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

BACHELOR OF TECHNOLOGY
IN
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CERTIFICATE

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ABSTRACT

The purpose of this study is to improve exhaust emissions with little or no deterioration in performance by applying retarded injections and various split-injection strategies to determine the optimal injection that improves fuel efficiency and reduces exhaust emissions, compared with single-injection combustion. 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). It is established that retarded injection timing and split injection timings reduce NOx and smoke emissions of a typical diesel engine. Hence engine testing was carried out using diesel fuel at 1500 rpm, under 50% to 100% of full load brake power, for retarded and split injection timings (varying injection interval) at 300 bar as constant fuel injection pressure (FIP).

Experimental results show that retarding the fuel injection timing enhanced the fuel economy by deteriorating the brake-specific fuel consumption (BSFC) mainly the BSFC is higher at high engine load and smoke, NO_X were improved. The results at 13^0 bTDC offered significant reduction in NOx and smoke with little deterioration of BTE compared to base condition (23^0 bTDC). Advancing SOMI from 13^0 bTDC to 16^0 bTDC in the intervals of 3^0 and SOPI at 30^0 bTDC shows good results. Best results were obtained at the 30^0 bTDC (SOPI) and 13^0 bTDC (SOMI) with considerable reduction in the NO_X and BTE is improved as compared with the single injection of 13^0bTDC. Advancing the SOPI from 30^0 bTDC to 36^0 bTDC in the intervals of 3^0 and SOMI as constant at 16^0 bTDC was revealed the no improvement of emissions. According to the analysis, a large injection interval and main injection timing around top dead centre (TDC) improve the BSFC and emission characteristics in split-injection diesel combustion.

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CHAPTER 1

INTRODUCTION

Diesel engine is any internal combustion engine in which air is compressed to a sufficiently high temperature to ignite diesel fuel injected into the cylinder, where combustion and expansion actuate a piston. It converts the chemical energy stored in the fuel into mechanical energy, which can be used to power many automobiles.

1.1 ORIGIN OF ENGINE:

The invention of the diesel engine goes way back – all the way to the 1890s. Since their introduction, they have remained one of the most common engines used in power generation applications. They have been useful in a variety of industries and functionalities.

In the 1870s, steam was the main supplier of power for factories and trains. Steam-powered cars were even being produced alongside those using internal combustion engines. Enter Rudolf Diesel, who invented diesel engines.

Fig1.1 diesel engine

Diesel was a student learning about thermodynamics at the time, and he got the idea for creating an engine that would be highly efficient and convert the heat it generated into power. He got to work developing what would become the diesel engine.

He set up his first shop in 1885 to start the development of this new engine and to put his theories into practice. One of his hypotheses was that higher amounts of compression would lead to higher efficiency and power.

Diesel received patents for his designs during the 1890s. The first diesel engine prototype was built in 1893, though the first engine test was unsuccessful,

1897, Diesel produced successful results after many improvements and tests. In February of that year, he was able to show an efficiency of 26.2% with the engine. Compared with the steam engine popular at the time, the engine Diesel had developed was more efficient by 16.2%.

1.2 WORKING OF DIESEL ENGINE:

Diesel engines, like gasoline engines, are considered to be 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. These include:

- **Intake stroke:** Air comes into the cylinders through the intake valve, and pistons move down.
- **Compression stroke:** The pistons move up, compressing the air.
- **Combustion stroke:** Fuel is injected and ignited at a specific time, forcing the pistons down again.
- **Exhaust stroke:** As the pistons move back toward the top, exhaust created during the combustion process is pushed out.

The heat of the compressed air is what ignites fuel in a diesel engine. Modern diesel engines are up to twice as efficient as gasoline engines, meaning you can travel farther on the same amount of fuel.

1.3 DIESEL ENGINE VS PETROL ENGINE:

The Diesel versus Petrol engine comparison has been ongoing since the time of their inception. The primary difference is that petrol engines use spark plugs to ignite the air-fuel mixture, while diesel engines rely on heavily compressed air without any spark plugs. So, in diesel engines, air is very heavily compressed, while in petrol engines, the compression ratio is generally much lower.

Diesel engines have lower specific fuel consumption than gasoline engines, so they are more economical, they have a better operating characteristic, i.e. they change the number of revolutions very little with the change of load.

Diesel engines provide more efficiency by using 15-20% less fuel compared to petrol engines. The low-end torque of diesel engines provide a much better highway driving experience.

Diesel engines have always been more fuel efficient, durable and delivered more torque than petrol engine. Typically, they contain less toxic pollutants but they did have higher quantities of carbon (soot) in their exhaust than gasoline engines.

1.4 EMISSIONS FROM DIESEL ENGINE:

Diesel engines are more widely used than petrol engines due to their low maintenance cost, energy efficiency, high durability and reliability. Although they have many benefits, they have a significant impact on environmental pollution issues worldwide which can cause serious environmental and health problems.

For ideal thermodynamic equilibrium, full combustion of diesel fuel produces only CO_2 and H_2O in the combustion chambers of the engine. However, many factors such as air-fuel ratio, ignition time, combustion chamber turbulence, combustion form, air-fuel density, combustion temperature, etc. put it out of question and many harmful products are produced during combustion. The most important harmful products are CO, HC, NOx and PM.

1.4.1 PARTICULATE MATTER (PM):

Particulate matter—most commonly associated with diesel engines—is responsible for the black smoke traditionally associated with diesel powered vehicles. The existing medical research suggests that PM is one of the major harmful emissions produced by diesel engines. Exhaust from trucks, buses, trains, ships, and other equipment with diesel engines contains a mixture of gases and solid particles. Some particles, such as dust, dirt, soot, or smoke, are large or dark enough to be seen with the naked eye. Others are so small they can only be detected using an electron microscope. These particles come in many sizes and shapes and can be made up of hundreds of different chemicals.

Particulate matter contains microscopic solids or liquid droplets that are so small that they can be inhaled and cause serious health problems. Some particles less than 10 micrometres in diameter can get deep into your lungs and some may even get into your bloodstream. Of these, particles less than 2.5 micrometres in diameter, also known as fine particles pose the greatest risk to health.

Fine particles are also the main cause of reduced visibility (haze) in parts of the United States, including many of our treasured national parks and wilderness areas.

1.4.2 CARBON MONOXIDE (CO):

Carbon monoxide is formed as a result of incomplete combustion in which the oxidation process does not take place completely. This concentration is largely dependent on the air / fuel mixture and is classified as a rich mixture where the excess-air factor (λ) is less than 1.0. This is especially true during engine start and instantaneous acceleration, which requires rich mixing. In enriched compounds, due to the absence of air and reactant concentrations, not all carbon is converted to CO and CO is not converted to concentration. Although CO is produced during operation in rich mixes, even a small fraction of CO is released under lean conditions due to chemical kinetic effects.

Diesel engines are lean combustion engines with a consistently high air-fuel ratio. Therefore, CO formation in diesel engines is very low. However, if the droplets in the diesel engine are too large or there is enough turbulence or swirl in the combustion chamber, CO will be produced.

Carbon monoxide is an odourless and colourless gas. In humans, the CO in the air is absorbed by the lungs and circulated in the bloodstream. It binds to hemoglobin and inhibits its ability to transfer oxygen. Depending on the concentration of CO in the air, it can lead to asphyxia, which affects the function of various organs, resulting in impaired concentration, slow reactions and confusion.

1.4.3 NITROGEN OXIDES (NOX):

It refers to nitrogen oxides. The purists would say that it refers to nitric oxide (NO) and nitrogen dioxide (NO_2). Nitrogen oxides are produced due to **High temperature combustion of fuels** where the temperature is hot enough to oxidize some of the nitrogen in air to NOx gases. This includes burning hydrogen, as it burns at a very high temperature. As the diesel engines operate at a higher temperature and pressure than petrol engines. These conditions favour the production of NOx gases. The quantity depends on the volume and duration of the hottest part of the flame.

NOx has direct and indirect effects on human health. It can cause breathing problems, headaches, chronically reduced lung function, eye irritation, loss of appetite and corroded teeth. Indirectly, it can affect humans by damaging the ecosystems they rely on in water and on land—harming animals and plants.

NOx emissions can be reduced by lowering the combustion temperature, typically by Exhaust Gas Recirculating (EGR). Some exhaust gas is cooled and injected back into the combustion chamber. There is less oxygen in the exhaust gas because some has been consumed by previous combustion, so there is not as much to feed the flame. The exhaust gas also has a higher heat capacity than air, so it takes longer to heat up.

1.4.4 HYDROCARBONS (HC):

Hydrocarbon emissions are composed of unburned fuels as a result of insufficient temperature which occurs near the cylinder wall. At this point, the air-fuel mixture temperature is significantly less than the centre of the cylinder. Hydrocarbons consist of thousands of species, such as alkanes, alkenes, and aromatics. They are normally stated in terms of equivalent CH_4 content.

Diesel engines normally emit low levels of hydrocarbons. Diesel hydrocarbon emissions occur principally at light loads. The major source of light-load hydrocarbon emissions is lean air-fuel mixing. In lean mixtures, flame speeds may be too low for combustion to be completed during the power stroke, or combustion may not occur, and these conditions cause high hydrocarbon emissions.In Diesel engines, the fuel type, engine adjustment, and design affect the content of hydrocarbons. Besides, HC emissions in the exhaust gas depend on irregular operating conditions. High levels of the instantaneous change in engine speed, untidy injection, excessive nozzle cavity volumes, and injector needle bounce can cause significant quantities of unburned fuel to pass into the exhaust.Unburned hydrocarbons continue to react in the exhaust if the temperature is above 600°C and oxygen present, so hydrocarbon emissions from the tailpipe may be significantly lower than the hydrocarbons leaving the cylinder.Hydrocarbons have harmful effects on the environment and human health. With other pollutant emissions, they play a significant role in the formation of ground-level ozone. Vehicles are responsible for about 50% of the emissions that form ozone. Hydrocarbons are toxic with the potential to respiratory tract irritation and cause cancer. Retarding SOI timings lowers the in-cylinder pressure and temperature during combustion, which in turn increases the unburnt hydrocarbon emissions. At higher FIPs, BSHC emissions increased sharply, when the SOI timings were close to TDC. This was possibly due to piston wall impingement of the fuel sprays because, during the fuel injection, the piston remains very close to the injector tip.

1.5 TECHNIQUES FOR CONTROL OF EMISSIONS:

Diesel particulate and NO_x emission cause several serious health problems; therefore, it is necessary to reduce these emissions from the tailpipe. In the past decades, significant technological advancements have been made in the field of engine emission control. In modern diesel engines, smarter electronic fuel injection strategies are being employed.

Control of engine emissions can be done by two ways:

1. Active control techniques
2. Passive control techniques.

Active control techniques are those which restrict the formation of the pollutants in the combustion chamber itself. Passive control techniques refer to after-treatment devices.

Active control techniques include advancement in the combustion chamber design, use smarter electronic fuel injection system, exhaust gas recirculation, high-pressure multi-fuel injection with precise injection timing, homogenous charge compression ignition, etc.

Although active control techniques are able to reduce the emission up to some extent, but in order to meet the modern emission regulations, passive techniques are also required in addition to active techniques. Passive control technique involves after-treatment devices like diesel oxidation control, diesel particulate trap, NO_x absorber, selective catalytic reduction.

1.5.1 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 1.2. Catalytic Converters

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 cannot work to its optimum efficiency, and to control emissions. The catalytic converter is the most important emission control device that destructs harmful emissions.

1.5.2 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.

1.5.3 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.

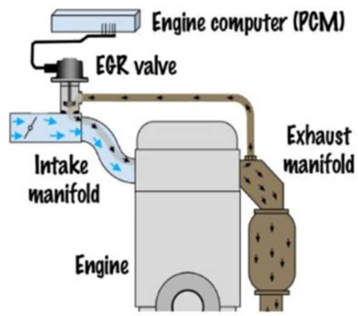

Fig1.3 Exhaust gas recirculation

1.5.4 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 1.4 Diesel particulate filter

1.5.5 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.

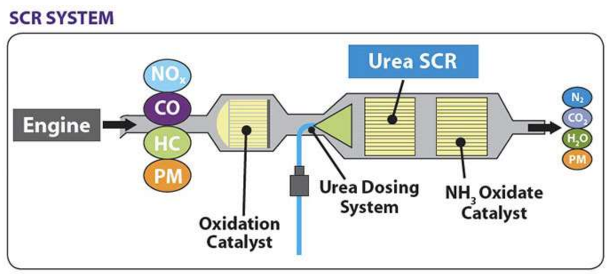

Fig 1.5 Selective catalyst reduction

1.6 INJECTION PRESSURE:

Fuel injection pressures in diesel engine plays an important role for engine performance obtaining treatment of combustion. The present diesel engines such as fuel direct injection, the pressures can be increased about 100 – 200 Mpa bar in fuel pump injection system

In present diesel engines, fuel injection systems have designed to obtain higher injection pressure. So, it is aimed to decrease the exhaust emissions by increasing efficiency of diesel engines. When fuel injection pressure is low, fuel particle diameters will enlarge and ignition delay period during the combustion will increase. This situation leads to increase pressure. Engine performance will be decrease since combustion process goes to a bad condition. When injection pressure increased of fuel particle diameters will become small. Since formation of mixing of fuel to air becomes better during ignition period, engine performance will be increase. If injection pressure is too higher, ignition delay period becomes shorter. Possibilities of homogeneous mixing decrease and combustion efficiency decreases.

1.7 INJECTION TIMING:

In an internal combustion engine, thermal energy transfers into mechanical energy. The created power moves an engine's pistons, therefore, moving the crankshaft. Thermal energy comes from the combusted air-fuel mixture inside the cylinder. A piston moves inside the cylinder from the bottom dead centre to the top dead centre during combustion.

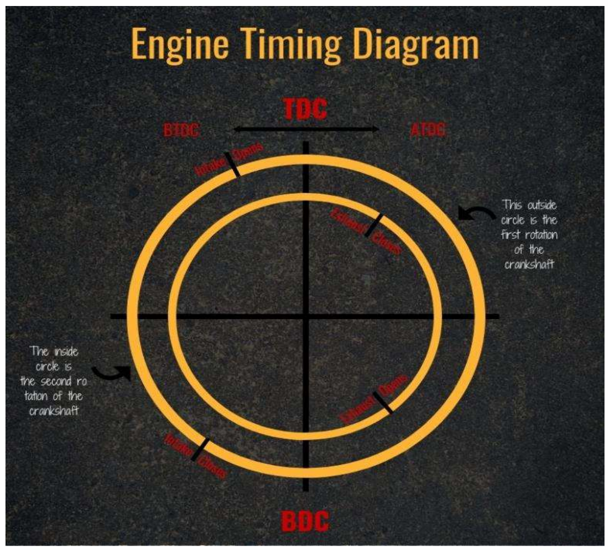

Fig 1.6 Injection timing

Injection timing, also called spill timing, is the moment when diesel fuel enters the cylinder during the combustion phase. When you adjust the timing, you can alter when the engine injects the fuel, therefore changing when combustion occurs.

An injection pump is often driven indirectly from the crankshaft by chains, gears or a timing belt that also moves the camshaft. The timing of the pump determines when it will inject fuel into the cylinder as the piston reaches the BTDC point.
There are a few terms you'll need to know to understand how the piston moves inside the cylinder, including:

– Top Dead Centre (TDC): Top dead centre is when the piston is at the top Of the cylinder, positioning itself farthest from the crankshaft.

– Bottom Dead Centre (BDC): Bottom dead centre is when the piston is closest to the crankshaft at the cylinder's lowest point.

– Before Top Dead Centre (BTDC): Before top dead centre is the point right before the piston reaches the highest area of the cylinder.

Advantages from adjusting fuel injection timing:

– Boosted engine power capabilities

– Higher peak cylinder pressure

– Lower exhaust temperatures

– Higher NOx emissions

– Increased fuel efficiency

1.8 FUEL INJECTION SYSTEM:

The fuel injection system lies at the very heart of the 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.

The purpose of the fuel injection system is to deliver fuel into the engine cylinders, while precisely controlling the injection timing, fuel atomization, and other parameters

1.8.1 SINGLE POINT INJECTION SYSTEM:

Single point fuel injection system is the type of fuel injection system that uses a single fuel injector for mixing of the fuel. It has only one injector that injects the fuel before entering into the intake manifold.

In this system, the fuel is mixed with fuel before the throttle valve. The single-point fuel injection system is also known as throttle body injection.

In a single-point fuel injection system, the fuel injector is arranged before the throttle body.

The amount of fuel to be injected is decided by the engine control unit. The engine control unit takes the input from different sensors and decides the amount of fuel to be supplied for the injection.

The fuel injector sprays the fuel for mixing with the flow of air and this air-fuel mixture enters the intake manifold.

The intake manifold further distributes the mixture to all cylinders.

1.8.1.1 ADVANTAGES OF SINGLE POINT FUEL INJECTION SYSTEM

1.Simple construction.

2. Accurate fuel supply.
3. Easy maintenance.
4. It uses only single injector.
5. Reliable operation.

1.8.1.2 DISADVANTAGES OF SINGLE-POINT FUEL INJECTION SYSTEM:

1. Uninform fuel supply to all cylinders.
2. Less efficient.
3. It wets the intake manifold by forming a layer of fuel on the intake manifold.
4. Lower fuel economy.

1.8.2 MULTI-POINT FUEL INJECTION SYSTEM:

While conventional fuel injection systems employ a single injection event for every engine cycle, newer systems can use multiple injection events. One or more injections before the main injection, pre-injections, provide a small amount of fuel before the main injection event.

The Multi-Point Fuel Injection system is a way of injecting the fuel in an internal combustion engine through multiple ports located on the intake valve of every cylinder the motor has. These ports work together to deliver the optimum quantity of fuel at the right time to every cylinder. In all, there are three varieties of MPFI units – Batched, Simultaneous, and Sequential.

In the first kind of Multi-Point Fuel Injection system, the fuel is released to the cylinders by the ports in batches without getting their intake stroke together. In the Simultaneous MPFI systems, the fuel is released in all the cylinders of the engine simultaneously, while in the sequential type, the fuel release is timed to take place at the same time as the intake stroke for each cylinder of the engine.

The term split injection is occasionally used to refer to multiple injection strategies where a main injection is split into two smaller injections of approximately equal size or into a smaller pre-injection followed by a main injection.

1.8.2.1 Advantages of MPFI System:

1. Improvement in Fuel Efficiency
2. Lower Carbon emissions
3. Improvement in Engine Performance
4. Improvement in Engine Refinement

1.8.3 SPLIT INJECTION:

The major pollutants from diesel engines are NOx and soot. NOx and soot emissions are of concerns to the international community. They have been judged to pose a lung cancer hazard for humans as well as elevating the risk of non-cancer respiratory ailments. Stringent exhaust emission standards require the simultaneous reduction of soot and NOx for diesel engines; however, it seems to be very difficult to reduce NOx emission without increasing soot emission by injection timing. The reason is that there always is a contradiction between NOx and soot emissions when the injection timing is retarded or advanced.

Split injection has been shown to be a powerful tool to simultaneously reduce soot and NOx emissions for direct injection & indirect injection 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 the recent years, the main studies about the effect of the split injection on the combustion process and pollution of DI and IDI diesel engines showed that the nitric oxide and the particulates could be reduced by over 83 % and almost 24 %, respectively while maintaining a reasonable value of specific fuel consumption.

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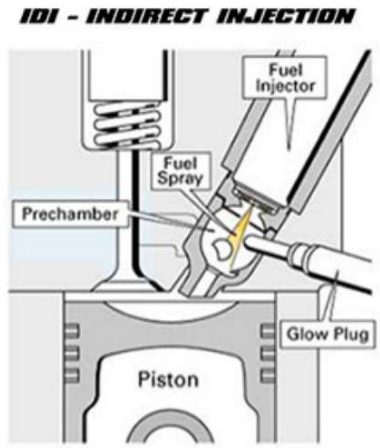
<td>
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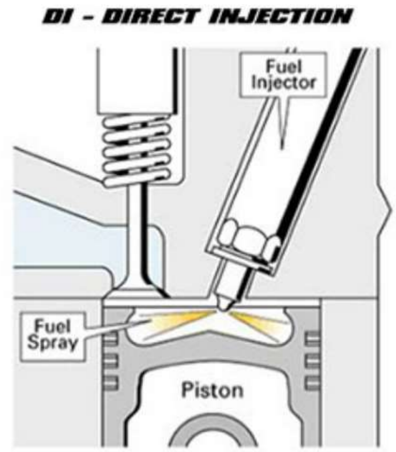
<td>
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Fig 1.7 Indirect injection diesel vs direct injection diesel

1.8.4 IDI - INDIRECT INJECTION DIESEL:

IDI diesel engines utilize a pre-combustion chamber, typically referred to as a swirl chamber or prechamber. Fuel is injected into the prechamber where it rapidly mixes with air and auto ignition occurs. As the flame front expands in the pre-chamber, it forces fuel to enter the combustion chamber rapidly, effectively mixing the fuel with air in the cylinder and atomization is achieved. The glow plug is also located in the prechamber, and the shape of the pistons in an IDI tend to resemble those of a gasoline engine..

1.8.5 DI - DIRECT INJECTION DIESEL:

DI diesel engines inject fuel directly into the combustion chamber, right into the top of the piston. The pistons on a DI engine typically have a bowl or cup machined into them that the fuel is directed into. DI engines operate at higher injection pressures and therefore more complete atomization occurs, meaning these engines do not require a prechamber to ensure proper diffusion of the fuel into the air.

1.9 COMMON RAIL DIRECT INJECTION:

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.

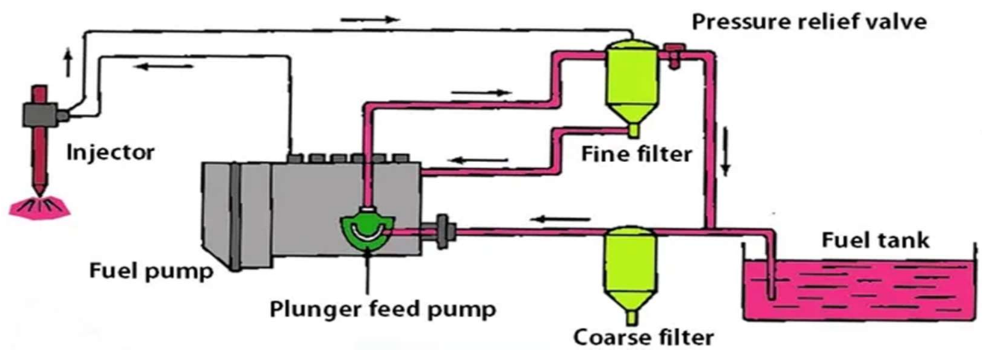


Fig 1.8 Common rail direction

Regular diesel direct fuel-injection systems have to build up pressure for every new injection cycle. 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.

1.9.1 WORKING OF CRDI SYSTEM OR COMMON RAIL DIRECT INJECTION:

1.As you can see in the diagram of the CRDI system, the high-pressure pump is used to supply fuel to the accumulator or the header from the fuel tank. In case pressure in the accumulator increases beyond the limit, the high-pressure relief valve which is connected to the accumulator helps to reduce the pressure.

2. Now, this fuel from the accumulator supplied to engine cylinders using fuel lines with the help of solid injectors.

3.Another spring-loaded high-pressure relief valve used to maintain the constant pressure in the system for smooth operations. It also returns the extra fuel of the accumulator to the fuel tank.

4.In the diagram, you can see the needle valve. It is used to control the opening and closing of the nozzle while it sprays the fuel into the cylinders. The upward and downward motion of the nozzle is measured by the cam.

5. Cam is connected to the spring with the help of a rocker arm and lever. During the dwell period of the cam, spring with the help of the needle valve prevents the injection of the fuel into the cylinder.

6.The wedge plays the main role in this system. It controls the amount of fuel to be injected into the cylinder in accordance with the power required for the engine. The wedge is operated by a governor, or it can be operated manually as per requirement.

1.9.2 COMPONENTS OF COMMON RAIL DIRECT INJECTION SYSTEM:

1. High Pressure Fuel Pump
2. Common Fuel Rail
3. Injectors
4. Engine Control Unit

1.9.2.1 HIGH-PRESSURE PUMPS:

The high-pressure pump compresses the fuel and supplies it in the required quantity. It constantly feeds fuel to the high-pressure reservoir, thereby maintaining the system pressure. The required pressure is available even at low engine speeds, as pressure generation is not linked to the engine speed. Most common rail systems are equipped with radial piston pumps. Compact cars also use systems with individual pumps which operate at a low system pressure.

1.9.2.2 COMMON FUEL RAIL:

Common rail is a fuel injection system found in modern diesel engines. Common rail systems provide a level of flexibility which can be exploited for class leading emission control, power, and fuel consumption.

1.9.2.3 INJECTORS:

The injector in a common rail system consists of the nozzle, an actuator for Piezo injectors or a solenoid valve for solenoid valve injectors, as well as hydraulic and electrical connections for actuation of the nozzle needle.

1.9.2.4 ENGINE CONTROL UNIT:

The Engine Control Unit is a central part of the Engine Management System, which is virtually the 'Brain' of the engine. It plays an important role in collecting, analyzing, processing, and executing the data.

1.9.3 ADVANTAGES:

1. CRDI engines are advantageous in many ways. Cars fitted with this new engine technology are believed to deliver 25% more power and torque than the normal direct injection engine.

2. It also offers superior pick up, lower levels of noise and vibration, higher mileage, lower emissions, lower fuel consumption, and improved performance. 3. In India, diesel is cheaper than petrol and this fact adds to the credibility of the common rail direct injection system.

1.9.4 DISADVANTAGES:

Like all good things have a negative side, this engine also has few disadvantages. The key disadvantage of the CRDI engine is that it is costly than the conventional engine. The list also includes high degree of engine maintenance and costly spare parts. Also, this technology can't be employed to ordinary engines.

1.9.5 APPLICATIONS:

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.

CHAPTER 2

LITERATURE REVIEW

2.1 STUDIES ON SPLIT OR MULTIPLE INJECTION:

EhlesIkog et al. [1] Investigated the effect of split injection on the emission formation and engine Performance of a heavy-duty DI diesel engine by KIVA-III code. The results revealed that Reductions in NOx emissions and brake-specific fuel consumption were achieved for short Dwell times whereas they both were increased when the dwell time was prolonged.

Verbiezen et al. [2] Investigated the effect of injection timing and split injection on NOx Concentration in a DI diesel engine experimentally. The results showed that advancing the Injection timing causes NOx increase. Also, maximum rate of heat release is significantly reduced by the split injection. Hence, NOx is reduced significantly.

Chryssakis et al. [3] Studied the effect of multiple injections on combustion process and Emissions of a DI diesel engine by using the multidimensional code KIVAIII. The results indicated that employing a post-injection combined with a pilot injection results in reduced soot formation; while the NOx concentration is maintained at low levels.

Jafarmadar et al. [4] Studied the effect of split injection on combustion and pollution Of a DI diesel engine by Computational Fluid Dynamics (CFD) code. The results show that 25% of total fuel injected in the second pulse, reduces the total soot and NOx emissions Effectively in DI diesel engines. In addition, the optimum delay dwell between the two Injection pulses was about 25ºCA.

Atul Dhar et al. [5] Investigated the effect of pilot injection on performance, emissions and combustion characteristics of Karanja biodiesel fueled CRDI engine. According to them pilot and post injections are being used in modern diesel engines for improving engine performance in addition to meeting stringent emission norms. Biodiesel produced from different feedstocks is gaining global recognition as partial replacement for mineral diesel in compression ignition (CI) engines. In this study, 10%, 20% and 50% Karanja biodiesel blends were used for investigation of pilot injections, injection pressures and injection timings on biodiesel blends. . At 500 bar BSRC is lowest at -15°CA and -12°CA, SOMI timing is lower at -21°CA and -18° CA. At 100 bar BSFC was lowest at -9° at different SOPI timings, BSNOx emissions at a fixed SOMI timing were CA and -6°CA SOMI timing. BSNOx emissions were higher for 1000 bar FIP in comparison to 500 bar FIP. Maximum in-cylinder pressure at 1000 bar FIP is higher when compared to 500 bar FIP at same SOPI and SOMI timings. Lowest BSCO emissions for all test fuels were observed at 18°CA SOPI and -6°CA SOMI timings at 1000 bar FIP.

Nivin Chacko et al. [6] Worked on analysing multiple-injection strategies with B20 operation in a CRDI engine. They briefly described their work and their recording on how to decrease the NOx emissions and soot emissions by multiple injection strategies most importantly single and triple injection cases. According to their parametric investigations they recommend a suitable pilot fuel quantity and longer dwell between the pilot and main injection for NO reduction, and a small quantity of post fuel and medium dwell between main and post-injection for achieving soot reduction without penalty on mean effective pressure. This report summarized that an increase in injection pressure is beneficial in terms of soot emission reduction and improvement in gross indicated fuel conversion efficiency while it increases NO emission. The NO emissions increased by 36.7% and 45% respectively in single and multiple injections. Whereas the soot emissions saw reduction up to 70.3% and 68.2% for diesel and B20 respectively.

S. d'Ambrosio et al. [7] Worked on the potential of double pilot injection strategies optimized with the design of experiments procedure to improve diesel engine emissions and performance. The parameters have been assessed experimentally on a Euro 5 diesel engine with a reduced compression ratio (16.3:1). The engine has been fuelled with conventional diesel fuel. The experimental tests on the engine have been carried out in a dynamometer cell under different steady state working conditions that are representative of passenger car engine applications over the European homologation cycle.

Furthermore, in-cylinder analyses of the pressure, heat-release rate, temperature and emissions have been performed in order to obtain more detailed knowledge on the cause-and-effect-relationships between the implemented injection strategies and the results of the experimental tests.The implemented double-pilot injection engine calibrations have been optimized by means of the design of experiments procedure. The plotted data of the engine performance and emissions have been

Compared with data from the original double-injection schedule, characterized by a retarded main injection timing, in order to intensify the premixed combustion phase. The benefits and the disadvantages of the PCCI concept are preliminarily discussed, on the basis of the experimental pilot–main injection strategy results. The substitution of the pilot–main injection schedule with the triple injection, for light engine loads and low engine speeds, has led to higher mean combustion pressures, lower heat release rates, shorter ignition delays and lower brake specific fuel consumption. Above all, a significant improvement in engine noise and in both CO and HC engine-out emissions has been achieved and the NOx emission have been limited by the application of high EGR rates. When medium engine loads and speeds are analyzed, the considered double-pilot injection strategy allows the NOx emissions to be reduced, compared to the base line pilot–main injection schedule. However, the combustion noise does not improve, and the soot deteriorates, even though the soot penalties are not relevant.

Selvakumar Ramalingam et al. [8] Had studied the effect of advanced injection strategy on diesel engine characteristics fueled with moringa oleIifera biodiesel and its blends. Experiment was carried out on a Common Rail Direct Injection assisted diesel engine under different fuel injection pressure from 300 bar to 600 bar and fuel injection timing was varied from 15°CA to 25°CA bTDC. The results showed that the maximum brake thermal efficiency of 33.49% was obtained for B20 at higher injection pressure and advanced injection timing. The B20 achieved maximum heat release rate of 41.7 kJ/m3°CA which is nearly equivalent to conventional diesel at advanced injection timing and higher injection pressure. But in view of nitric oxide emission, pure biodiesel showed an increasing trend at around 1389 ppm even after advancement of injection timing and injection pressure, this study revealed that advanced injection strategy along with Moringa Oleifera biodiesel at various percentages of blends reduces the exhaust emissions except oxides of nitrogen which thereby improves the efficiency of the engine.

2.2 STUDIES ON INJECTION TIMING:

Ravikumar Jayabala et al. [9] Investigated the influence of oxygenated additives (dimethyl carbonate, n-butanol blend) injection timing (IT) and Exhaust Gas Recirculation (EGR) on different engine characteristics with diesel/biodiesel blends. Results reveal that the peak in-cylinder pressures and heat release rate (HRR) falls slowly as the injection timing was late from 25°CA bTDC to 21°CA bTDC at all the EGR rates. During retarded injection timing 21°CA bTDC and 15% EGR rate there was a marginal 7% reduction in NOx emission for n-butanol blend operation compared to the dimethyl carbonate blend, but nearly 60% of NOx reduction was achieved against diesel operation. The experimentation was done at 21° CA, 23° CA, 25°CA bTDC and 5% to 15% EGR rate. It is observed that NOx emissions were less at higher EGR rates. Retardation of the injection timing from 25°CA to 21°CA favored the reduction of NOx emissions. The smoke opacity is less at 23°CA bTDC and 5% EGR rate. Injection timing advancement abates smoke formation. At 23° CA it is recorded lesser hydro carbon (HC) emission. Retardation of timing from 25°CA to 21°CA was unfavorable in HC oxidation which results in increasing of UHC. Advancing injection timing is beneficial in CO to CO2 Oxidation. The 23°CA bTDC is the preferred case for better emission characteristics.

D. Babu, R. Annand [10] have used biodiesel-diesel-hexanol and biodiesel-diesel-hexanol blends along with diesel to experimentally investigate the effect on the combustion, emission, and performance characteristics of a CRDI assisted diesel engine operating under various NOP and FIT at 100% load conditions as a conventional diesel engine produced more exhaust emissions, which did not meet the emissions norms and caused global warming and environmental pollution. Improvement in engine performance and lower emissions could be attained by a

CRDI assisted diesel engine through fuel modification and engine modification. Engine modification was done by varying nozzle opening pressure and fuel injection timing from 200 to 600 bar and 19 to 27 °CA bTDC respectively. The minimum CO emission of 0.07% vol. was obtained in diesel at NOP of 500 bar and FIT of 27°CA bTDC. The maximum CO emission of 0.17% vol. was obtained in diesel at NOP of 200 bar and 19°CA bTDC. The CO emission is increased at retarded injection timing rather than advanced fuel injection timing irrespective of NOP due to less time available for the fuel-air mixture, which led to incomplete combustion and caused more CO emission. It is observed that an increase in NOP and advanced FIT reduced the Unburnt Hydrocarbons (UBHC) emission for tested fuels. This was because higher NOP improved the fuel spray characteristics, which led to complete combustion and lower UBHC emission. The UBHC emission is increased at retarded FIT. This was due to a lesser ignition delay period, lower wall temperature, and poor air-fuel mixture. The lowest UBHC emissions of 19ppm were observed at a NOP of 500 bar and FIT of 27°CA bTDC. The highest UBHC emissions of 39ppm were observed at a NOP of 200 bar and FIT of 19°CA bTDC. The NO emission is mainly because of oxygen availability, residence time, and cylinder temperature. The NO emission was extremely high at higher NOP and advanced FIT for all tested fuels; this was because of better atomization, evaporation, homogeneous mixing, and more ID period. The minimum NO emission was produced at retarded FIT rather than advanced FIT. This was because of poor air-fuel mixtures due to a lower injection delay period resulting in incomplete combustion. Thus, it reduced the cylinder peak pressure (CGPP) and temperature caused minimum NO emission. The maximum NO emission of 2106ppm was obtained in diesel at NOP of 600 bar and FIT of 27°CA bTDC. The lowest NO emission of around 715ppm was obtained in a diesel engine at NOP of 200 bar and FIT of 19°CA bTDC. The smoke is an indication of incomplete combustion. The smoke emission generally depends on the volatility of the fuel, air-fuel ratio, fuel composition, latent heat of evaporation, mixing distribution, ignition delay period, and fuel-burning velocity, it can be seen that increase in NOP and FIT reduced the smoke emission. The diesel engine has the maximum smoke emission of 4.46 FSN at NOP of 200 bar and FIT of 19°CA bTDC. The lowest smoke emission of around 2.55 FSN at NOP of 600 bar and FIT of 27°CA bTDC was observed.

Limin Geng a et al. [11] Investigated that the effects of injection timing, rail pressure on combustion characteristics and cyclic variations of a common rail DI engine fueled with Fischer-Tropsch (F-T) diesel synthesized from coal. It shows the advanced (ignition timing) IT results in an initial BTE increase and a subsequent BTE decrease. When the IT was advanced from 2°CA to 18°CA bTDC, the ignition delay periods (IDP) first decreased and then increased, whereas the combustion durations (CD) first lengthened and then shortened; peak cylinder pressure (PCP), peak pressure rise rate (PPRR), and peak combustion temperature (PCT) gradually increased; peak heat release rate (PHRR) first decreased and then increased at the low loads, whereas it always increased at medium and high loads. Advancing the injection timing or increasing the engine loads can decrease the cyclic variations.

Pandianet al [12] Examined and achieved lower BSEC, CO, HC, and Smoke with higher BTE, NOx at 225 bar IP, 2.5mm nozzle tip protrusion (NTP), and 30˚BTDC IT. The lower NOx and higher performance achieved at 225 bar IP, 21˚ bTDC IT and 2.5 mm nozzle tip penetration with neat Pongamia biodiesel in RSM optimization approach.

Ganapathy et al [13] investigated that the air cooled, single cylinder 5.59 kW power naturally aspirated DI CI engine using Jatropha biodiesel by varying IT 340, 345 (base) and 350 Crank angle degree (CAD). The lower BSFC, CO, HC and Smoke and higher BTE, ICP, HRR peaks but higher NOx at early IT from engine ratings. The 340 CAD (retarded) IT influenced significantly @ torque 15 N-m, 1800 rpm to reduce 5.1% BSFC, 2.5% CO, 1.2% HC and 1.5% Smoke and to increase 5.3% BTE, 1.8% Pmax, 26% HRR max and 20% NOx emissions. The 340 CAD IT was optimal for better engine characteristics.

2.3 STUDIES ON INJECTION PRESSURE:

C. Syed Aalam et al. [14] Hypothesized that the atomization of fuel during the injection is related to the higher viscosity of methyl ester blend. In order for better utilization of mahua methyl ester blend, fuel injection pressure increased from 22 MPa to 88 MPa. High fuel injection pressure (88 MPa) exhibits higher BTE and better combustion characteristics when compared to that of other injection pressures. HC, CO and smoke level gradually falls with increase in the fuel injection pressure due to better mixture formation because of the well-atomized spray. In cylinder pressure was increased with the increase in fuel injection pressure. HRR was also increased with the increase in fuel injection pressure. When the fuel injection pressure was increased from 22 MPa to 88 MPa, the BSFC was drastically reduced to 7.5%. BTE was increased with the increase in fuel injection pressure. The CO, HC and smoke emissions were further reduced, when the fuel injection pressure was increased from 22 MPa to 88 MPa. Increase in fuel injection pressure further increases the NOx emission. Results show that utilization of MME20 in CRDI diesel engines with 88 MPa fuel injection pressure can improve the efficiency while reducing the exhaust emissions.

A.J. Deokar et al. [15] Studied the effect of injection pressure, injection timing and nozzle geometry on performance and emission characteristics of diesel engine operated with Thevetia peruviana biodiesel. The results showed that using 230 bar IP, 26° bTDC IT, and 5-hole nozzle raises BTE and nitric oxide (NOx) emissions while lowering HC, CO, and smoke emissions. Among the other readings, IP of 230 bar gives higher BTE, NOx emissions and lower HC, CO and smoke emissions. The 26° bTDC IT gives higher BTE, NOx emissions and lower HC, CO and smoke emissions when compared to other IT.

Joonsik et al. [16] Researched on the effect of injection pressure, injection timing on combustion and emission characteristics of a single-cylinder common-rail direct injection (CRDI) diesel engine with waste cooking oil (WCO) biodiesel and commercial diesel fuel. The engine tests were conducted at two injection pressures (80 and 160 MPa) and different injection timings from -25 to 0 crank angle degree (CAD) after top dead center (aTDC) under two different engine loads. The results showed that the indicated specific fuel consumption (ISFC) with respect to the injection timings of the biodiesel was higher than that of the diesel fuel under all experimental conditions. The peak cylinder pressure and the peak heat release rate of the biodiesel were slightly lower, while the ignition delay was little longer under all operating conditions. It is found that at high fuel injection pressure there is reduction of the emissions such as smoke, carbon monoxide (CO), hydrocarbon (HC).

Channapattana et al. [17] Investigated that the Single cylinder water cooled DI CI engine at rated operating parameters by varying IP of 30 bar above and 30 bar below of rated (210 bar) Injection pressure. The BSFC of neat Hone oil biodiesel at 240 bar IP, 18CR, and 23˚ bTDC IT attributed to 0.042 Kg/kW-hr higher with reasonable lower emissions than Diesel fuel. The increase in NOx emissions was observed with increase in IP and blends.

CHAPTER 3

EXPERIMENTAL SET-UP

3.1 METHODLOGY:

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 and open ECU

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 of 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 schematic diagram 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 1.

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 signaling 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.

Table 3.1 Specifications of test engine

Item	Particulars
Make/Model	Kirloskar/TV1
Engine	1 cylinder, 4-S, water cooled, Diesel engine
Bore/Stroke	87.5 mm/110 mm
Cubic capacity	661 cc
Rated power	3.5 kW @ 1500 rpm
Fuel injection	Common rail direct injection with pressure sensor and pressure regulating valve
Injector	Solenoid driven, six hole
Nozzle hole diameter	0.127 mm
Injection angle	152^{o}
Dynamometer Type	eddy current, water cooled
ECU	Nira i7r (with solenoid injector driver) with programmable ECU software and Calibration cable
Loading/Make	Eddy current dynamometer / AG10 of Saj Test Plant Pvt. Ltd., Pune
Fuel tank	Capacity 15 lit with glass fuel metering column
Piezo powering unit	Model AX-409.
Data acquisition device/Make	NI USB-6210, 16-bit, 250 kS/s / National Instruments,USA
Overall dimensions	W 2000 x D 2500 x H 1500 mm

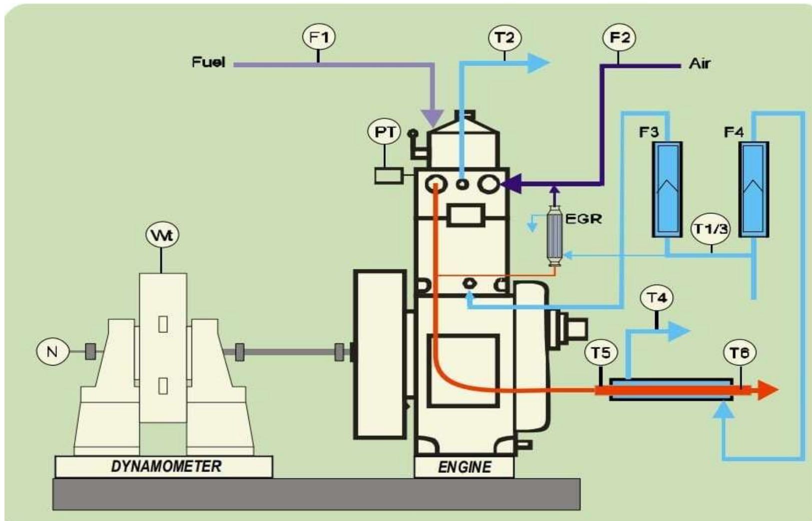


Fig 3.2 components of CRDI Diesel Engine

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 LabVIEW 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**			

The above table represents the shaft order code for Kubler make Incremental type 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):

Fig3.3. sensors Piezo

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 : 0.4 kgf of aluminium &potted waterproof enclosures Operational voltage: 6 – 22v DC supply
Operational temperature : -30°C to 75°C depends on loading Active voltages : 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 0 to 5.25. The DC input coupling with timing accuracy 50ppm of sample rate and resolution 50 ns.

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.

Fig 3.4 Specifications of load cell

3.2.6 Differential pressure Fuel transmitter:

Fig 3.5 Differential pressure fuel transmitter

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-stands for **Wetted parts material** of Cover flange, process connector: ASTM CF- 8M Capsule: Hastelloy DiaphragmC-276[4]; F316L SST, 316 L SST Capsule gaskets: Teflon-coated316L SSTVent/Drain plug 316 SST

5-stands for **Process connections** without process connector (1/4 NPT female on the cover flanges)

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

3.2.7 IC Engine Soft:

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.

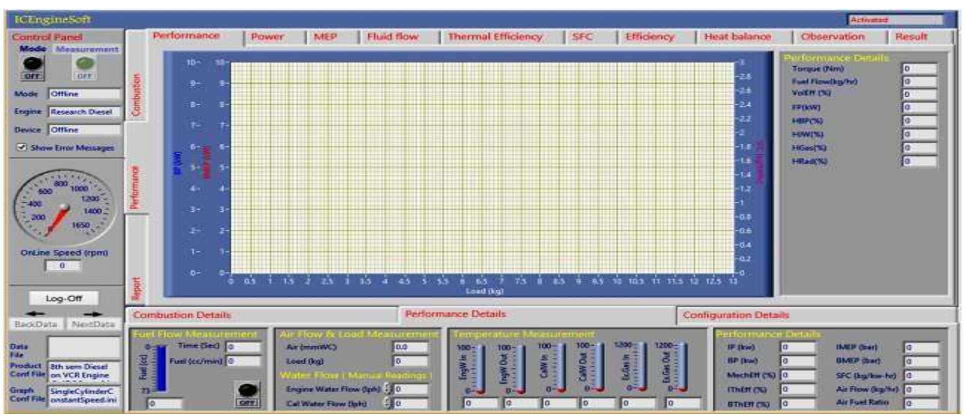


Fig 3.6 IC engine soft

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.

Fig 3.7 Airflow transmitter

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 5 gas analyzer. The Smoke emissions measured by using AVL 437C model Smoke Meter. The detailed specifications of above 5 gas analyzer and Smoke meter given in Table 4.4 and their photographic views are presented in Figure 4.3.

AVL 5 gas analyser, 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

Table 3.3 Specifications of emissions analysers

Fig 3.8 Photographic view of the emission analyse.

3.4 EXPERIMENTAL DETAILS

Table 3.4 EXPERIMENTAL DETAILS

Condition	Injection Pressure (bar)	Injection timing (BTDC)	Injection Quantity
Base Line	210	23°	100%
Retarded Injection	300	7° ,10 ° ,13° ,16° ,19°	100%
Split Injection	300	13°, 16°,19°	10% + 90%
Split Injection	300	16°	10% + 90%

3.5 TEST METHOD:

Studies on single cylinder CRDI diesel engine by using retarded and split injection strategy were evaluated in this study. The properties of fuel were found as per the ASTM standards at fuels and IC engines lab, NIT Warangal (TS), India. 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 0C, and the lubricating oil temperature was maintained in the range of 85 ± 2 0C throughout the testing.

Engine testing was carried out at constant speed at varying loads, using retarded ad split injection timings and fuel injection pressure is maintained constant at 300 bar pressure for analyzing its performance, combustion, and emission characteristics. By considering all factors compression ration is fixed as 18 throughout the test. Eddy current dynamometer was connected to the engine to apply load on the engine. Table 3.4 presents the details of parameters used during experimentation.

Test was carried out in three stages for analyzing its performance, combustion, and emission characteristics. At stage 1 the experiment was carried out by retarding the injection timing from 19^0 bTDC to 7^0 bTDC in the intervals of 3^0. The best performance, combustion and emissions characteristics are obtained at 13 and 16 bTDC. To obtain a broader view on reducing the emission and increasing the performance spilt injection is introduced in our study. In stage 2, the experiment was conducted on advanced pilot injection timings at intervals of 3 degrees from 30 bTDC to 36 bTDC and kept main injection timing constant as 16^0 bTDC, to analyze the combustion and emission characteristics in a single-cylinder diesel engine. The best result was obtained at 30 BTDC pilot injection timing. The next set of testing conditions were carried out at constant pilot injection timing as 33^0 bTDC and main injection was retarded at intervals of 3 degrees from 13 BTDC to 19 BTDC. The averaged combustion data was then used to compute variation of CP, HRR, PRR with load, occurrence of HRR and PRR, combustion duration, etc. Emissions of NOx, CO, UHC, and smoke opacity were measured.

CHAPTER 4

RESULTS AND DISCUSSIONS

4.1 PERFORMANCE CHARACTERISTICS

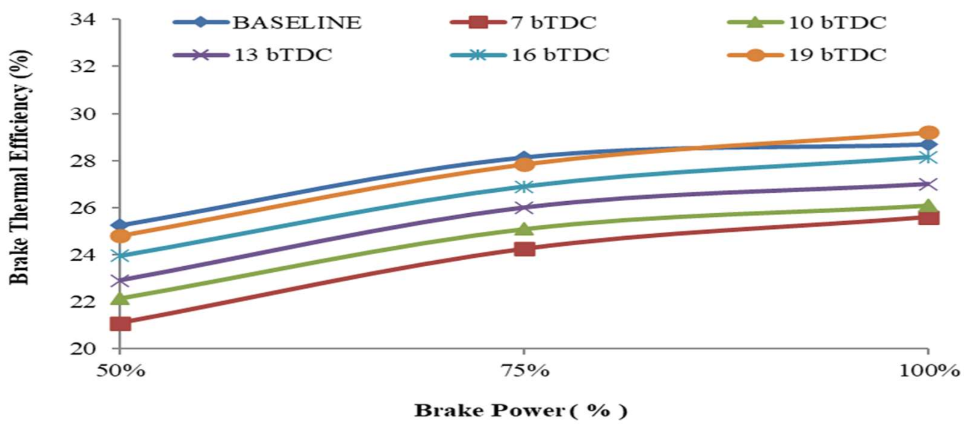
4.1.1 BREAK THERMAL EFFICIENCY:

The fig 4.1(a) shows the effect of load on the brake thermal efficiency at different fuel injection timings. At a given brake power, it is observed that as the injection timing retards, the brake thermal efficiency reduces. When compared to the baseline condition (23bTDC at 210 bar FIP), as the fuel injection timing advances too much (before 19bTDC), the BTE decreases as the combustion initiates before TDC and there is a high chance that maximum pressure occurs before the piston reaches TDC. In such a case, the high pressure of gases restricts the upward movement of the piston and hence energy loss takes place. Therefore, very less indicated power is generated which in turn is converted into brake power after frictional losses take place during the expansion stroke. For a given injection timing, as the load increases, the brake thermal efficiency also increases as BTE is a direct function of brake power.Hence, retarding injection timings too much might also lead to a loss in BTE because in such a situation, the maximum pressure is attained during the expansion stroke and the high pressure is not completely received by the downward moving piston. So, the injection timing must be adjusted in such a way that the maximum pressure inside the combustion chamber is attained after TDC (about 10 to 15aTDC) i.e., during the expansion stroke. This is achieved by injection of fuel just before TDC.

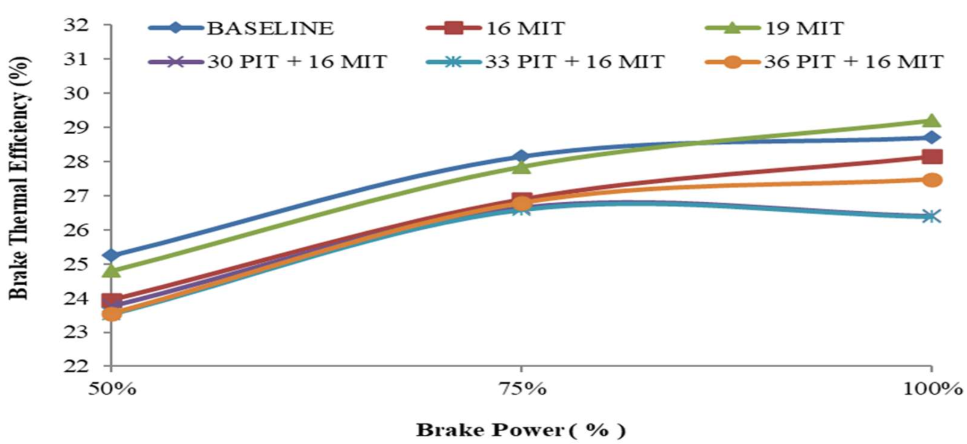
Fig 4.1 (b) shows the effect of load on brake thermal efficiency at different injection timings using pilot injection. As the PIT advances by keeping the MIT same, it is observed that the BTE almost remains the same until 75% load but slightly increases after that as at high loads the there would be a uniform increase in the pressure rate in the combustion chamber as the dwell time between PIT and MIT increases.

As the MIT advances, an increase in the BTE is observed as there would be enough time for complete combustion of the air present in the chamber and the maximum pressure is attained just after the compression stroke (at about 10 to 15aTDC).

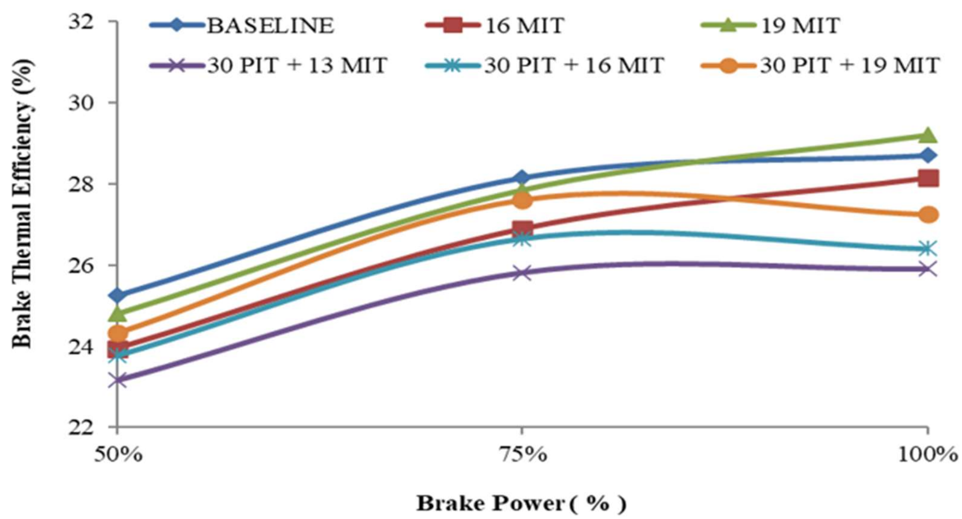
Fig 4.1(c) shows the effect of load on the brake thermal efficiency at different injection timings with pilot injection. At a given load, as the dwell between the pilot injection and the main injection increases the BTE reduces as the pilot injection reduces the maximum cylinder pressure gradient (rate of pressure rise). Though the dwell period doesn't affect the maximum cylinder pressure, it does affect the IMEP. If we compare these conditions to that without pilot injection, BTE is comparatively high because there would be high-pressure gradients. As the load increases, the BTE increases and then decreases in the case of pilot injection.



(a)



(b)



(c)

Fig 4.1 variations of BTE at different operating conditions

4.1.2 BREAK SPECIFIC FUEL CONSUMPTION

The fig 4.2(a) shows the effect of load on the brake specific fuel consumption at different fuel injection timings. At a given load, as the injection timing advances, the BSFC decreases. When compared to the baseline condition (23bTDC at 210 bar FIP), as the fuel injection timing advances, nearly homogenous mixtures are prepared inside the combustion chamber due to longer ignition delay and hence better combustion leads to lower BSFC.

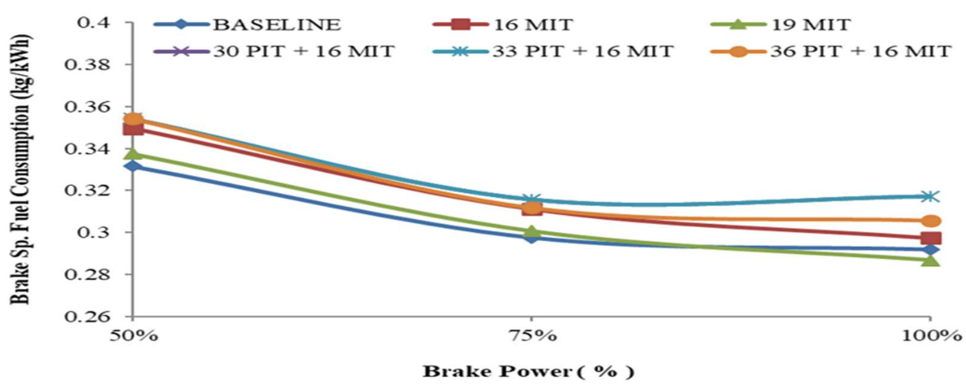
The BSFC says how effectively the amount of fuel gets converted into brake power.

At a given ignition timing, as the load increases the BSFC reduces as it is inversely proportional to the brake power. The low values of the BSFC indicate that the engine is more efficient and if we increase the value of BSFC consequently the efficiency of the engine decreases.

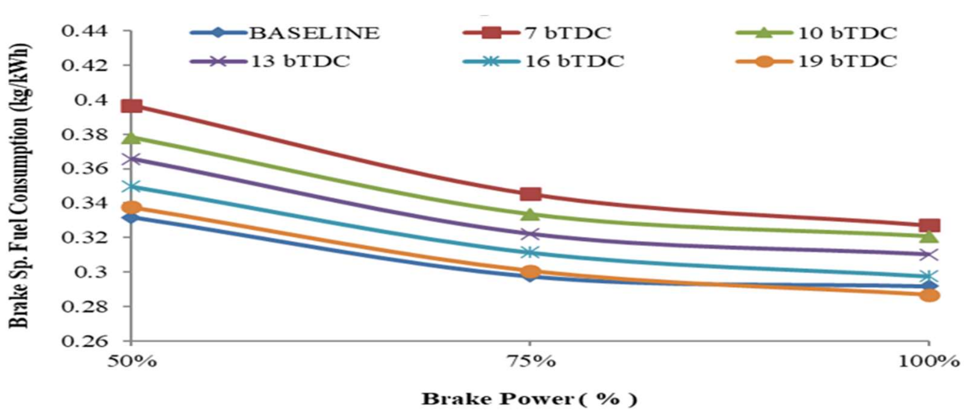
So, if we keep the fuel economy in mind, it is suggested that to retard the injection timing to such a value that both the engine efficiency and the BSFC are maintained within certain limits.

The fig 4.2(b) shows the effect of load on the brake specific fuel consumption at different fuel injection timings with pilot injection. At a given load, it is observed that as the pilot-main injection time gap decreases the BSFC reduces because of the advanced combustion phasing, which is caused by the increased heat release from pilot injection and advanced ignition timing of the main injection. On the other hand, for the main injection, the curves are lower when compared to the ones with pilot injection as there is a sudden increase in the pressure rate and hence more air is converted into power and hence more fuel consumption.

The fig 4.2 (c) shows the effect of load on brake thermal efficiency at different injection timings using pilot injection. As the PIT advances by keeping the MIT same, it is observed that the BTE almost remains the same until 75% load but slightly increases after that as at high loads the there would be a uniform increase in the pressure rate in the combustion chamber as the dwell time between PIT and MIT increases. As the MIT advances, an increase in the BTE is observed as there would be enough time for complete combustion of the air present in the chamber and the maximum pressure is attained just after the compression stroke (at about 10 to 15aTDC).



(a)



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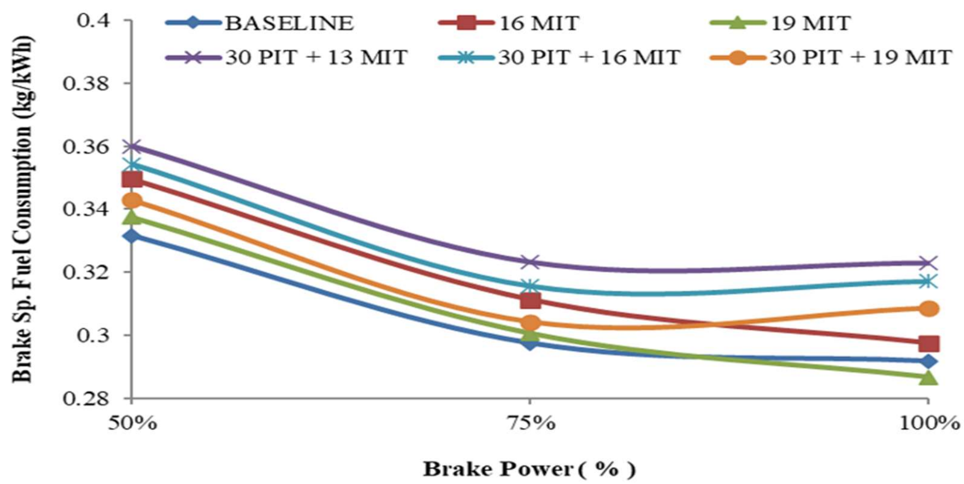


(c)

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

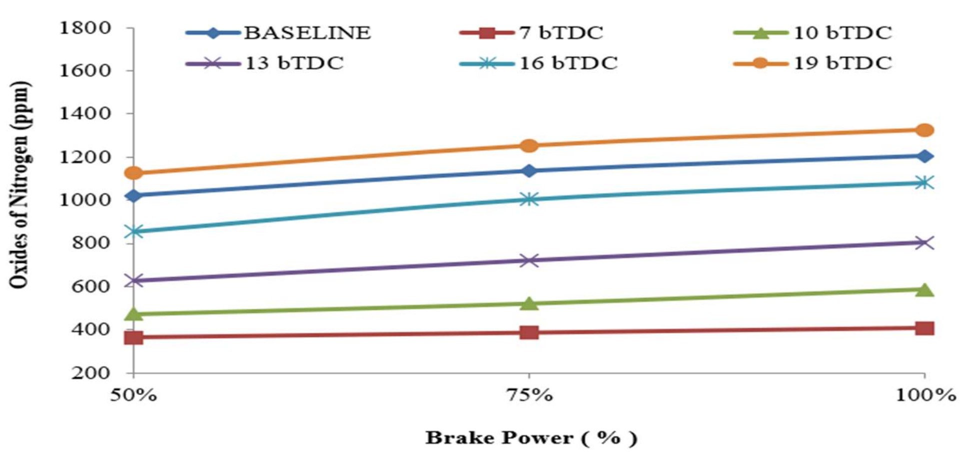
4.2 EMISSIONS CHARACTERISTICS

4.2.1 NOx emissions

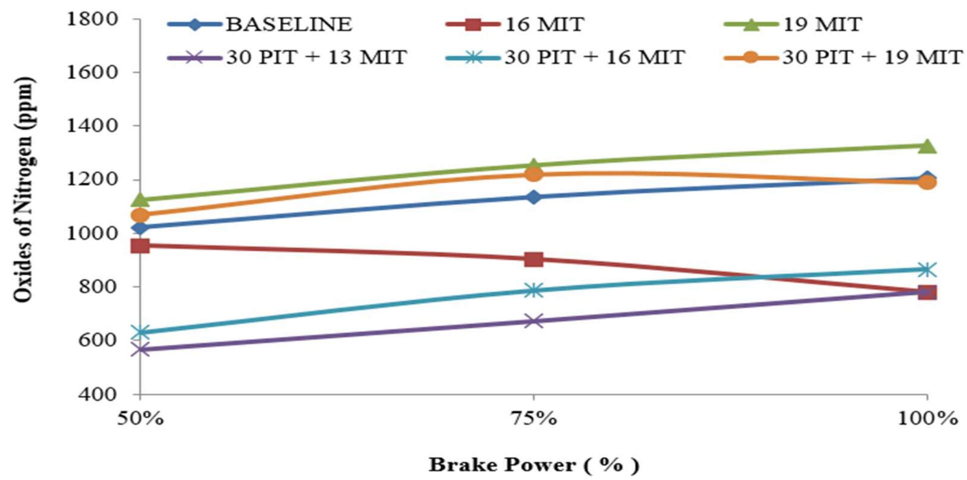
The formation of the NOx emission mainly depends on the in-cylinder temperature, oxygen concentration, and the time available for the reactions to take place. Dissociation of the diatomic nitrogen (N2) and oxygen (O2) molecules into their atomic states at very high temperatures undergo a series of reactions in the combustion chamber to produce thermal NOx. The fig 4.3 (a) displays the variation of NOx emissions at various injection timings and load conditions. It contains baseline data that is at 23 bTDC and 210 bar the remaining readings were taken at different injection timings and constant pressure of 300 bar. The results showed that there is overall increase in the NOx emissions with increase in load. As the brake power increases the temperature becomes very high due to the presence of higher oxygen content, which enhances combustion and results in higher NOx emissions.

Fig 4.3 (b) the oxides of nitrogen are compared with the brake power for different injection timings using split injection strategy. Here the injection is done twice first the pilot injection injection is done at 30bTDC and latter the main injection is done at varrying retarded injection timings. The NOx emissions decrease with the retardation of main injection timing this is because as the premixed combustion is the main reason for increase in NOx emissions formation and as here it is relatively low. The more quantity of the second injection pulse into the lean and hot combustion zones, causing the newly injected fuel to burn rapidly and efficiently at high temperatures resulting in less NOx emissions.

The fig 4.3 (c)is between the NOx emissions and brake power at different injection timings. Here split injection strategy is utilized with the pilot injection advancing but the main injection being constant at 16bTDC. Here we can observe that the NOx emissions by split injection strategy are increasing but not at a significant rate as the injection is advancing the time period between main injection and pilot injection increases thus as the combustion temperatures generated due to pre ignition decreases with the advanced injection timings therefore the in cylinder temperature near the main injection are not optimum to the rapid evaporation of the fuel which leads to more oxygen availability at more NOx emissions as seen below.



(a)



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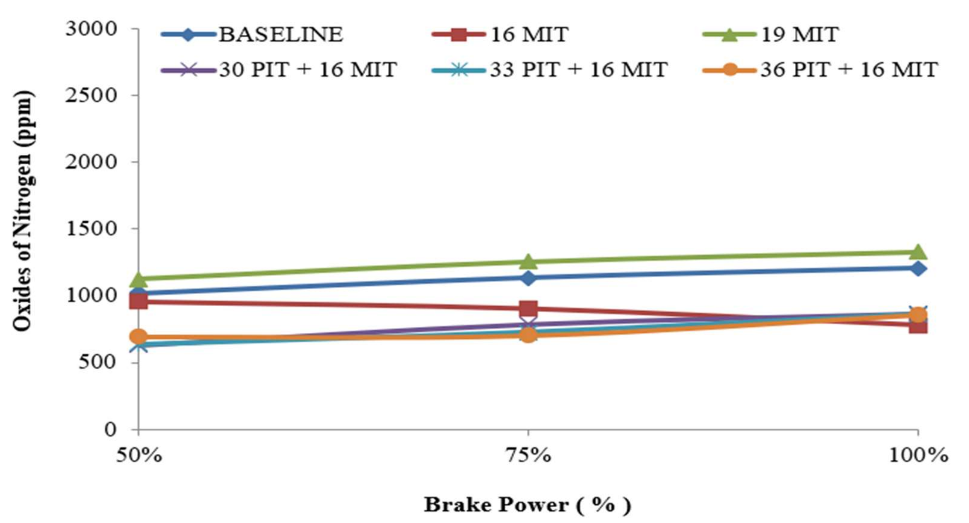


(c)

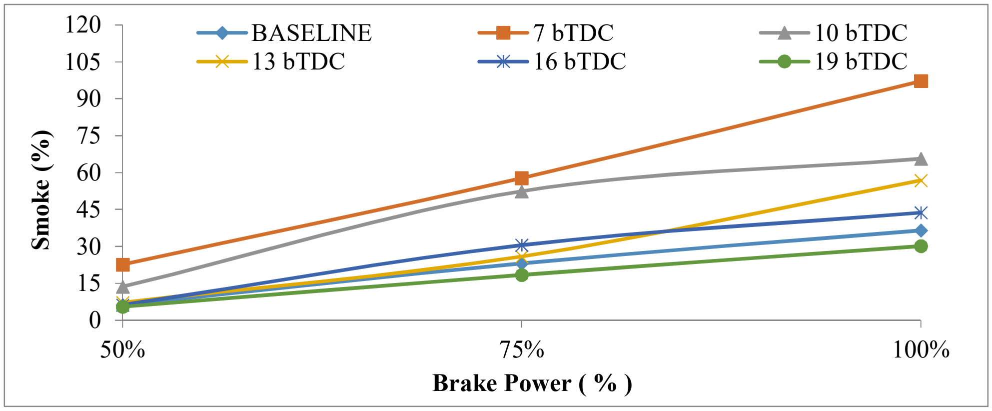
Fig 4.3 Variation of NOx emissions with Brake power

4.2.2 SMOKE EMISSIONS:

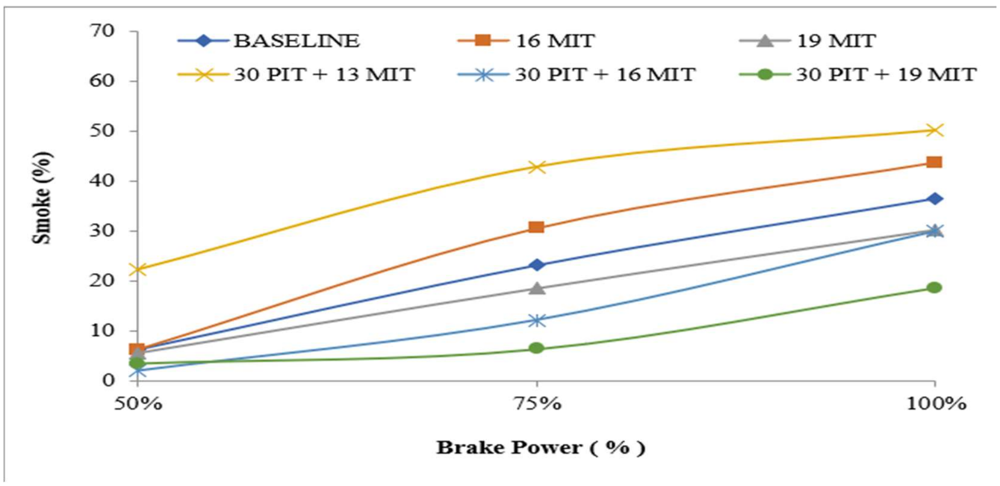
The formation of smoke commonly effects from the unfinished burning of the hydrocarbon fuel and the partly reacted carbon content material in the liquid fuel. Here we can observe that there are some abnormal changes for different injection timings The effect of the fuel injection timing on the smoke level for different engine loads and speeds is shown below. The fig 4.4 (a) shows the variation of smoke emission with respect to brake power at different injection timings. It contains baseline data that is at 23 bTDC and 210 bar the remaining readings were taken at different injection timings and constant pressure of 300 bar. Smoke formation can occur for retarded injection timing, which in turn reduces the ignition delay. The ignition delay increases when advancing the start of fuel injection as more fuel is injected before ignition, which leads to higher temperatures in the combustion cycle (faster combustion and better mixing). Accordingly, the smoke level reduced with retarded compared with advanced injection timing. This is due to the large amount of evaporated fuel that accumulates for advanced

In the fig 4.4 (b) the Smoke emission percentages are compared with the brake power for different injection timings using split injection strategy. Here the injection is done twice first the pilot injection injection is done at 30bTDC and latter the main injection is done at varrying retarded injection timings. The in-cylinder temperaure 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.

The fig 4.4 (c) is between the Smoke emissions percentage emissions and brake power at different injection timings. Here split injection strategy is utilized with the pilot injection advancing but the main injection being constant at 16bTDC. As the injection timings are advanced that results in increasing of in cylinder temperature which inturn 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.



(a)



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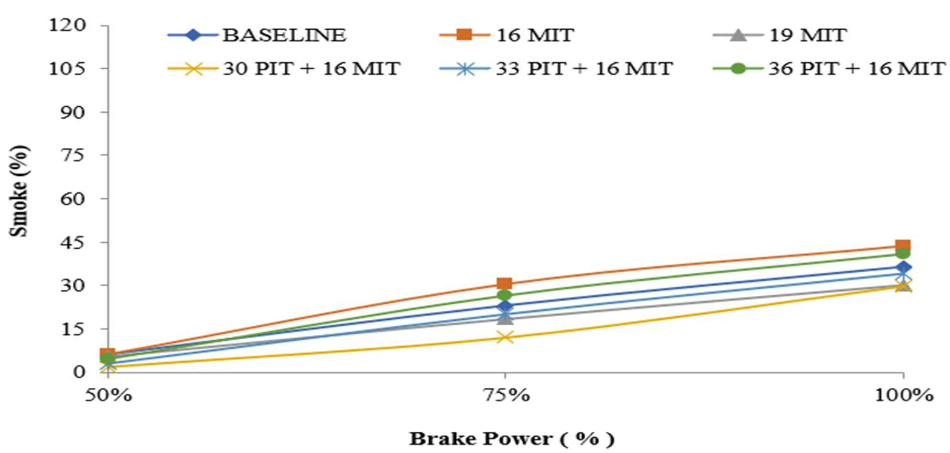


(c)

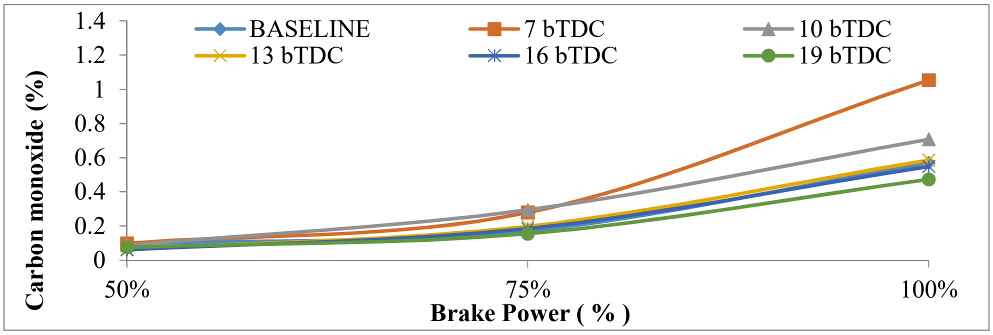
Fig 4.4 Variation of smoke emissions with Brake power at different operating conditions.

4.2.3 CO EMISSIONS:

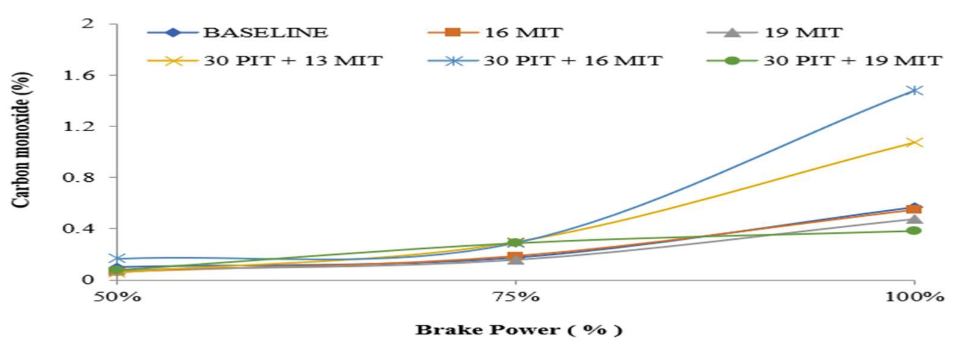
Carbon monoxide results from the incomplete combustion where the oxidation process does not occur completely. CO is produced if the droplets in a diesel engine are too large or if insufficient turbulence or swirl is created in the combustion chamber. The effect of the fuel injection timing on the smoke level for different engine loads and speeds is shown below. The fig 4.5(a)shows the variation of carbon monoxide emission with respect to brake power at different injection timings. It contains baseline data that is at 23 bTDC and 210 bar the remaining readings were taken at different injection timings and constant pressure of 300 bar. 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.

In the fig 4.5 (b) the Carbon monoxide emission percentages are compared with the brake power for different injection timings using split injection strategy. Here the injection is done twice first the pilot injection injection is done at 30bTDC and later the main injection is done at varrying retarded injection timings. As the injection timings are retarded and the in cylinder temperatures are decreased the oxygen concentrations are decreased. With the decreased oxygen concentration the free carbon atoms form a bond with the oxygen atoms and form carbon monoxides. Therefore due to this reason the carbon monoxide emission percentages are increased with retarding the injection timings.

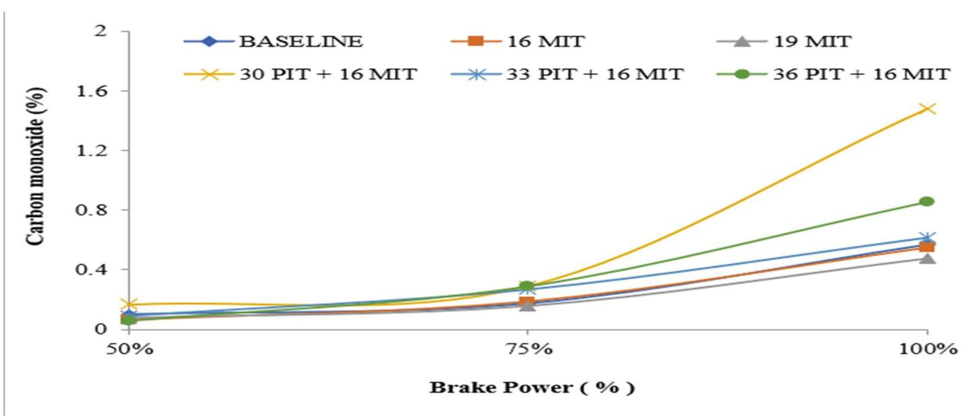
The fig 4.5 (c) is between the Carbon monoxide emissions percentage emissions and brake power at different injection timings. Here split injection strategy is utilized with the pilot injection advancing but the main injection being constant at 16bTDC. With the advanced injection timings the oxygen concentration in the cy;inder 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. Comparitively CO2 is less harmful that CO . Therefore this results in less carbon monoxide emissions in the cylinder.



(a)



(b)



(c)

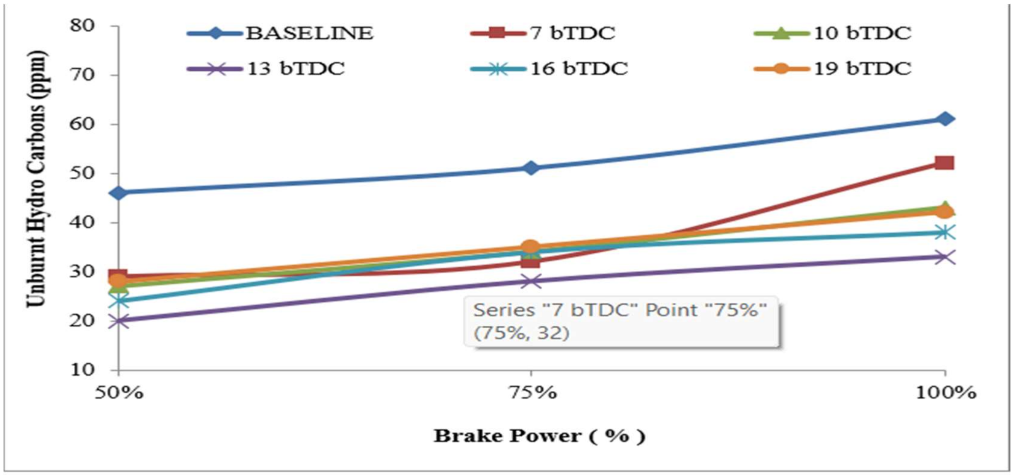
Fig4.5 Variation of CO emissions with Brake power at different operating conditions.

4.2.4 HC EMISSIONS:

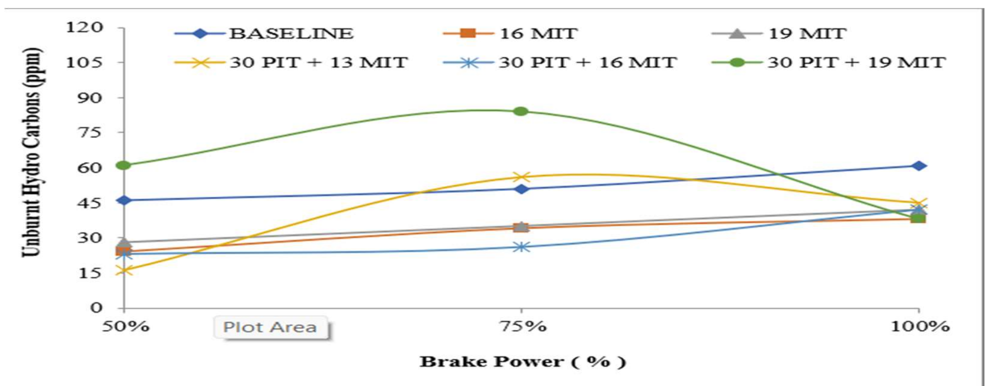
HC emissions mainly due to incomplete combustion of fuel in combustion chamber and partial combustion of lubricating oil. Richer as well as leaner fuel–air mixtures, both lead to hydrocarbon emissions, due to lesser availability of oxygen in the combustion zone. With increasing engine load, BSHC emissions either increases or remain constant. This is due to more fuel is injected into combustion chamber with constant mass of air intake. As a result, rich mixture is formed in combustion chamber with increasing engine load, leading to higher HC emissions. In the fig 4.6 (a) the graph is drawn between Hydrocarbon emissions and Brake power at retarding injection timings the emissions decrease with the retarding injection timings. For 19,16,13bTDC the emissions increase till half the brake power in the later part the graph is seen depleting whereas the graph for other timings is seen otherwise.

In the fig 4.6 (b) the unburnt hydro carbon emission percentages are compared with the brake power for different injection timings using split injection strategy. Here the injection is done twice first the pilot injection injection is done at 30bTDC and later the main injection is done at varrying retarded injection timings. As the timings are retarded the oxygen concentration decreases and due to the formation of CO the and with the availability of less oxygen atoms the formation of H_2O is not happening therefore the formation of HC is occurred. Thus the HC emissions increasse .

The fig 4.6 (c) is between the Unburnt Hydrocarbons emissions percentage emissions and brake power at different injection timings. Here split injection strategy is utilized with the pilot injection advancing but the main injection being constant at 16bTDC. As the pilot injections are advanced the oxygen concentration increased and causes to the formation of lean mixture. As the oxygen avalability is abundand the formation of HC decreases. Thus the HC emisions decrease with the advancement of pilot injections.



(a)



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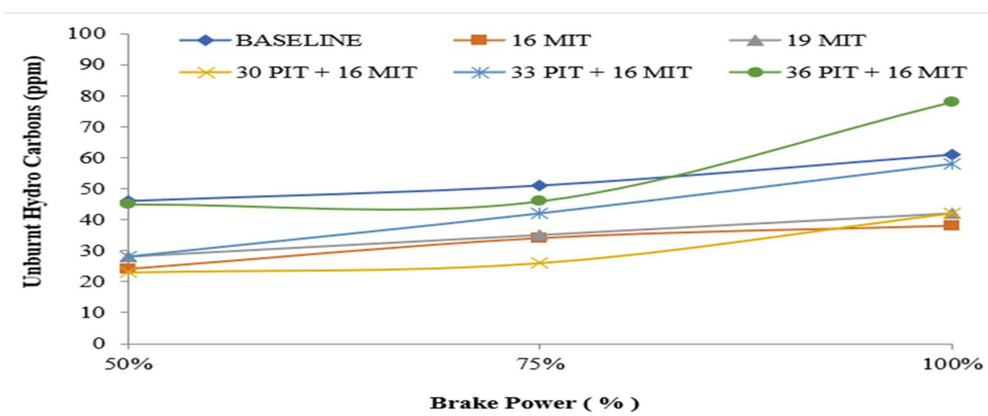


(c)

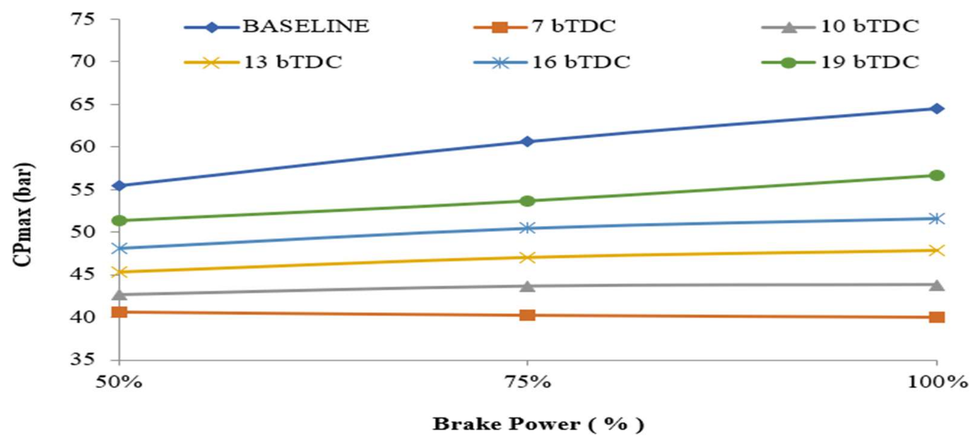
Fig4.6 Variation of UHC emissions with Brake power at different operating conditions

4.3 COMBUSTION CHARACTERISTICS.

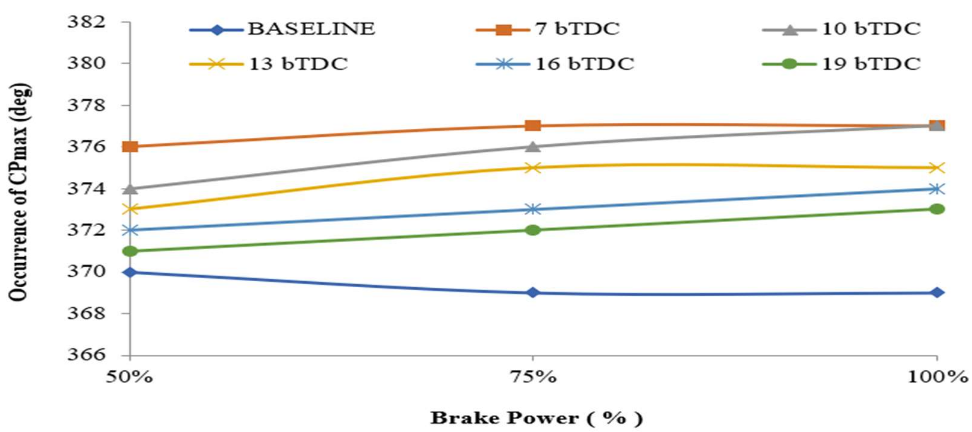
4.3.1 CYLINDER PRESSURE:

The measurement of in-cylinder pressure is an important parameter for understanding engine combustion. The analysis of in-cylinder pressure is used in finding various engine combustion parameters such as heat release rate; cumulative heat release and pressure rise rate. The in-cylinder pressure vs. break power for various SOI timings are shown in figure for different engine loads. From these graphs, it is generally observed that retarding the SOI leads to lower in-cylinder pressures at all engine loads. Retarded SOI results in less time available for formation of premixed charge. Therefore, relatively smaller fraction of fuel is burnt in the premixed phase. Retarding SOI leads to shorten the ignition delay which lowers the fuel quantity available at SOC. Once combustion starts smaller quantity of fuel is burnt in premixed phase results in lower in-cylinder temperatures which in turn reduces the peak pressures. Figure 4.7(a) shows variation in peak in-cylinder pressure at various retarded injection timings for different loads. It can be seen from figure that peak cylinder pressure is lower for retarded SOI timings compared to base line condition (23^0 bTDC). Fig 4.7(b) shows the occurrence of max cylinder pressure at different loads. The occurrence of peak pressure delayed with retarded injection timings at all loads. With increasing engine load, peak cylinder pressure increased, and its position shifted away from TDC due to higher fuel quantity being burnt, which results in longer combustion duration therefore the pressure peak appears relatively later in the expansion stroke.

Figure 4.7 (c), Fig 4.7(d), Fig 4.7 (e) and Fig 4.7 (f) shows the variation of peak cylinder pressure and their occurrence at different loads using various injection strategies. Compared to base line operation and single injection, the split injections show low peak cylinder pressures at all loads and their occurrence is also delayed. With increasing load occurrence is shifted away from TDC due to higher fuel quantity is being burnt which results in longer combustion duration therefore pressure peak appears relatively later in expansion stroke. The split injection timing (30 PIT + 13 MIT) shows low cylinder peak pressure and their occurrence more away from TDC. It is due to increase in injection interval results in advancing ignition timing and accelerates combustion end; combustion chamber pressure lower in expansion stroke (Suhan Park et al).



(a)



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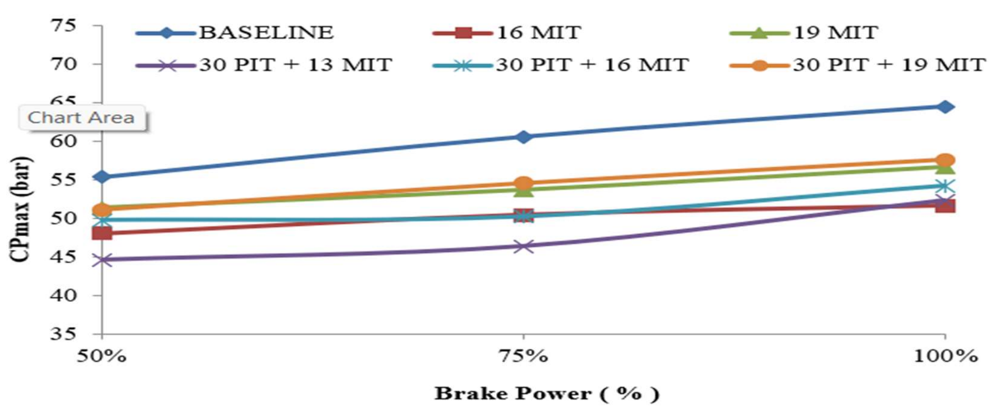
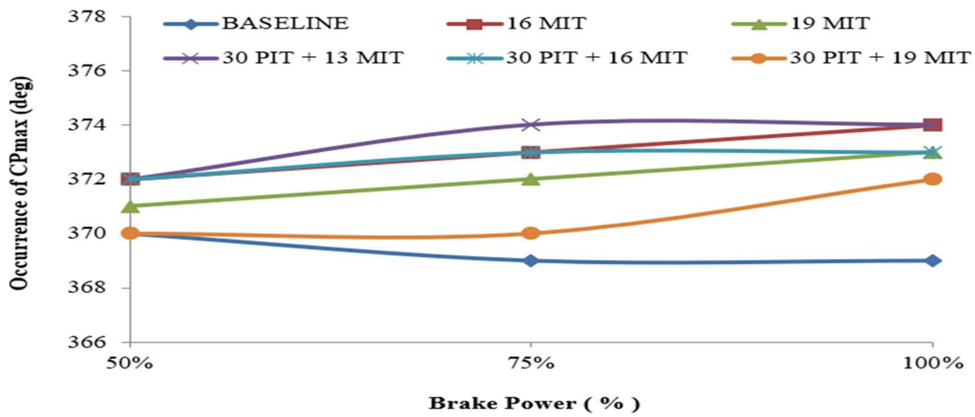
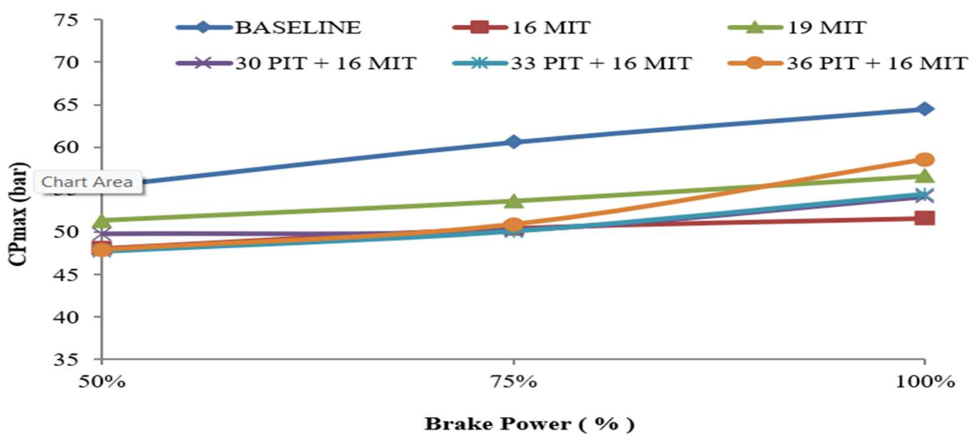


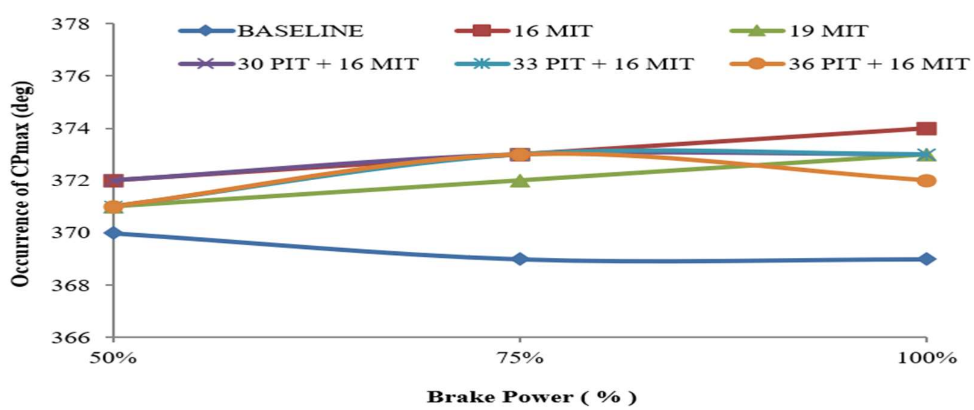
(c)



(d)



(e)



(f)

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

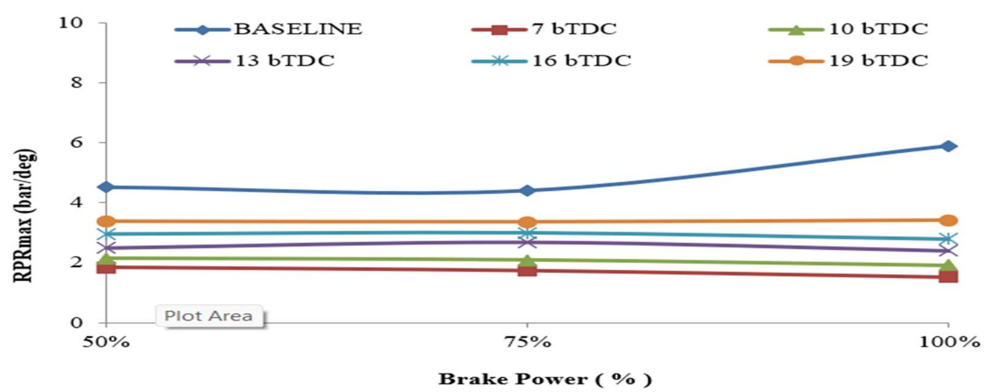
4.3.2 PRESSURE RISE RATE:

RPR provides information regarding the structural safety of the engine. Rate of force exerted on the piston and on other linkages will increase with the increase in RPR. It should not be too high from the viewpoint of engine safety. Therefore, it is essential to keep this parameter in view while changing the operating conditions of the engine. The variation of PRR at various SOI and their occurrence at diff loads is shown in Fig 4.8(a) and Fig4.8(b).

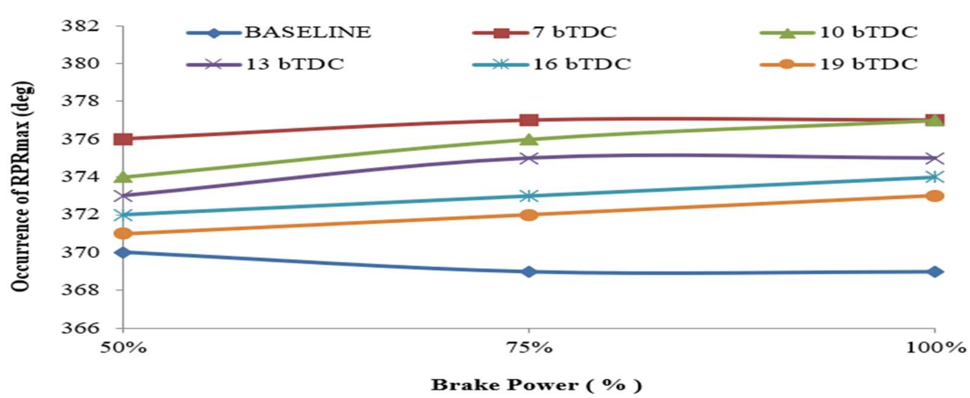
The rate of pressure rise reaches its maxima during premixed combustion phase because of rapid combustion and very fast premixed heat release. After reaching the maxima, it reduces in the expansion stroke because of mixing controlled combustion, in which the combustion is relatively slower in addition 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 RPR, 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. 4.8 (c), Fig4.8(d) shows the variation rate of pressure rises at various retarded injection timings with pilot injection as constant and load conditions. As we retarded the injection timing the gap (Dwell period) between pilot and main injection 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. Compared 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 4.8(e) and Fig 4.8(f) 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)



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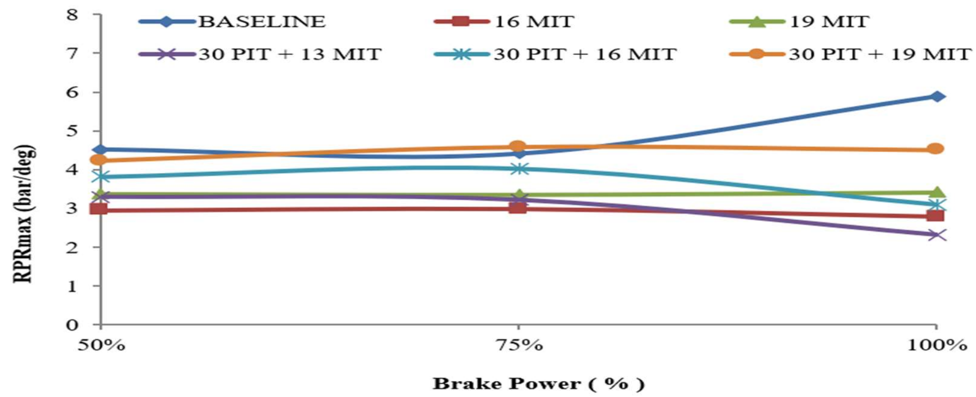
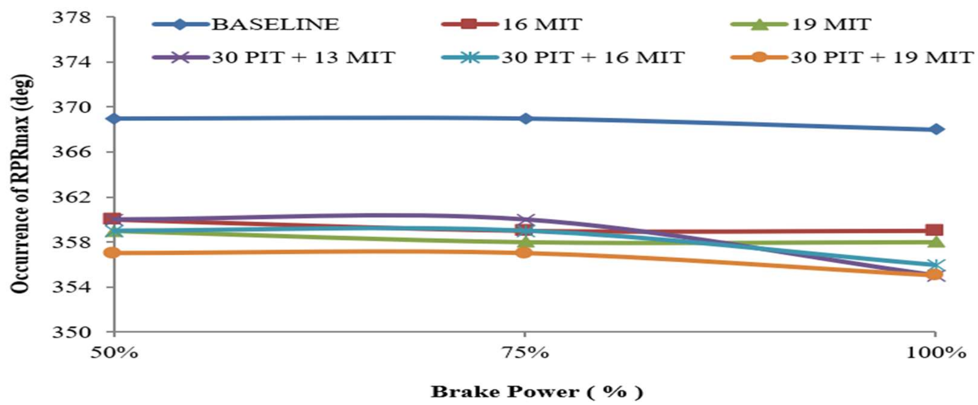
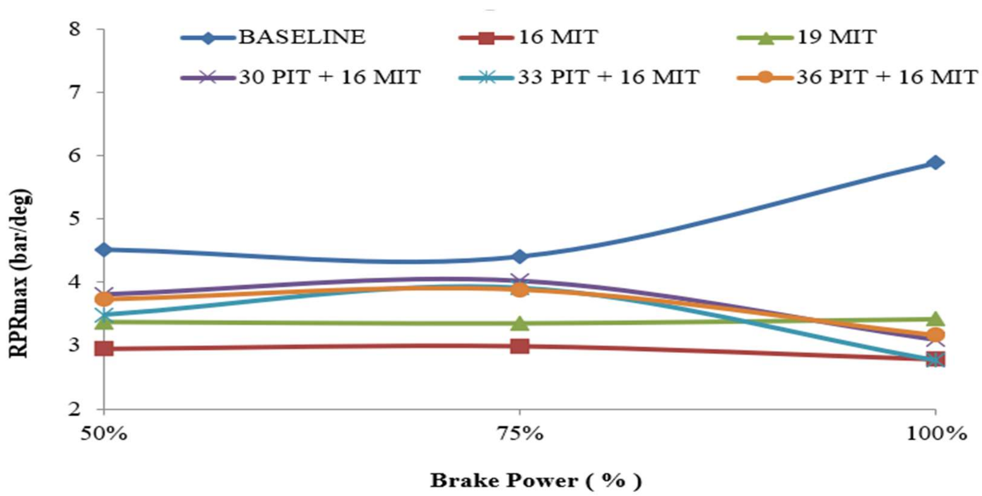

BASELINE
16 MIT
19 MIT
30 PIT + 13 MIT
30 PIT + 16 MIT
30 PIT + 19 MIT
RPRmax (bar/deg)
Brake Power (%)

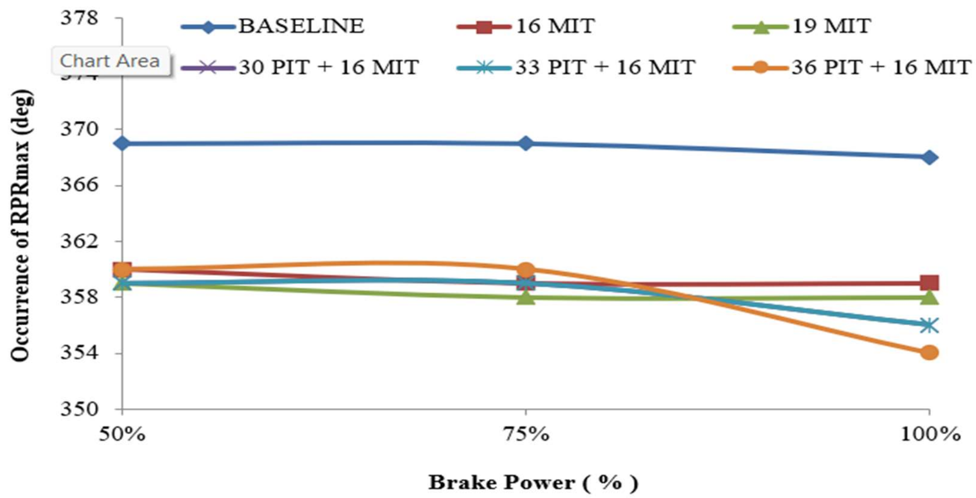
(c)


BASELINE
16 MIT
19 MIT
30 PIT + 13 MIT
30 PIT + 16 MIT
30 PIT + 19 MIT
Occurrence of RPRmax (deg)
Brake Power (%)

(d)


BASELINE
16 MIT
19 MIT
30 PIT + 16 MIT
33 PIT + 16 MIT
36 PIT + 16 MIT
RPRmax (bar/deg)
Brake Power (%)

(e)



(f)

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

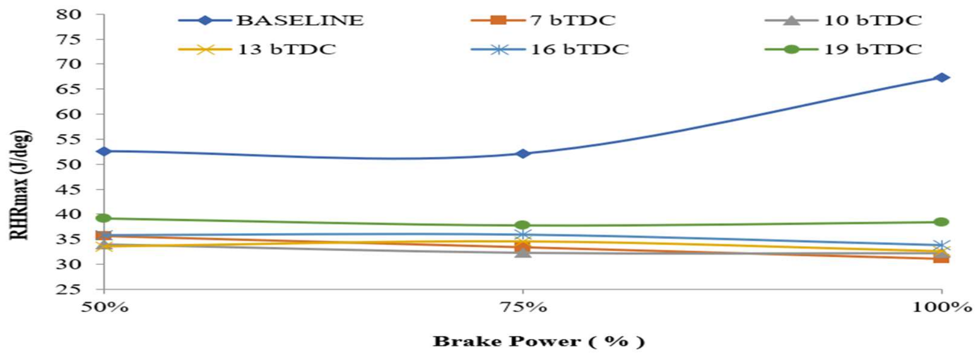
4.3.3 HEAT RELEASE RATE:

Fig 4.9 (a) and Fig 4.9(b) shows the heat release rate at various retarded SOI timings and their occurrence for different loads. Heat release takes place in two distinct stages. The first is immediately after the SOI to a point, where the heat release rate sharply drops. This is due to combustion primarily in the premixed combustion phase. The second phase starts from the end of first phase (Premixed combustion) to the end of combustion and this is called 'mixing-controlled combustion phase'. This is generally a slower heat release phase among the two, therefore, it spreads over a longer combustion duration and is essentially controlled by the rate, at which, the fuel and air can mix inside the combustion chamber.

Maximum rate of heat release rate is seen to be lower for retarded SOI timings compared to base line condition and advanced injection timing because relatively smaller fraction of the fuel quantity injected burns in the premixed combustion phase for retarded SOI timings. 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. The occurrence of heat release peaks is also delayed i.e., moved away from TDC with retarded injection timings. At higher engine loads, the fraction of heat release taking place in the mixing-controlled combustion phase is higher because the ignition delay is shorter for higher engine load. Therefore, smaller fuel quantity is available in combustion chamber at the time of premixed combustion, which lowers the peak and the crank angle position of this peak of heat release rate also shifts towards TDC.

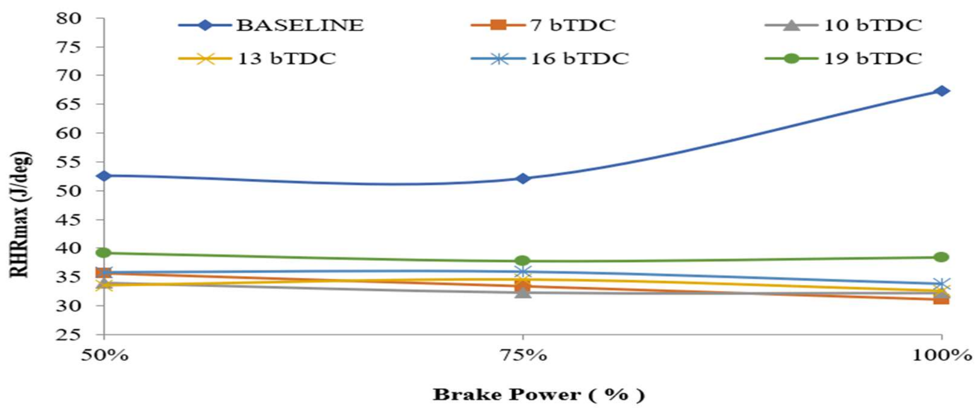
Fig.4.8 (c) and Fig 4.8(d) shows the variation of rate of heat release at various retarded injection timings with pilot injection as constant and load conditions. Retarded the main injection timing by fixing the pilot injection as constant RHR maximum was decreased because the ignition delay was decreased. 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 (e) and Fig 4.8 (f)show the variation of heat release rate with BP for different pilot injection timings. Higher heat release rate contributes 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).



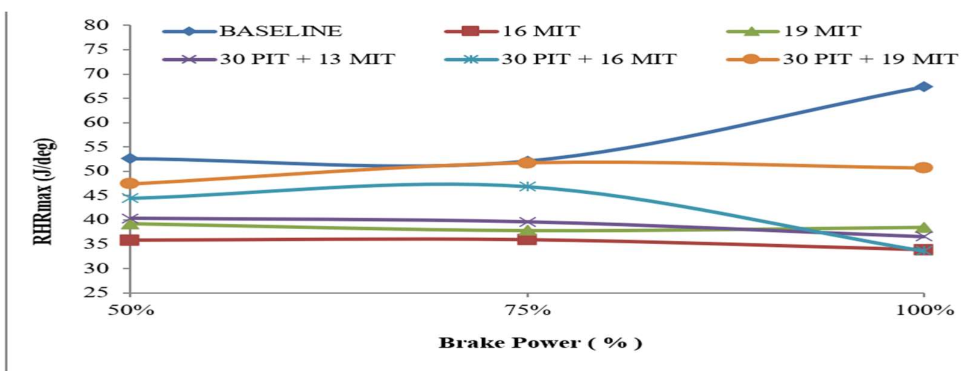
BASELINE
7 bTDC
10 bTDC
13 bTDC
16 bTDC
19 bTDC
RHRmax (J/deg)
Brake Power (%)
50%
75%
100%

(a)



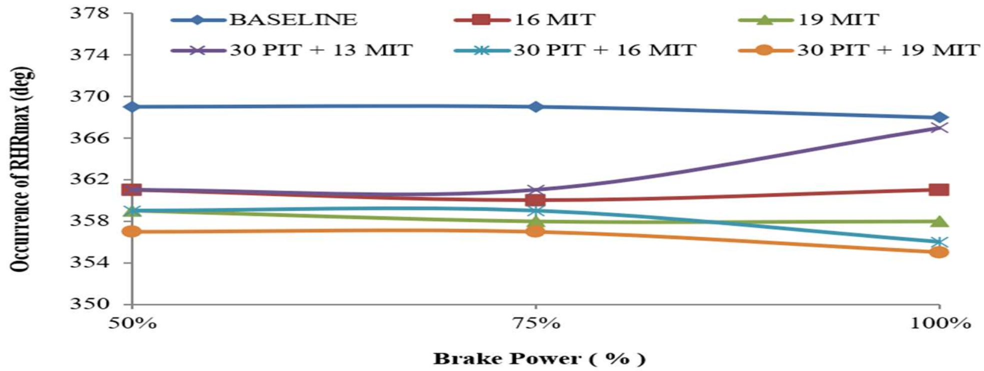
BASELINE
7 bTDC
10 bTDC
13 bTDC
16 bTDC
19 bTDC
RHRmax (J/deg)
Brake Power (%)
50%
75%
100%

(b)

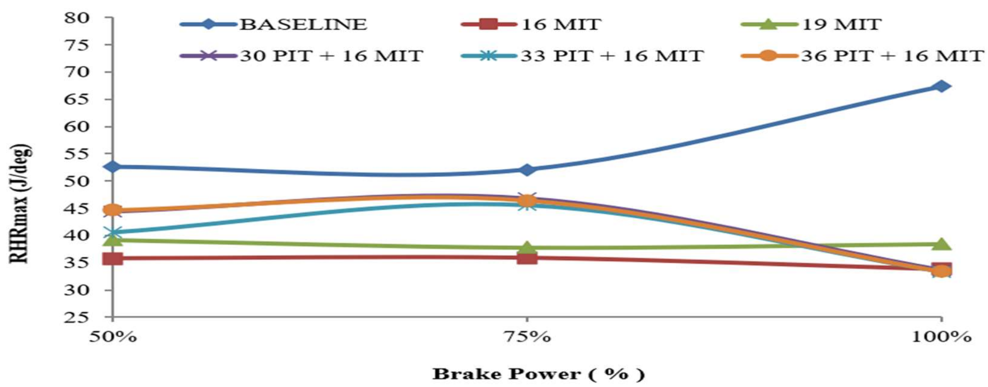


BASELINE
16 MIT
19 MIT
30 PIT + 13 MIT
30 PIT + 16 MIT
30 PIT + 19 MIT
RHRmax (J/deg)
Brake Power (%)
50%
75%
100%

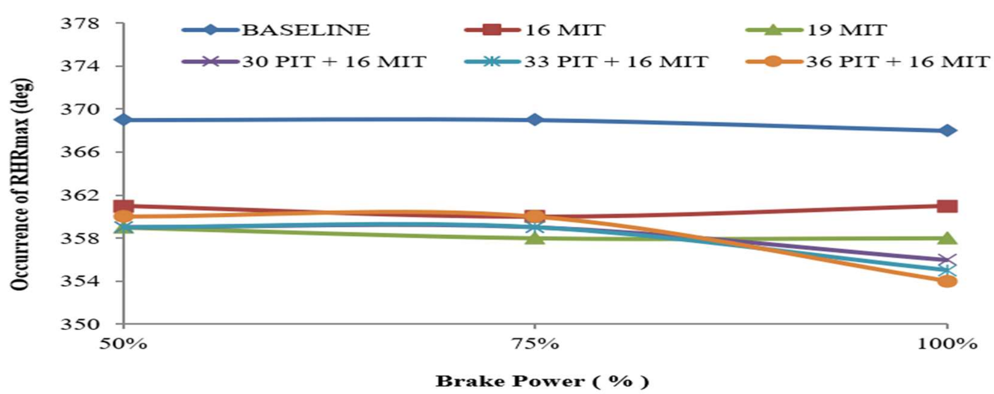
(c)



(d)



(e)



(f)

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

CHAPTER 5

CONCLUSIONS

The present study dealt with performance and emission analysis of a VCR single cylinder water cooled constant speed engine, CRDi incorporated system equipped with open ECU. The ECU equipped system enabled to precisely control the injection timings and FIPs. These modifications enabled online control of FIP, SIT, injection pulses, injection rate, and injection duration more precisely under varied engine operating conditions. The impact of delayed SITs and split injections on various combustion, performance and emission parameters of this modified engine was experimentally investigated at constant fuel injection pressure of 300 bar.

With the use of delayed start of injection timings and high injection pressure (300 bar) it is observed to reduce NOx and HC emissions simultaneously with increase CO and Smoke emissions. Also, the occurrences of peak CP, PRR and HRR values shifted away from TDC, and their peak values also improved at all loads.

Increase in injection interval caused a deterioration of the BSFC with decrease of Break Thermal Efficiency. Retarding the SOMI from 19^0 bTDC to 13^0 bTDC in the intervals of 3^0 with SOPI at 30^0bTDC shows decrease of NOx, HC emissions to greater level, simultaneously with deteriorated CO and smoke. CP, PRR and HRR are improved at all loads compare to base line and their occurrence shifted away from TDC.

Retarding the SOPI from 36^0 bTDC to 30^0 bTDC in the intervals of 3^0 with fixed main injection timing at 16^0bTDC it is observed that deterioration of the BSFC with decrease of Break Thermal Efficiency. The NOx emissions are improved but HC, CO and Smoke emissions are increased compared to base condition. Also, the occurrence of CP, PRR and HRR move away from TDC and the peak CP, PRR and HRR are improved.

According to analysis, large injection interval and main injection timing (SOMI) around top dead center shows little deterioration of BSFC and improvement in emissions especially NOx. i.e., 30^0 bTDC (SOPI) and 13^0 bTDC (SOMI).

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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