantity is available in combustion chamber on the time of premixed combustion, which lowers the peak and also the crank angle role of this peak of heat release rate also shifts in the direction of TDC. The HRR peaks diminish due to the fact the injection timing is retarded from 19°bTDC and 7°bTDC. This is often due to the fact retarding the injection timing shortens the ignition delay duration and additionally the time to be had for air–fuel mixture is also less resulting in inferior combustion which led to decrease HRR peaks. Compared to baseline operation, retarded injection timings have exhibited smaller heat release peaks at all engine loads for low injection pressures. The occurrence of heat release peaks is also delayed i.e. moved away from TDC with retarded injection timings. However, heat release peaks and their occurrences improve at higher FIP for all engine loads. This can be attributed to increased fuel delivery and improved combustion at higher FIPs owing to improved atomization and quick fuel-air mixing

Fig.4.8 (b) shows the variation of rate of heat release at various retarded injection timings with pilot injection as constant and load conditions. Retarded the main injection timing by fixing the pilot injection as constant RHR maximum was decreased because the ignition delay was decreased .Increase the load increases the peak RHR. compared to the baseline operation, Retarded the main injection timing while maintaining constant pilot injection timing have exhibited very low peak RHR. The occurrence of RHR is moved towards the TDC.

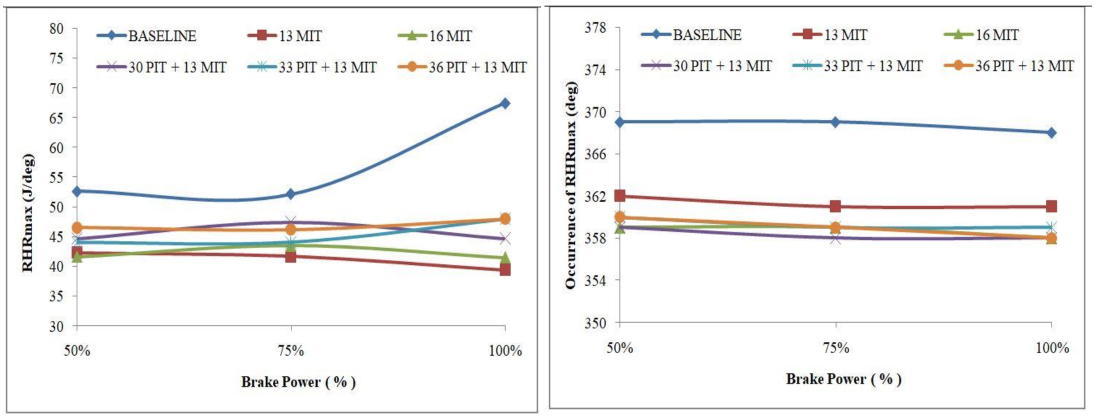
Fig.4.8 (c) show the variation of heat release rate with BP for different pilot injection timings. Higher heat release rate contribute to increase in in-cylinder temperature which favour NOx formation conversely lower heat release rates contribute to reduction in the formation of oxides of nitrogen. It is observed from the graphs that with the introduction of pilot injection the peak heat release rate is reduced. Advancing the pilot injection timing RHR is increased because dwell period is increased so fuel mixtures with air correctly. compared to baseline operation, advancing pilot fuel injection timing while maintaining constant timing of the main fuel injection have resulted lower peak rate of heat release at all engine load conditions. The occurrence of rate of heat release was shifted towards Top dead center (into compression stroke).



(a)



(b)



(c)

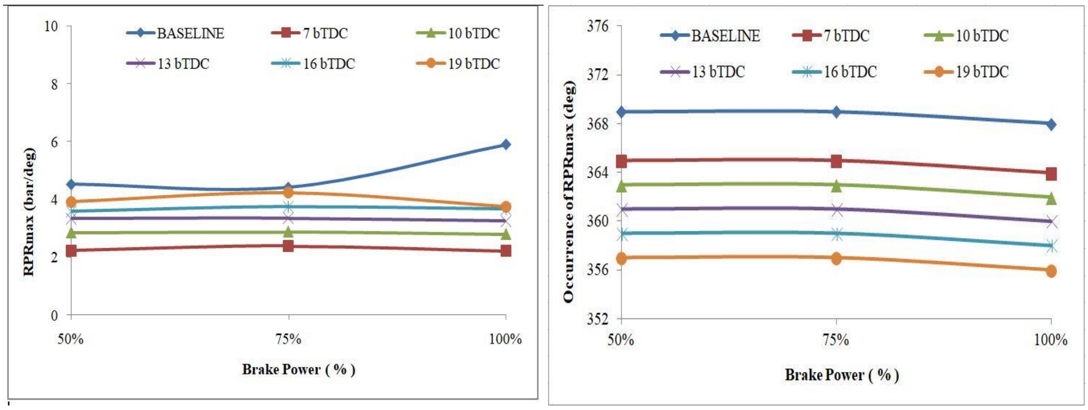
Fig 4.8. Variation of heat release rate with Brake power at different operating conditions.

4.3.3. Pressure Rise Rate

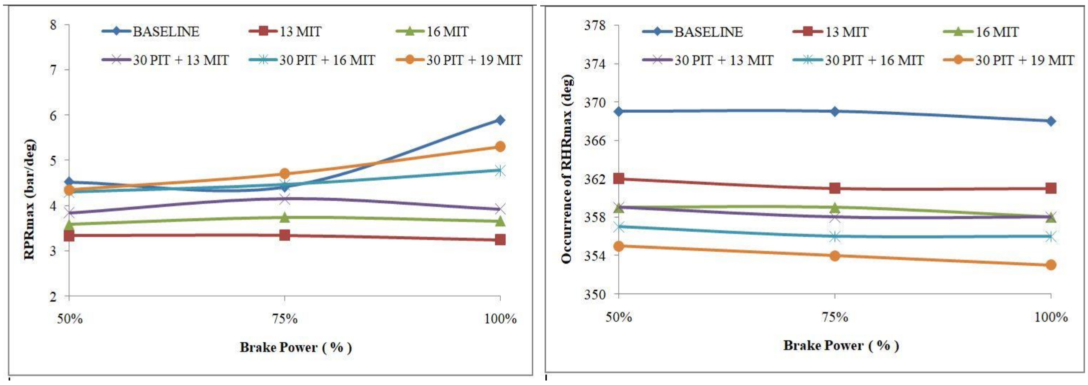
The rate of pressure rise is a parameter, which offers information about the rate of force transfer because of in-cylinder combustion pressure exerted by burning and expanding gases. Fig.4.9 (a) indicates the rate of pressure rise for different SOI timings at various engine loads. The rate of pressure rise reaches its maxima during premixed combustion phase because of rapid combustion and very fast premixed heat release. After reaching the maxima, it reduces in the expansion stroke because of mixing controlled combustion, in which the combustion is relatively slower in addition to increase in combustion chamber volume because of movement of piston in expansion stroke. As the engine load increases, a relatively higher internal cylinder temperature is observed, which reduces ignition delay. This results in relative earlier ignition of premixed charge; hence, there will be lesser fuel accumulation with inside the combustion chamber because of shorter ignition delay, leading to reduction in pressure rise fee with increasing engine load. Peak of pressure rise rate shifts away from the TDC because of relatively slower combustion and heat release in predominantly mixing controlled combustion section at higher engine loads. For retarded SOI timings, the rate of pressure rise become decreased. Due to the ignition delay period is shorter. Compared to baseline operation, retarded injection timings resulted in lower peaks of PRR, particularly at part loads. The occurrence of pressure rise rate shifts away from the TDC because of relatively slower combustion and heat release in predominantly mixing controlled combustion section at higher engine loads.

Fig.4.9 (b) displays the variation of rate of pressure rise at various retarded injection timings with pilot injection as constant and load conditions. With retarded injection, the gap between pilot and main injections (Dwell period)is increased so that ignition delay decreases with retarded fuel injection timing which lowers the time available for start of combustion yielding lower heat release rates. This in turn results in lower pressure rise rates at retarded injection timings with pilot injection as constant. Compare to baseline operation, RPR was decreased for injection timings. The occurrence of RPR was moved towards the TDC because of main ignition delay was decreased by introducing the pilot injection.

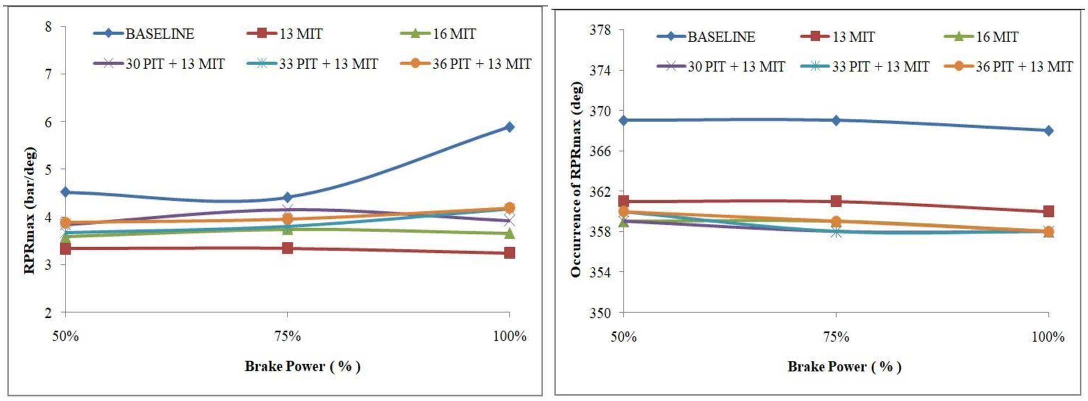
Fig.4.9 (c) indicates the rate of pressure rise for different SOI timings with pilot injection at various engine loads. Advancing pilot fuel injection timing while maintaining constant timing of the main fuel injection increases the Rate of pressure rise for all loads. Advancing pilot fuel injection produces a longer pilot ignition delay resulting in an intermediate main ignition delay. Advancing the pilot fuel injection increase the RPR. Compared to baseline operation, advancing the pilot injection timings resulted in lower peaks of PRR, particularly at full load. The occurrence of peak RPR is also moved towards TDC with advancing pilot injection timings.



(a)



(b)



(c)

Fig 4.9. Variation of Rate of Pressure Rise (RPR) and occurrence of RPR with Brake power at different operating conditions.

CHAPTER 5

5. CONCLUSIONS AND FUTURE SCOPE

5.1. CONCLUSIONS

The effects of varying the pilot injection timing and pressure on NOx, Smoke, CO and HC emissions in CRDI diesel engine at part load to full load condition has been studied in this work. Results showed that potential benefits can be achieved in reducing harmful emissions without significant impact on the energy consumption of the engine. The main conclusions of this study can be summarised as follows.

1. Introducing pilot injection, maintaining the injection pressure at 400 bar and varying the injection timing was found to have significant effect not only on performance parameters like brake specific energy consumption, cylinder pressure, heat release rate but also on HC, CO, CO_2, NO and smoke emissions.
2. Retarded the main injection timing (MIT), Brake Thermal Efficiency was decreased compared to baseline. The best performance was observed at 13^0 bTDC. By using the split injection at 30^0 PIT + 13^0 bTDC MIT was improved BTE then 13^0 bTDC MIT. BSFC for retarded injection timing is low at higher loads.
3. Retarding of main injection can reduce NOx emissions. NOx compared with baseline is minimum at 7^0 bTDC retarded MIT. For best performance and less emissions optimum value of 13^0 bTDC is chosen as the best retarded MIT. Multiple injection helps in reduction of both smoke and NOx, but only a slight increase in BTE can be observed. NOx emission was found to be lower with a maximum reduction at 30^0 PIT +13^0 MIT bTDC and at a pressure of 400 bar with spilt injection strategy compared to the conventional single mode. However, the emission value was found to be little higher at 13^0 bTDC that is at retarded single injection mode. In further study, at constant MIT and varying PIT the NOx emissions were drastically reduced at 36^0 PIT + 19^0 MIT split injection mode with an increase in HC, CO and smoke emissions. Hence, it can be concluded that, 30^0 PIT + 13^0 MIT bTDC injection angle and 400 bar pressure gave the optimum reduction in NOx emission. Split

injection with smaller amount of fuel injected in first pulse is observed to slow down premixed phase of combustion with a significant reduction in NOx emissions.

4. CO and HC emissions were found to be minimum at 16^0 MIT in retarded single injection mode when compared to baseline. In split injection mode, the CO and HC emissions were found to be minimum at 30^0 PIT and 19^0 MIT. Smoke emission is lower at 30^0 PIT and 13^0 MIT.

5.2. FUTURE SCOPE

In the present study, experiments were conducted on a modified single-cylinder CRDI engine by implementing different retarded injection timings and split injection strategy at a fixed injection pressure using diesel as fuel. Further investigations can be carried out in the following aspects.

- Optimum combination of injection pressure and injection timing is to be identified resulting in improved engine out emissions without affecting the engine performance.
- For the identification of optimum dwell period and pilot injection quantity with different combinations of pilot and main injections by varying pilot injection quantity.
- Strategies like exhaust gas recirculation (EGR), supercharging can also be implemented.

CHAPTER 6

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