Characterizing Impact of Fuel Quality with Multiple Injection Strategies on CRDI CI Engine Emissions & Performance

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Department of Mechanical and Aerospace Engineering

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Abstract

The experimental investigation was led for a better understanding about the reduction of exhaust emissions in CRDI diesel engine such as NOx, HC and CO employing the multiple injection strategies. Two multiple injection strategies were used: *pilot injection* and *split injection.* It was revealed that maximum value of combustion pressure in two pilot injections was increased to almost the same level of single injection combustion although its maximum heat release rate (HRR) was decreased compared to single injection combustion. It was also observed that two pilot injections improves combustion efficiency, based on the results of increased IMEP (0.8% and 2.4% under medium and high load respectively). Moreover, in early pilot injection combustion, more CO formation and less HC emission were observed during combustion process. Reduction of NOx up to 22.23% and 6.7% was recorded under medium and high load.

Remarkable reductions of 41.5% and 63.8% were observed in NOx levels with *split injection* technique but the combustion pressure and HRR values also degraded due to the retarding of SOI timing to limit the combustion noise occurring because of discontinuous combustion.

Availability analysis of the various processes involved in engine operation under different injection strategies are also reported.

Nomenclature

AFR	Air-Fuel ratio
ATDC	After top dead centre
A_{tm}	Thermo-mechanical availability
b.m.e.p	Brake mean effective pressure
bsfc	Brake specific fuel consumption
BTDC	Before top dead centre
CA	Crank angle
CI	Compression Ignition
CR	Compression ratio
CRDI	Common rail direct injection
CO	Carbon monoxide
CO2	Carbon dioxide
COV	Coefficient of variation
DI	Direct injection
ΔA	Change of availability of the system
ECU	Electronic control unit
HC	Unburned hydrocarbons
IC	Internal combustion engine
IMEP	Indicated mean effective pressure
HRR	Net heat release rate
NOx	Oxides of nitrogen
PM	Particulate matter
SOI	Start of injection
TDC	Top dead centre

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1 Introduction:

1.1 Background

The internal combustion engines were first developed in 1876 when Nicoulas A. Otto introduced the first spark-ignition engine and, in 1892 Rudolf Diesel invented the first compression-ignition engine. The internal combustion engines have continued to evolve in development since that time as there has been a continuous increment in our knowledge of engine and combustion processes, as new technologies became available, as the variety of fuels got introduced, as the emissions legislation gets more stringent day by day. The internal combustion engines have a wide area of application, they have been extensively used in transportation (land, sea and air) facilities and power generation units. As the population of the world is increasing at a very fast rate, there has been a tremendous increment in the use of IC engines over past several decades for transportation as well as power generation sectors, enormous increase in the number of vehicles have started dominating over the demand of fuel.

Diesel engine usually operates on the overall lean equivalence ratio over the entire operating range. So it offers greater thermal efficiency, but produces the greater amount of exhaust emissions like particulate matter and smoke. The major emissions, which generates from a diesel engine are NOx and particulates. Particulates are the solid carbon soot particles that are generated in the fuel-rich zones within the cylinder during combustion in spite of the lean operating conditions due to very low volatility of the diesel fuel. However, UHC and CO emissions are quite less owing to the lean operating conditions. Alternate bio-fuels such as alcohols and bio-diesels have drawn attention of many researchers owing to their ability to significantly reduce particulate matter emissions, without seriously penalizing the NOx, unburned HC and CO emissions. Former researchers have experimented many alcohols, biodiesels, their esters, ethers to investigate the impact of these oxygenates on the performance and emissions of diesel engine[1]–[8]. A common conclusion is that with the increment in the oxygen concentration of the fuel PM decreases, especially at high loads. At this condition maximum amount of fuel is injected into the combustion chamber to supply maximum power but due to low volatility of the diesel fuel results in local rich regions. Carbon spheres are generated where the mixture is locally rich and there is not enough oxygen to convert all the carbon to CO2. Fuel with oxygenated additives can effectively introduce more oxygen into those locally rich mixtures and convert the left out carbon atoms into CO_2.

HC emissions are generally due to the unburned fuel components from the combustion. But, due to overall lean burning conditions HC emissions are very low in diesel engine. Some HC particles condenses onto the surface of the solid soot particles generated during combustion, Most of it gets burned as the combustion proceeds, only a small amount comes unburned into the exhaust. This contributes to the HC emissions of CI engine. CO emissions are generated when the engine operates with a fuel-air rich equivalence ratio, when there is scarcity of oxygen to convert all carbons to CO_2. But due to lean burning characteristics of diesel engine CO emissions are very low in CI engine operation.

At low temperatures nitrogen exists as a stable diatomic molecule. However, at high temperatures some diatomic nitrogen breaks down to monoatomic nitrogen atoms 'N'. Other gases which are stable at lower temperatures such as oxygen and water vapour becomes reactive at higher temperatures and contribute to the formation of NOx. Considerable amount of N is generated at the very high temperatures of about 2500 K which is possible inside an IC engine. The higher the combustion temperature, more and more diatomic nitrogen will break down to monoatomic nitrogen atom and more NOx will be generated. Although, maximum flame temperatures occur at a stoichiometric air-fuel ratio. When the equivalence ratio is close to stoichiometric conditions and slightly lean flame temperatures are very high and there is an excess of oxygen to react with the monoatomic nitrogen atoms to form NOx.

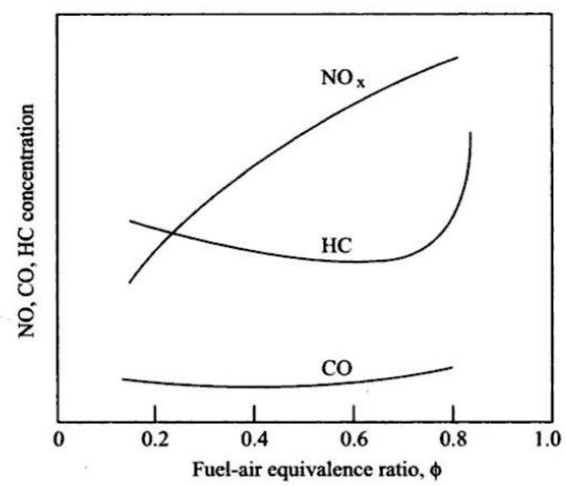


Figure 1: Emissions as a function of Equivalence ratio for a CI engine[9].

Figure 1 shows the qualitative picture of HC, CO and NOx emissions with respect to the equivalence ratio for a four stroke DI diesel engine. HC will decrease slightly with the increase in the equivalence ratio due to higher cylinder temperatures, making it suitable to burn up any locally lean or rich region. However, at high loads HC may rise again if the fuel in regions is too rich to burn. Due to excess air CO emissions will always be very low. NOx will increase gradually with the increase in the equivalence ratio due to increasing fraction of cylinder contents being burnt gases close to stoichiometric during combustion, and also due to higher peak temperatures and pressures.

Because of a conjunction of factors, such as environmental pollution, increment in demand and depleting of petroleum products, increasing oil prices, development of alternate fuels for petroleum products has become a subject of great concern for many governments and vehicle manufacturers around the world. Owing to the fact that petroleum fuels are not renewable and a constant and very fast increase in the automobile density across the world, a day would come in the near future when the pressing requirements for these fuels would exceed the supply, triggering a notable world crisis. One more reason for the development of alternative fuels is the fact that a large fraction of the petroleum is imported from the other countries and the majority of oil fields are associated with problems- both political and economic.

Multiple injection technique has been widely investigated by the researchers as a measure of reducing the NOx and the combustion noise as well[10]–[12]. The ignition delay in a CI engine is defined as the interval between the start of injection and the start of combustion. In a multiple injection strategy, a small amount of fuel is injected prior to the main injection in the form of one or two pilot injection during the compression stroke which gives comparatively an improved fuel-air mixture than the conventional single main injection technique, when this small amount of fuel injected combusts results in slight increase in combustion chamber temperature prior to the main injection which further leads to reduction of ignition delay and decreased pressure rise rate due to smooth pressure rise rate. Therefore, decreases the NOx formation and combustion noise.

1.2 Motivation

The statistics show that both the population and demand for energy are growing with the increase in worldwide population the global demand for energy has also at even more faster rates every year. The trend is shown in Fig. 2. The major fuel source of the IC engines are the fossil fuels which have started to deplete slowly. With the increment in the demand and depletion of fossil fuels, Alternative fuels have become part of the major research in the field of IC engines.

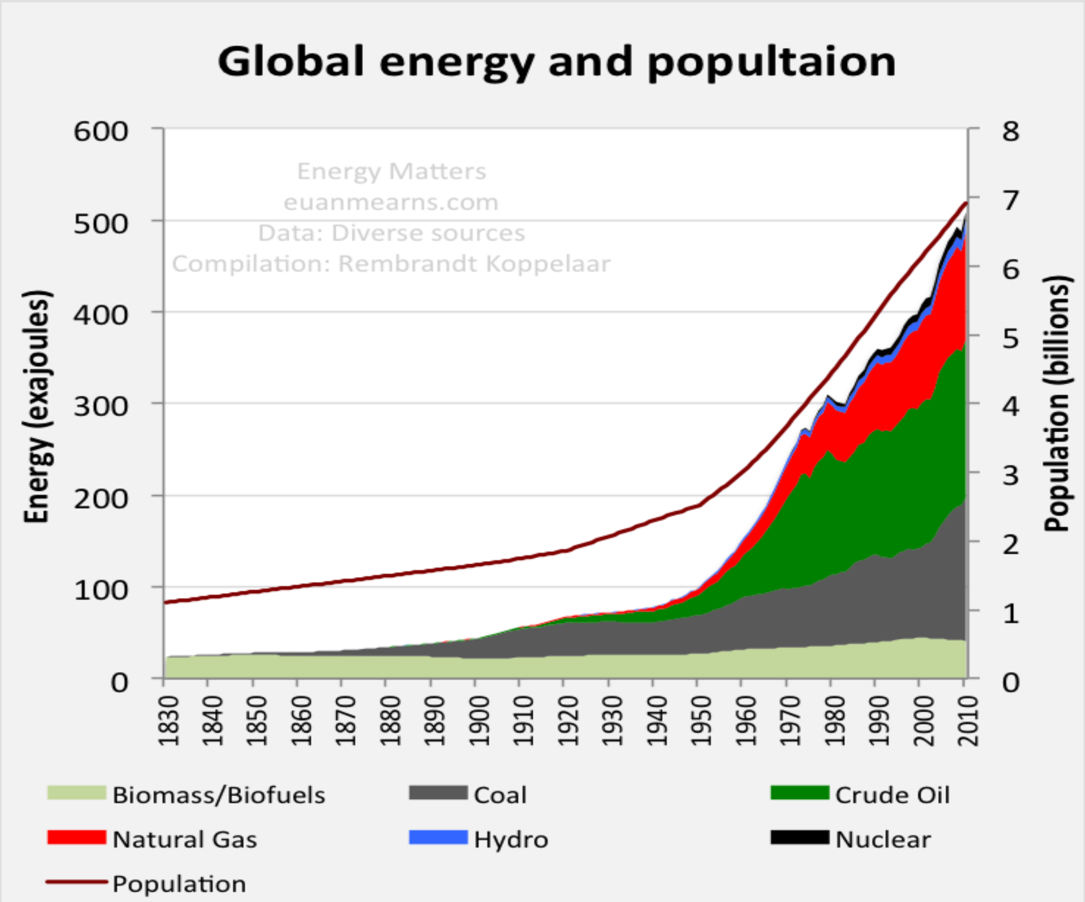


Figure 2: Year-wise global population and energy consumption.

The increasing urbanization and mechanization of the world has led to a sudden rise in the environmental degradation and demand of fuels derived from petroleum products. Petroleum products are extracted from crude oil and there are limited reserves in the world from which the crude oil is obtained. These reserves are mostly concentrated in certain areas in the world. Therefore, the countries which are not having these resources are facing energy foreign exchange crisis, mainly due to the import of crude petroleum. Hence, it is necessary to look for alternative bio-fuels which can be produced from resources available locally within the country such as alcohol, vegetable oils, biodiesel etc. Owing to different chemical and physical properties these biofuels and their blends with diesel fuel are capable in reducing exhaust emissions while maintaining a satisfactory engine performance[1]–[8]. The direct advantage of using bio-fuels are reduction in the petroleum fuel quantity used and reduction in exhaust emissions.

Gasoline and diesel engine are the two most commonly used IC engines. Diesel engine having advantages of higher compression ratio and overall lean equivalence ratio running conditions offers greater thermal efficiency than the gasoline engine, but induces higher exhaust emissions such as NOx and particulate matters. There has been a noticeable drop in the fuel economy compared to the earlier models because the modern diesel engines need to be designed in order to meet even more stringent future emission legislations. These restrictions have provoked the researchers and corporations to explore advanced technologies to control diesel emissions. One such technology is multiple injection technique. Multiple injection technique has been widely investigated by the researchers as a measure of reducing the NOx and PM emissions simultaneously[10]–[13], as well as combustion noise[10]. The aim of this research is to evaluate the technique of multiple injection with diesel-oxygenated fuel blend to check whether the combination potentially reduces the exhaust emissions while maintaining a decent engine performance.

1.3 Objectives of research

This study focuses on analysing the performance, combustion and the exhaust emissions of the compression ignition engine equipped with the common rail diesel injection system employing the 'multiple injection' strategies. In this study two different multiple injection strategies (split injection and pilot injection) are employed. The effect on combustion, performance and emissions of the engine has been carried out by varying the injection timings, injection rates and injection concentrations under the employed strategies. Second law (Availability) analysis is applied to each process and sub-process to determine the combustion exergetic efficiency and the availability transfer.

This study will help in providing a comparison between engine performance, combustion and exhaust characteristics of a single main injection and the multiple injection strategies. Availability analysis will provide an idea about the exergetic quantification of the multiple injection strategies.

2 Literature Review

2.1 Fuel Blending

Auto mobiles are the major contributors to the increasing air pollution across the world. As the emissions legislation is getting more and more stringent day-by-day diesel engines need to further reduce the emissions to meet the legislation, this fact has led the researchers and companies to search for the advanced technologies. Lot of efforts have been gone into for mitigating the emissions of diesel and gasoline fuelled engine. However, more efforts are needed for the improvement of the ever-increasing air-pollution due to automobile population. The world is facing a dual challenge of fossil fuels depletion and environmental deterioration. Alternate fuels which can provide a balanced correlation between energy conservation and environmental preservation has become highly essential due to oil crisis and stringent emission legislation throughout the world. Several alternative energy resources have been investigated by the researchers in order to get the solution for the obstacle of ever increasing energy thirst of the world's population. Studies demonstrate that the blends of bio-fuels (alcohols[1]–[5] and biodiesels and their esters[6]–[8]) and diesel reduce the engine exhaust emissions like CO, UHC, oxides of nitrogen, particulates and soot maintaining the performance comparable to that of fuelled with diesel fuel. Engines fuelled with alternate fuels have always been in use since the discovery of the IC engines in small numbers. Because of the high cost of the petroleum products, some developing countries are trying to use alternate fuels for the vehicles.

2.1.1 Alcohols in CI Engines

Alcohols are an appealing alternate fuel because they can be prepared from both renewable and non-renewable sources. Apart from the manufacturability alcohols have some other advantages which include[9] lesser exhaust emissions compared to diesel fuel, higher flame speeds, high latent heat of vaporization reduces the in-cylinder combustion temperature which results in lower NOx emissions, oxygen content which results in reduction of soot particles. Despite having many advantages it has some disadvantages which restrict the direct use of alcohols in IC engines such as lower Cetane number, low energy content, produces more aldehydes, poor cold weather starting characteristics. But to exploit the advantages of alcohol it can be utilized in IC engines as fuel blends Regulations to lower the NOx emissions are getting stringent year by year. Studies show that, Alcohols when used as a blend with diesel fuel up to a great deal reduces the particulate emissions without inducing a serious penalty on NOx [2]–[5].

Rakopoulos et al.[4] investigated the impacts of using the blends of n-butanol and conventional diesel fuel on a single cylinder Ricardo/Cussons 'Hydra', naturally aspirated diesel engine with a constant speed of 2000 rpm at 19.8:1 compression ratio. It was demonstrated from the results that exhaust smoke (soot) density emitted by the diesel-butanol blends was significantly lower than the one emitted by the conventional diesel. The reduction in soot density is higher with the increasing percentage of butanol in the blend fuel because of the enrichment of oxygen content even in the locally fuel rich zones in the combustion process. NOx levels for the blends were also slightly lower than the base fuel, which can be explained with the slightly lower exhaust gas temperatures recorded with the fuel blends in the experiments compared to nest diesel, the reductions being higher with the higher percentages of butanol in the blend. There was a slight increase in brake thermal efficiency with the blends. Figure 3 demonstrates the soot density and the nitrogen oxides (NOx) exhaust emissions for the neat diesel and the various concentrations of butanol blend.

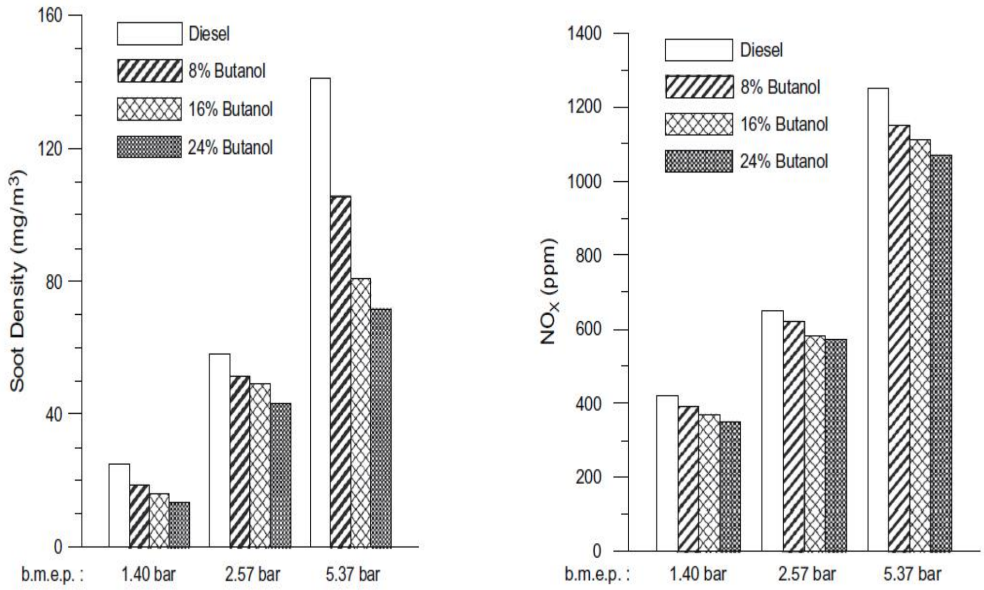


Figure 3: soot density and NOx emissions[4].

Another study from Rakopoulos et al.[5] investigated the comparison of effects of ethanol and n-butanol blends with diesel fuel on the combustion behaviour of a turbocharged, six cylinder 'Mercedes Benz' engine at engine speeds of 1200 and 1500 rpm at compression ratio of 18:1. The key results demonstrated a significant reduction in particulates (smoke opacity) which is due to the oxygen enrichment of the locally fuel rich regions which leads to the oxidation of carbon atoms. A very slight decrease in NOx emissions is because of the slightly lower combustion temperatures being dominant over the recorded increase in the ignition delay with the increasing concentration of biofuels in the blends. Figure 4 shows the NOx and soot emissions.

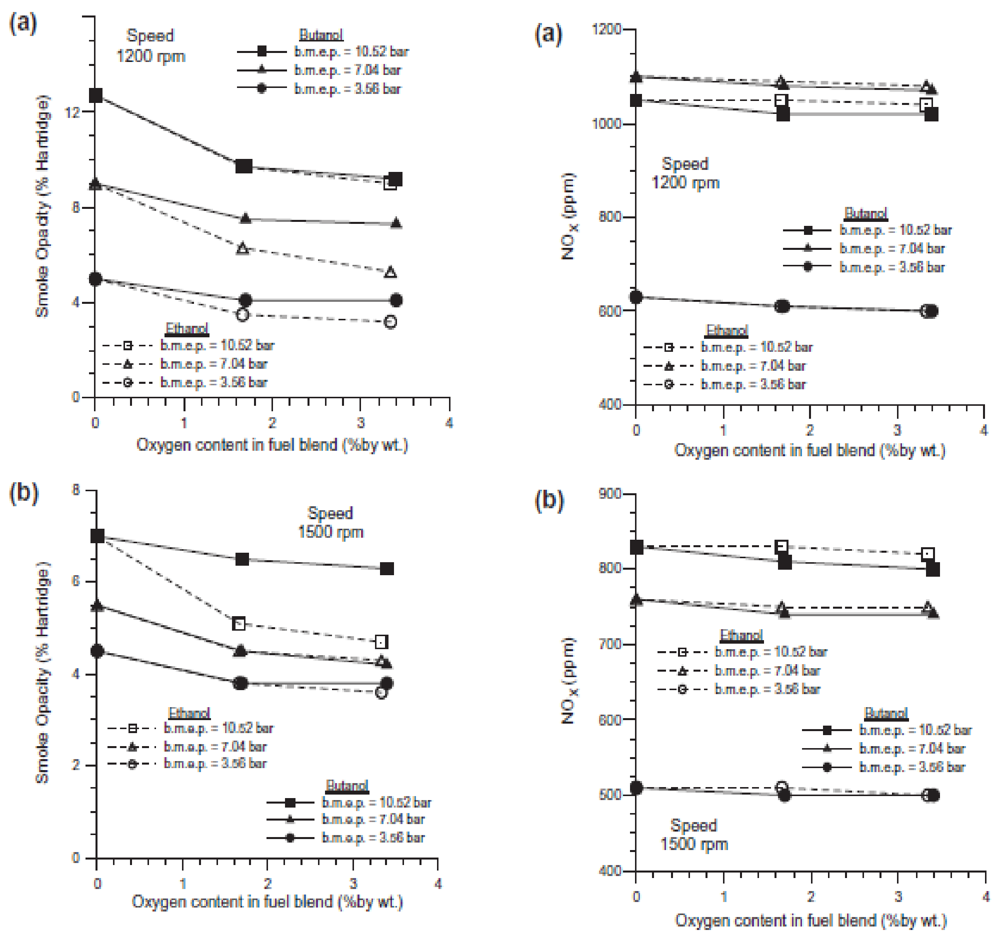


Figure 4: smoke opacity and NOx emissions[5].

Some properties of n-butanol makes it a suitable alternative fuel for CI engines as compared to methanol and ethanol. The Cetane number and Calorific values of n-butanol is higher as compared to the other widely used alcohols such as methanol and ethanol. Water affinity characteristic of n-butanol is low which allows it to easily mix with petroleum fuels. As it can be observed from the Fig. 5 that almost all the properties of n-butanol are closer to the diesel fuel as compared to methanol and ethanol. Generally speaking, (1) methanol and ethanol have very low cetane numbers of only 3 and 8, and the latent heat values are much higher than the diesel fuel. These attributes prolong the ignition delay, and result in much higher pressure rise rate, and worse engine cold start performance,(2) the auto-ignition temperatures of methanol and ethanol are also much higher than the diesel fuel thus it is necessary to add combustion improver to help ignition. (3) solubility of methanol and ethanol to diesel are not very good.

Properties of diesel fuel, ethanol and *n*-butanol.

Fuel properties	Diesel fuel	Ethanol C_2H_5OH	*n*-Butanol C_4H_9OH
Density at 20 °C (kg/m^3)	837	788	810
Cetane number	50	~8	~25
Lower calorific value (MJ/kg)	43	26.8	33.1
Kinematic viscosity at 40 °C (mm^2/s)	2.6	1.2	3.6[a]
Boiling point	180–360	78	118
Latent heat of evaporation (kJ/kg)	250	840	585
Oxygen (% weight)	0	34.8	21.6
Bulk modulus of elasticity (bar)	16,000	13,200	15,000
Stoichiometric air–fuel ratio	15.0	9.0	11.2
Molecular weight	170	46	74

[a] Measured at 20 °C.

Figure 5: Fuel properties[12].

2.1.2 Biodiesels in CI Engines

Many researchers have reported that with the use of edible oil ester as a fuel in diesel engines, harmful exhaust emissions can be reduced while maintaining a satisfactory engine performance[6]–[8]. Most of the esterified oils tried in diesel engines were soybean, sunflower, safflower, and rapeseed. Vegetable oils having high cetane number and calorific values near to diesel fuel can be directly used in CI engines without any major modifications. However, due to high viscosity and very low volatility vegetable oils result in lower brake thermal efficiency than diesel fuel, because of poor atomization of fuel and mixing with air. Several other methods have been tried by the researchers to utilize vegetable oils efficiently in diesel engines. Some of them are transesterification, blending with diesel fuel and dual fuelling with gaseous and liquid fuels.

M. Senthil Kumar et al.[6]experimentally compared the different methods to use methanol and Jatropha oil on a single cylinder, direct injection, CI engine at a constant speed of 1500 rpm and compression ratio 15:1. Three different methods of using methanol with Jatropha oil were employed which are blending, transesterification and dual fuel operation. From the experiments, reduction in smoke with all three methods and minimum with dual-fuel operation was recorded, NO level compared to Diesel fuel was lower with Jatropha oil. With the increased amount of methanol in dual-fuel operation Brake thermal efficiency increased in comparison to the standard diesel operation but showed poor performance at lower outputs.

H. Raheman and A.G. Phadatre[7] demonstrated the results of investigations carried out in studying the fuel properties of karanja mthyl ester and its blend with diesel fuel. The karanja methyl ester and its blends with diesel were used to test a single cylinder, DI, water-cooled diesel engine having a rated output of 7.5 KW at 3000 rpm and compression ratio of 16:1. The investigation performed reported the decrease in exhaust emissions together with increase in engine performance parameters like torque, brake power, brake thermal efficiency and reduction in brake-specific fuel consumption.

2.2 Multiple Injection

Nitrogen oxides are the major emissions generated in the exhausts of diesel engine along with particulates (soot). A substantial reduction in the NOx emissions in the diesel engine exhaust, with low levels of PM, HC and CO is a major issue in the field of IC engine, due to the stringent emission legislations, which is further getting stricter year by year. Further reduction of NOx from the diesel engine exhaust has prompted many researchers to explore advanced technologies. Multiple injection technique has be researched by many researchers as a means of reducing the NOx and PM emissions together[10]–[12].

Hotta et al.[10] investigated the effects of multiple injection strategies on a single cylinder diesel engine with low to high loading conditions with different speeds and injection pressures. Figure 6 shows the effect of pilot injection timing on engine performance and emissions. IMEP starts dropping after 40 BTDC and increases after 40 BTDC that can be directly explained by the plot of equivalence ratio after 40 BTDC the equivalence ratio is almost unity. The stoichiometric conditions will generate higher combustion temperatures and pressures leading to higher IMEP. HC emissions are more when equivalence ratio is close to one. ‘Early’ pilot injection will result in proper mixing of fuel and air, and a mild pressure rise rate whereas ‘close’ pilot injection will result in the accumulation of pilot injection with the main injection and when together it will combust will give a higher pressure rise rate that explains the ‘noise’ trend.

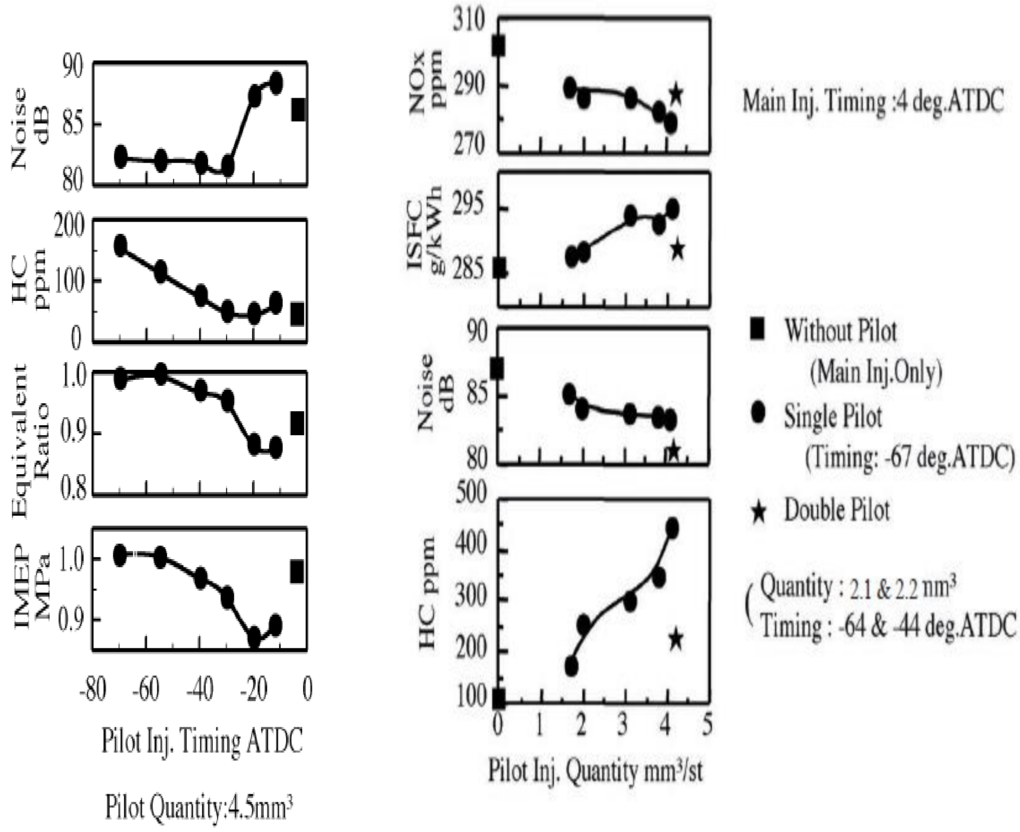


Figure 6: Effects of pilot injection strategies[10].

At medium loads, splitting up the single pilot injection into double injection results in lower noise, HC and CO emissions, but slightly higher NOx emissions for the same quantity of fuel as shown in Fig. 6 because splitting up the pilot injection will reduce the fuel adhered to the combustion chamber walls.

Hyun Kyu Suh[11] conducted an experimental analysis for understanding the combustion stability and reduction of exhaust emissions in a single cylinder low compression ratio engine. Analysis demonstrated that single injection had the highest values of heat release rate and combustion pressure among different injection strategies.

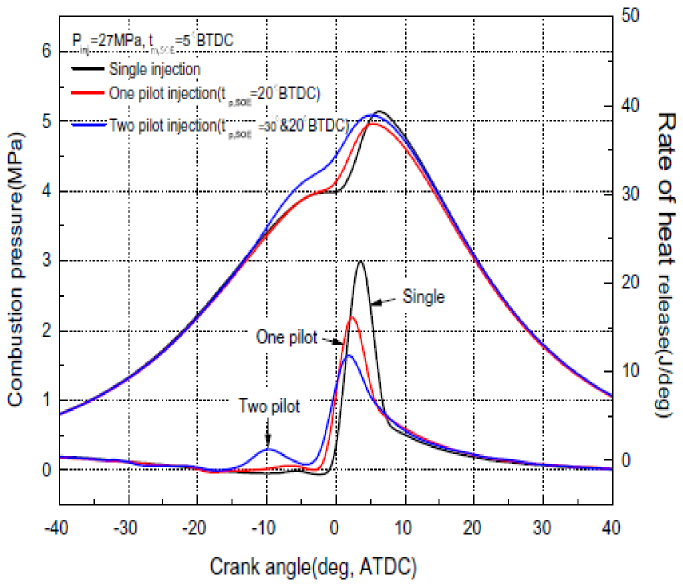


Figure 7: Effect of pilot injection strategies on combustion parameters[11].

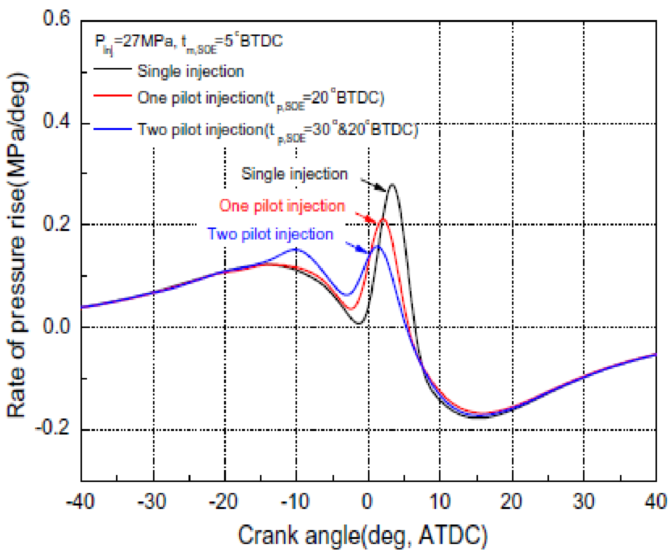


Figure 8: Effect of pilot injection strategies on Pressure-rise rate[11]

Pilot injections will result in locally fuel rich regions which will result in increase of CO emissions. HRR rate decreased to 27.8% and maximum pressure also reduced to 3.2% compared to single injection. Pressure rise rate of two pilot injection is the fastest because some fuel is injected before main injection, however it shows lowest peak pressure values. Multiple injection improves the complete combustion of main injection, thus, less unburned HC emission remained during combustion process. Remarkable reduction of NOx emission was observed in multiple injection combustion, 58.9% in two pilot injection. 25 % lesser soot formation in both injections due to lower combustion temperatures resulting from decreased ROHR of multiple injections.

Experimental study was conducted by Yao et al.[12] to investigate the influence of the diesel fuel n-butanol content on the performance and emissions of a heavy duty direct injection, six cylinders diesel engine with multi-injection capability at constant engine speed of 1849 rpm and compression ratio 16:1. Technique of multiple injection was employed for the experimentations which consisted of pilot-main, main-post and pilot-main-post injection strategies. Exhaust gas recirculation was employed to keep the NOx emissions at 2.0 g/KWh. The results demonstrated similar impact of pilot and post injection of blends as diesel fuel. Early pilot injection reduces soot emissions but increases CO, because the early pilot injection will result in proper mixing of air and pilot fuel which will provide comparatively rich mixture than no pilot condition. Post injection reduces soot due to increase in the temperature just after the main combustion which results in oxidation of the left out carbon soot particles to CO and CO2. Addition of n- butanol significantly reduced CO and soot emissions without penalizing the *bsfc* and NOx emission and the triple injection strategy with highest n-butanol fraction resulted in lowest soot emission due to oxygen content of the butanol blends.

2.3 Availability Analysis

Diesel engines have the maximum thermal efficiency among IC engines. However, recent emission standards have considerably reduced the fuel economy compared to the past models. The first law of thermodynamics gives the quantitative measure of the process, whereas the second law deals with the quality of the process in the system. In simpler words, the efficiency based on the first law shows the completeness of the combustion, but the extent of the conversion of the chemical energy of fuel into mechanical work is indicated by the second law.

The availability of a system at a given state is defined as the amount of useful work that could be obtained from the combination of the system and its surrounding atmosphere, as the system goes through reversible processes to reach thermal, mechanical and chemical equilibrium with the atmosphere. It is a property of the system and its surrounding atmosphere. The entire fuel energy cannot be converted to the useful mechanical work because the availability of the fuel gets destroyed due to irreversible processes during combustion. Exergetic efficiency by taking the irreversible losses into account represents the effectiveness of the fuel conversion to work.

Sahoo et al.[14] performed the Availability analysis on the experimental data recorded from a four cylinder direct injection engine of Bajaj make Tempo, model D-301 at a CR of 19.4:1. The main objective was to observe the effect of throttle opening position on engine speeds at a single engine load operation for several testing combinations. Saleel Ismail and Pramod S. Mehta[15] discussed the method of estimating the availability destructions and exergetic efficiencies of combustion for different classes of fuels using the MATLAB simulation of the combustion in constant volume and constant pressure processes.

3 Experimentation

3.1 Experimental Setup

Figure 9 shows the two cylinder optical access research engine having an operational range of 400 RPM to 1300 RPM. Compression ratio ranges from 6.7 to 18. A hydrodynamic dynamometer is coupled to the engine through which load is applied on the engine shaft. One out of two cylinder can work at a particular time, one is thermodynamic cylinder and other cylinder is optically accessed. The requirements of this study does not include the optical diagnosis of the combustion process, so the optical accessed cylinder is cut off throughout the experimentation. The combustion chamber geometry is toroidal bowl in a piston top, which ensures compact combustion chamber and faster burning. 4 inlet valves (2 per cylinder) and 4 exhaust valves are driven by overhead camshafts equipped with the engine.

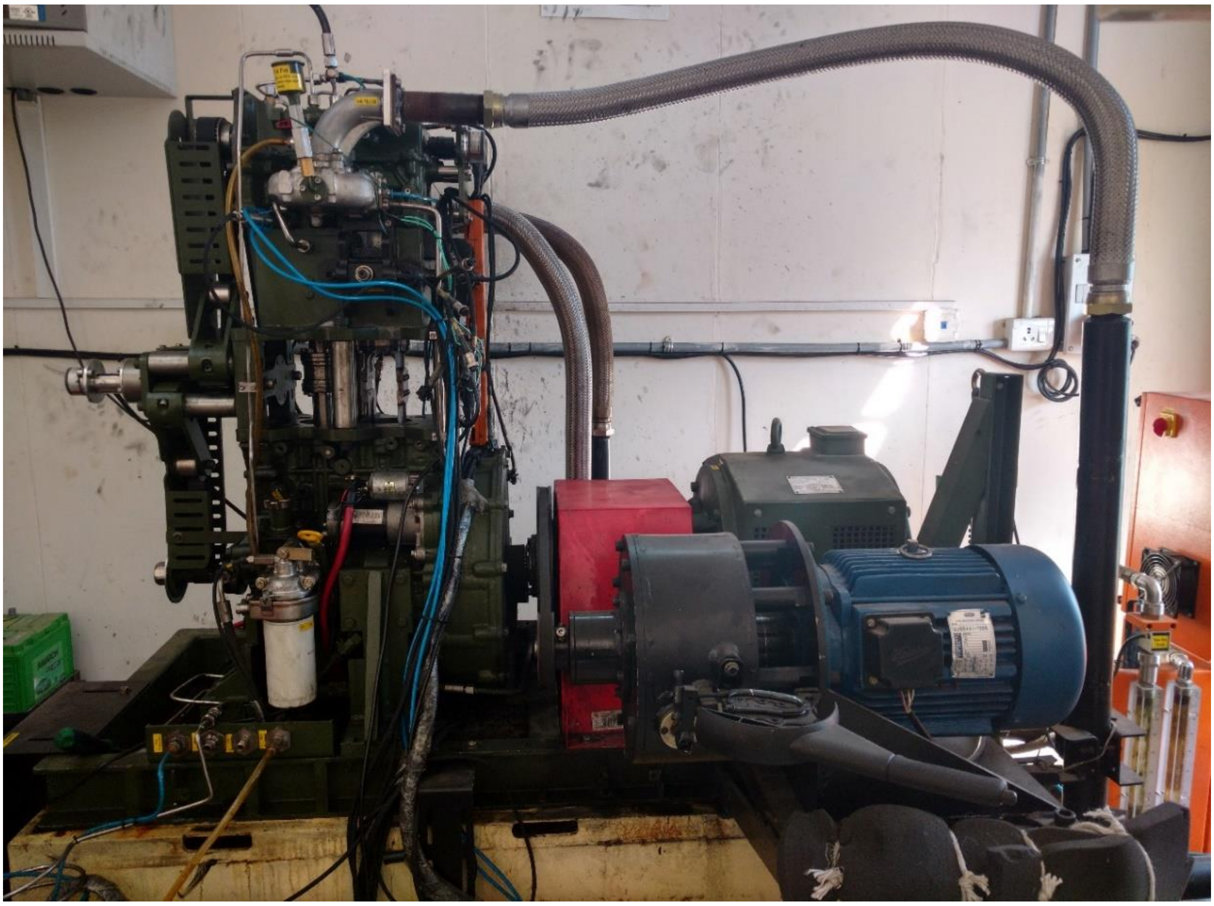


Figure 9: Optical access research engine

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

No of running cylinders	1 out of 2
Stroke (mm)	100
Bore (mm)	94
Connecting rod length (mm)	235
Compression ratio	18:1
Speed range (rpm)	1050±30
Inlet open (degree)	5 ATDC
Inlet close (degree)	21 ABDC
Exhaust open (degree)	25 BBDC
Exhaust close (degree)	9 BTDC
Injection system	CRDI
Injection pressure (bar)	500
Injection timing (degree)	-90 before start of intake stroke

Table 1: Specifications of research engine

An air box instrument measures the air flow rate, wherein, through an orifice plate air passes from a large volume box, the pressure drop is measured across the orifice is measured. For the accurate measurement of the air flow this pressure drop signal is fed to the ECU. The air-fuel ratio, NOx, THC, CO2 and CO emissions were measured using AVL.

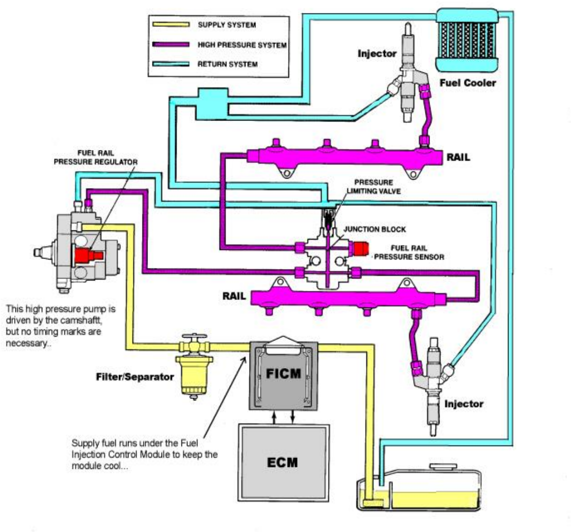
An automatic volumetric fuel flow meter measures the fuel flow rate injected into the combustion chamber. It contains two sensors, one at the top and another at the bottom of a 100 ml measuring burette. The fuel is filled in this burette and the time is recorded for emptying the burette, and then fed to the ECU. The ECU calculates the mass flow rate of fuel based on density of fuel which is fed to it manually. The fuel properties are provided in table 2.

Fuel property	**Value**
Name	Diesel fuel
Cetane rating	~50
Density (kg/m^3)	830
Calorific value (kJ/kg)	43000

Table 2: Properties of a fuel

The measurement of the in-cylinder pressure is done using a piezoelectric pressure transducer, which is fitted to the cylinder head. It collects the gas pressure through a passage drilled in the cylinder head, up to the centre of the cylinder, open to the combustion chamber. An eddy current type dynamometer is coupled directly to the crankshaft of the engine for application and measurement of load. The load range varies from 0kg to 10 kg.

The engine is equipped with common rail diesel injection system. It consists of CRDI driver module and CRDI kit, which controls injection pressure, injection timing and duration. Fuel injection pressure ranges from 200 to 1000 bar. The schematic diagram of CRDI system is shown in Fig. 10.



(a) (b)

Figure 10: CRDI system. (a) Schematics of fuel flow in CRDI; (b) CRDI kit

The data acquisition software is developed by legion brothers. The system allows real-time, on screen display of the measured and recorded parameters such as in-cylinder pressure, exhaust gas temperature, cooling water temperatures to the engine and the calorimeter. It also displays calculated parameters such as specific fuel consumption, air-fuel ratio, volumetric and brake thermal efficiency. Pressure data is recorded for 100 consecutive cycles and averaged, on every test point. The parameters like IMEP, peak pressure etc. are calculated for each cycle and then averaged.

3.2 Experimental Procedure

3.2.1 Test methodology

It is advisable to read the operating and safety manual of research engine, provided by the engine supplier before embarking any work on an engine first time. It is necessary to attend certain daily check points for better and uninterrupted operation, before cranking the engine.

For steady state operation, it is required to warm up the engine approximately for 10 minutes.

3.2.2 Test procedure

This study focuses on evaluating the performance, combustion and emissions parameters of the diesel engine employing different multiple injection strategies at different loads and constant speed. All the engine tests are conducted at the speed of 1050±30 rpm and the fuel injection pressure of 500 bar at a compression ratio of 18:1. Firstly, the experiments have been done to characterise the engine based on performance and emissions with multiple loads at 500 bar injection pressure, then the optimum start of injection is decided based on the experimental results.

3.3 Injection Strategies

Multiple injection strategies have been researched extensively as a measure of simultaneous reduction in PM and NOx emissions, along with combustion noise. To achieve cleaner exhaust emissions, two different types of multiple injection strategies including *pilot injection* and *split injection* have been incorporated in a diesel engine operation and the results based on combustion, performance and emissions were compared with that acquired from single injection operation. The injection strategies employed for the experiments and the fuel distribution is shown in Fig. 11.

3.3.1 Pilot Injection

The prime objective of pilot injection is to minimize the NOx emissions along with combustion noise from the CI engine. When a small quantity of fuel is injected prior to the main injection, it burns and increases the temperature and pressure inside the combustion chamber which results in reduction of the ignition delay of main injection. The main injection combustion takes place at a lower peak temperature compared to single injection which lowers the NOx formations. Single and double pilot injections are used in the experiments. Pilot quantity is kept fixed in the experiments which is 20% of the total fuel input per cycle.

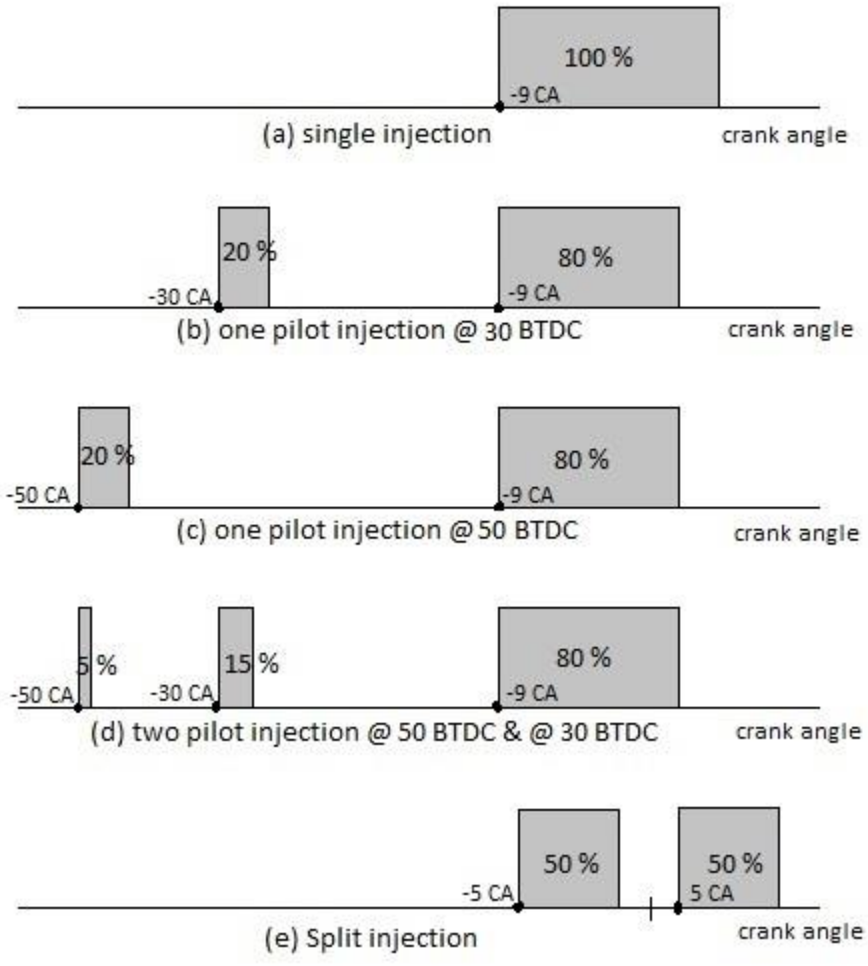


Figure 11: Injection timings and fuel distribution for different injection strategies

3.3.2 Split Injection

Injection retardation reduces the NOx emissions in CI engine. A split injection strategy with retarded SOI is introduced to investigate the impact on the emissions while maintaining the satisfactory engine performance. An interval of 10º CA was introduced between the first and second injections to prevent the interaction between them. First injection timing is 5 BTDC and the second injection is timed at 5 ATDC.

4 Results and Discussions

4.1 Single Injection

In a diesel engine, retarding the injection timing has been a mutual means of controlling NO*x* emissions, by reducing the combustion temperature and pressure. However, overly retarded injection timing causes growing engine instability; sometimes, the engine can misfire. Therefore, an appropriate injection timing that concurrently satisfies exhaust emission and engine performance limitations should be picked for the engine. In this study, to assess the impact of multiple injection strategies on the CRDI Diesel engine, the experimental results of multiple injection strategies were compared with those acquired by single injection with the best suited injection timing that simultaneously fulfils exhaust emission and engine performance requirements.

For determining the optimum start of injection timing for a single injection, various points have been chosen for the SOI under different loads ranging from 2kg to 8kg by the dynamometer. Table 3 is representing the crank angles at which the SOI has been checked. The load and the corresponding b.m.e.p values are shown in the Table 4.

SOI Tested (in CA)
4 BTDC
7 BTDC
9 BTDC
12 BTDC
15 BTDC
18 BTDC

Table 3 : Tested SOI timings

Load (in dynamometer)	Torque (Nm)	B.M.E.P (in bars)
2 kg	4.8	0.87
4 kg	9.51	1.72
6 kg	14.65	2.65
8 kg	18.15	3.28

Table 4: B.M.E.P values for corresponding load values.

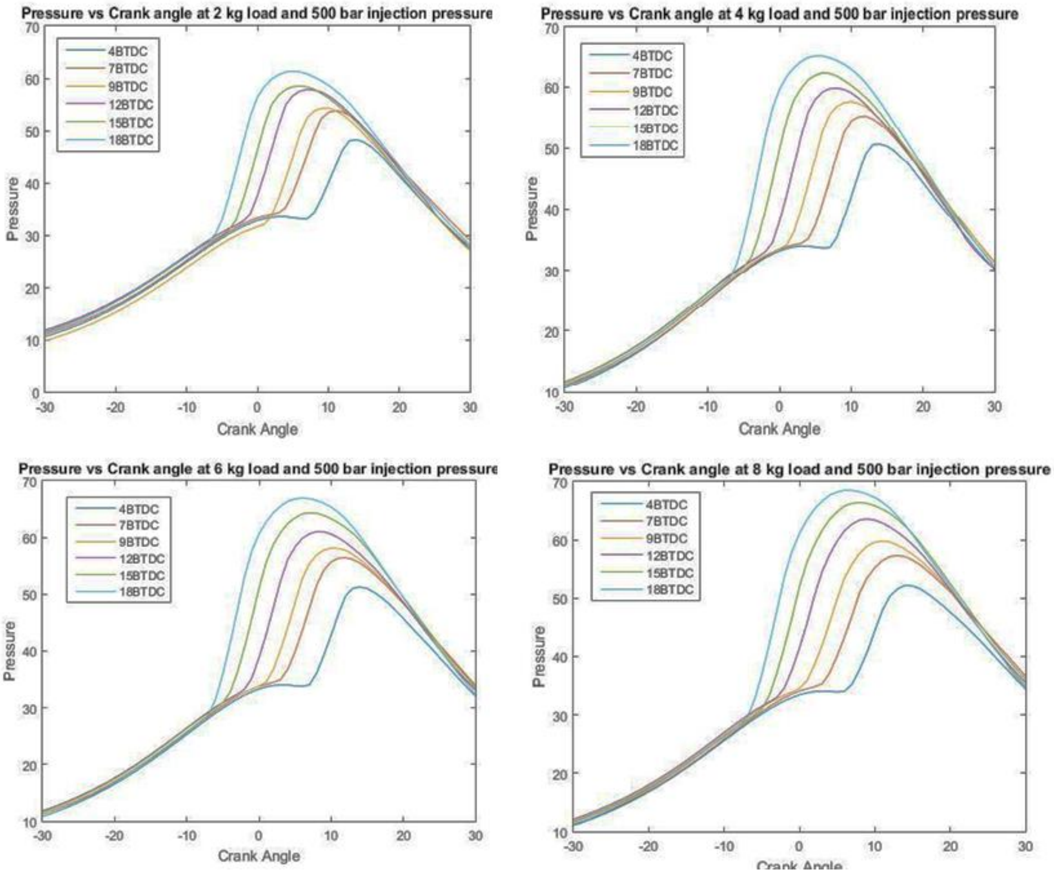


Figure 12: Pressure vs. crank angle curves for different SOI at various loads.

The pressure vs. crank angle data is shown in Fig. 12. As shown from the figure for 15 BTDC and 18 BTDC the combustion starts early and the peak pressure values lie nearer to TDC. Which shows that the premixed combustion phase for these 2 SOI lies in compression stage which will increase the negative work on the engine which is not suitable for the economy and due to rapid burning pressure rise rate will be high which will increase the temperature inside the combustion chamber resulting in higher NOx formation. So it is concluded that 15 BTDC and 18 BTDC are not the optimum SOI for the research engine from emissions as well as economic point of view. Figures 13 and 14 represents the IMEP, COV and the emissions data for the 4, 7, 9 and 12 BTDC. In figures the IMEP and COV plots are shown for the medium (4 kg) and high (8 kg) load. The trend for both IMEP and COV increases with the advancement in the SOI, but after 9 BTDC it does not vary much.

Figure 16 shows NOx emissions reduce with the retardation of the SOI **[13]**, same trend can be seen here as well NOx emissions with 4 BTDC injection timing are the lowest and starts increasing with the advance of SOI timings due to start of combustion shifting towards TDC which results in more temperature inside the combustion chamber. Figure 17 shows HC emissions are lowest with 9 BTDC SOI. Considering both emissions and IMEP trends 9 BTDC comes out to be the most optimum SOI for the engine having lesser NOx and other emissions plus lesser variations in COV and higher IMEP values than the other later SOI timings.

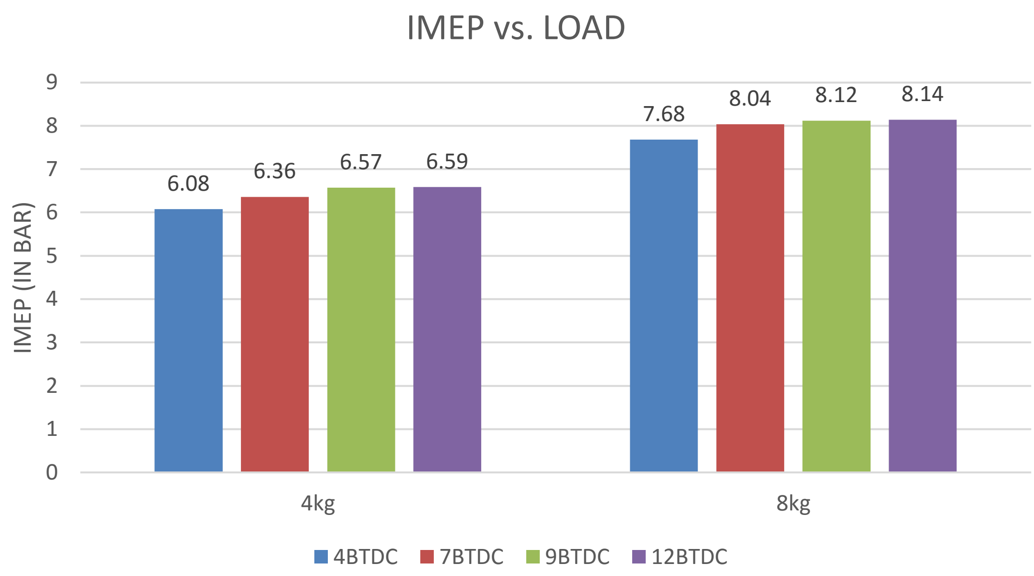


Figure 13: IMEP vs. LOAD curve for different SOI timings.

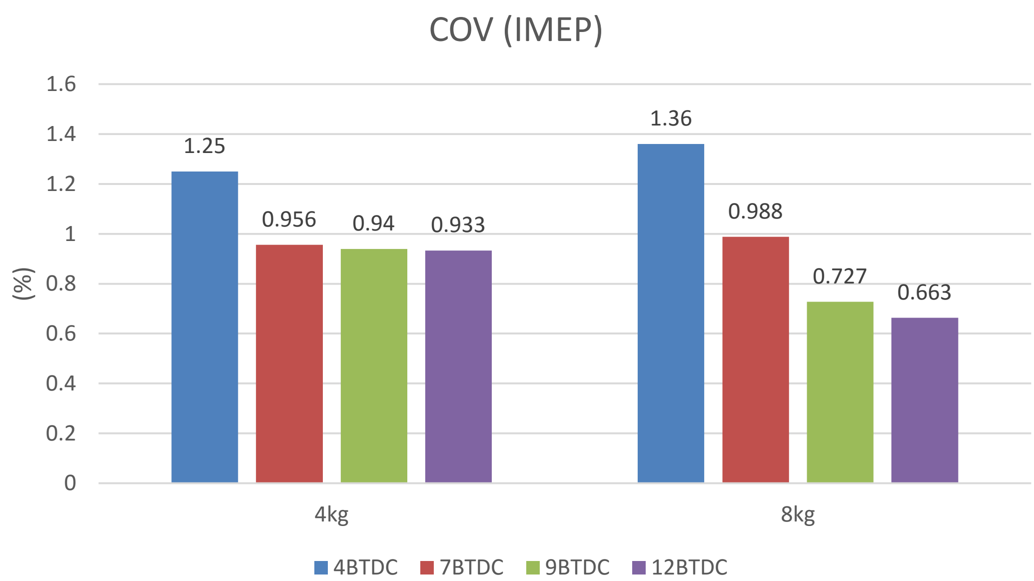


Figure 14: COV vs. LOAD variation with different SOI timings.

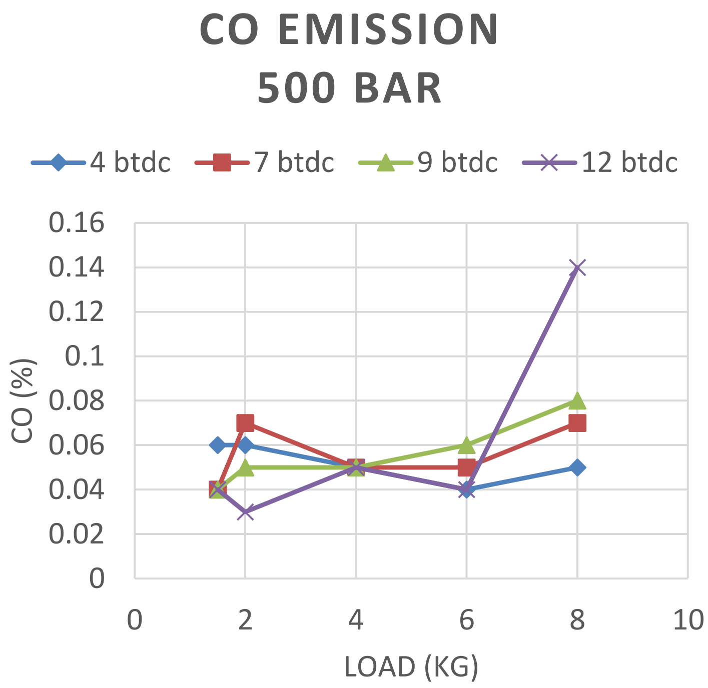


Figure 15: CO emissions with different SOI timings.

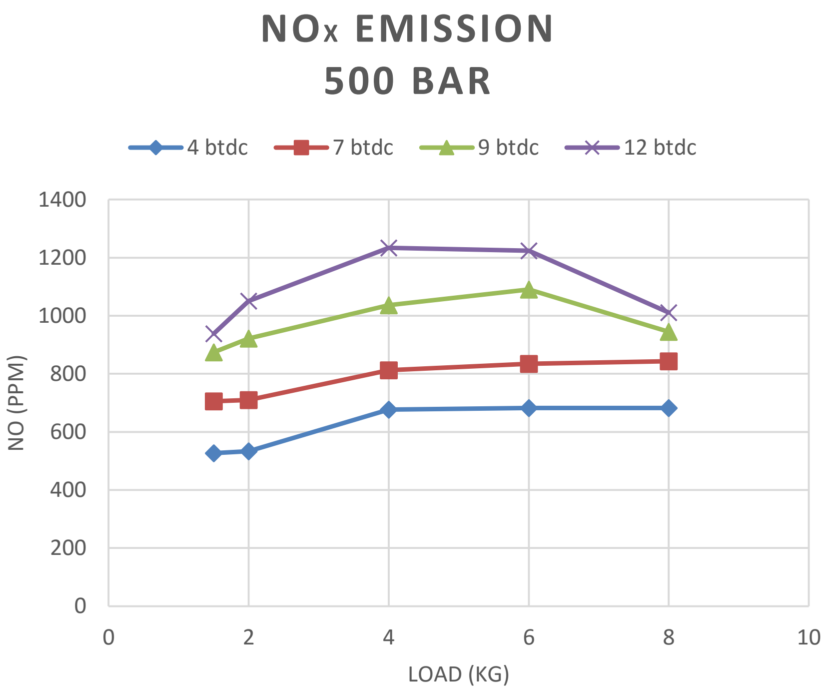


Figure 16: NOx emissions with different SOI timings.

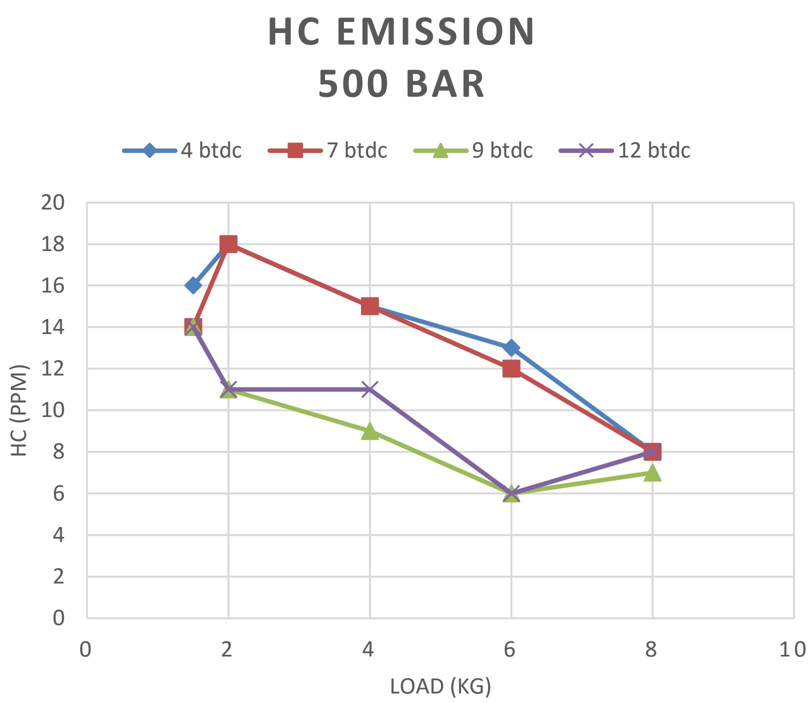


Figure 17: HC emissions with different SOI timings.

4.2 Multiple Injections

4.2.1 Effects on Combustion Performance

The combustion pressure and heat release rate (HRR) under different injection strategies under 4 kg load are shown in Fig. 18 and 19. The multiple injection strategies for fuel injection selected are single injection, one pilot with early and medium timing pilot injection, double pilot injection and split injection. Since most of the fuel gets injected during the ignition delay period in single injection, rapid premixed combustion takes place and the injected fuel is combusted immediately as soon as the mixture reaches the auto-ignition temperature resulting in harmful emissions and combustion noise. The peak pressure for the double pilot injection was slightly reduced as compared to the other multiple injection strategies. Ignition delay of the main injection in the cases of one pilot injection with 30 BTDC and double pilot is reduced, because when the pilot injection combusts, it increases the temperature and pressure inside the combustion chamber prior to main injection which shortens the ignition delay. For early pilot injection i.e. pilot at 50 BTDC the peak pressure values are the maximum because the fuel quantity in the early injection does not combust due to low temperature, most of the fuel gets accumulated inside the combustion chamber and some fuel gets stuck on the cylinder walls. When the main injection fuel quantity fuel comes it gets mixed with the premixed pilot-air mixture and combusts at a very faster rate resulting in high premixed combustion phase which leads to higher temperature inside the cylinder. The combustion pressure and the HRR values are lowest with the split injection technique because of the retardation in the SOI timing and the discontinuous combustion. The SOI timing in split injection technique were retarded to limit the combustion noise from the engine.

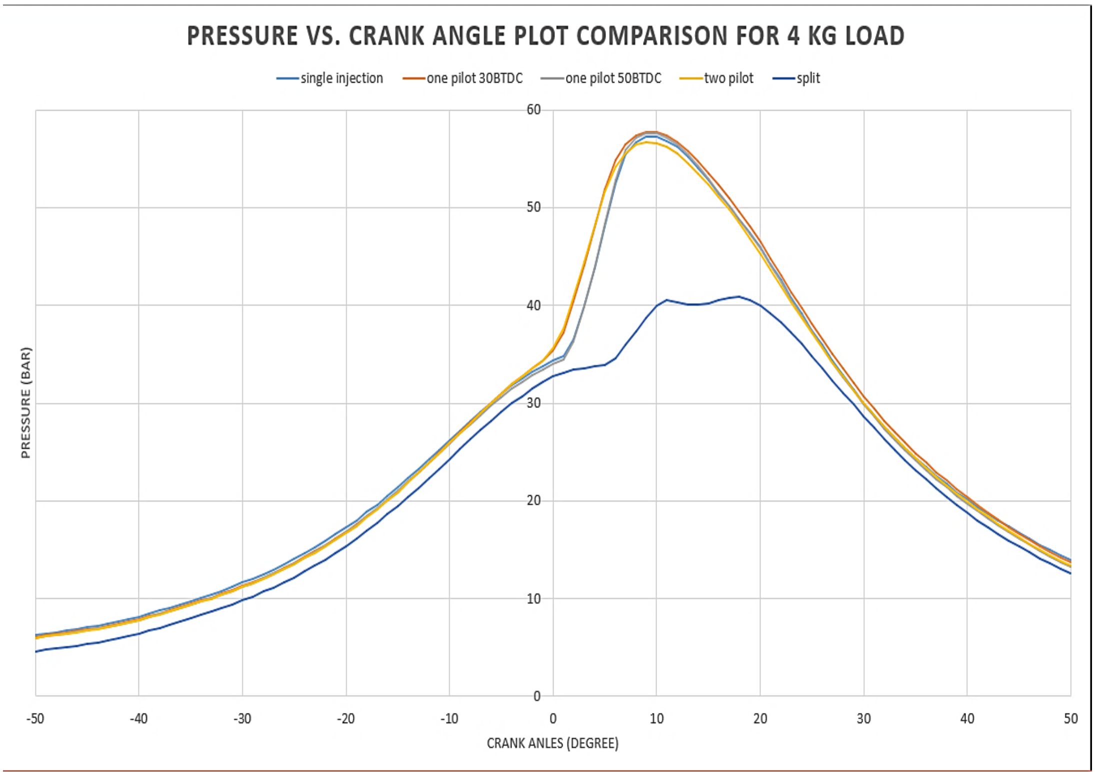


Figure 18: Pressure data for 4kg load with different Multi-injection strategies.

Figure 19 shows the HRR data. With pilot at 30 BTDC and double pilot injection, as it was expected the maximum HRR decreased because of increase in the temperature and pressure prior to main injection. In double pilot injection, air fuel mixture becomes rich due to fuel injections two times as compared to one pilot injection which results in less rapid combustion, that's why the maximum HRR decreased more with double pilot than the single pilot at 30 BTDC. Maximum HRR was observed during early pilot injection even more than the single injection case due to accumulation of the pilot fuel and combined burning with the main injection which leads to rapid combustion phase. Split injection due to discontinuous combustion and the retarded SOI timing due to which the combustion occurs later during the expansion stage which results in lesser combustion pressure and maximum HRR.

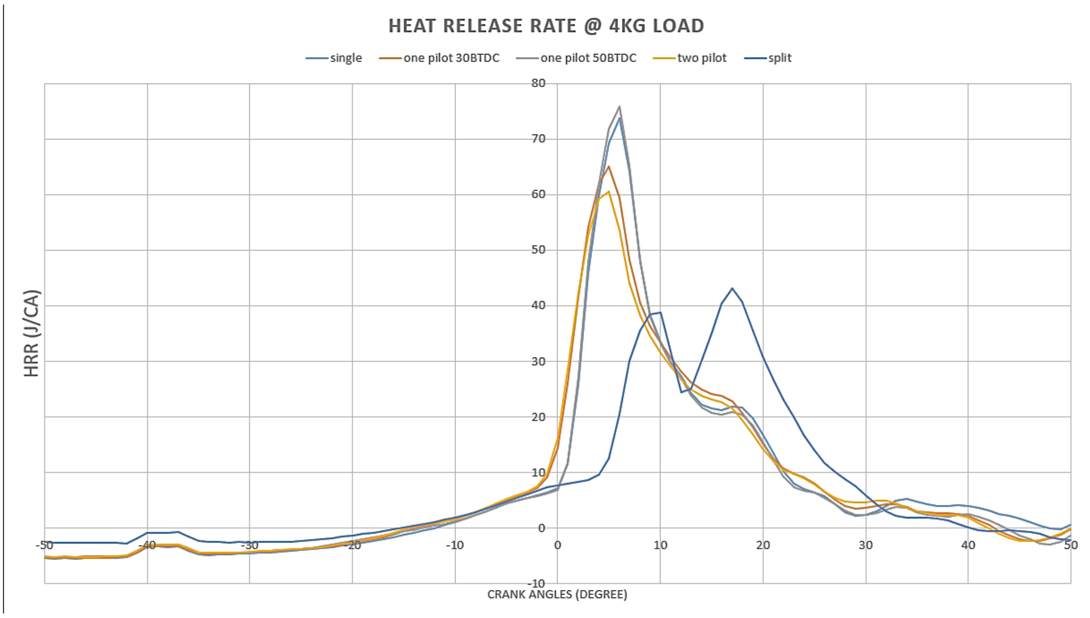


Figure 19: HRR data for 4 kg load with different Multi-injection techniques.

Figure 20 shows the pressure rise rate during the combustion under different multiple injection strategies. The pressure rise rate is fastest with the double pilot injection and the peak value is also lower than other pilot injections and single injection due to enhanced combustion because burning of two pilots with some interval results in smoother pressure rise rate. As discussed already early pilot injection has the highest rate of pressure rise.

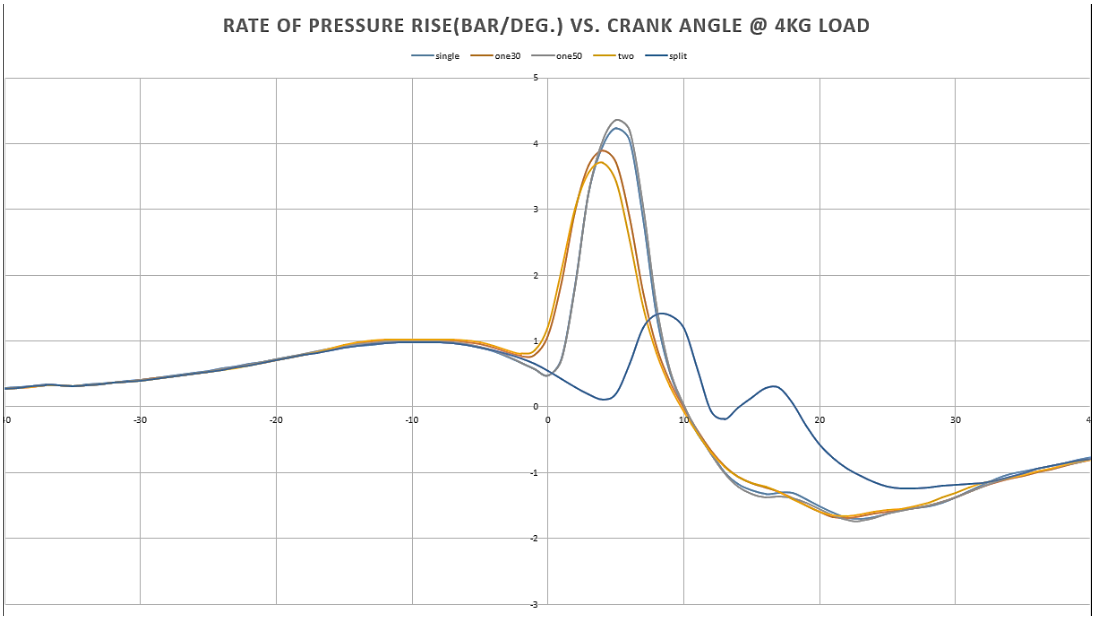


Figure 20: Pressure rise rate for 4kg load with different injection strategies.

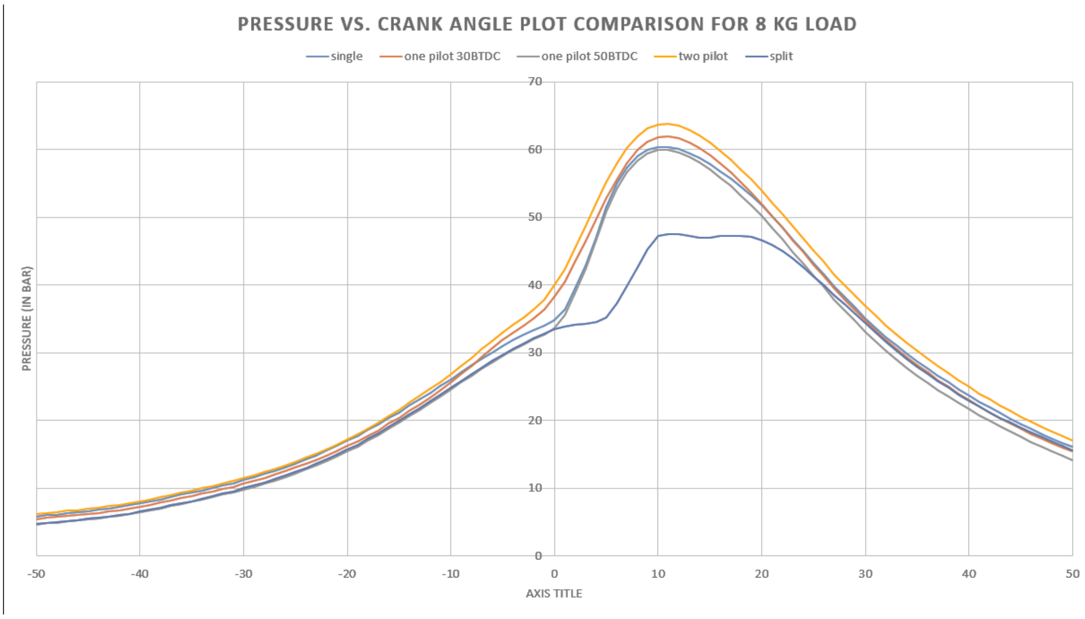


Figure 21: Pressure data for 8kg load with different Multi-injection strategies.

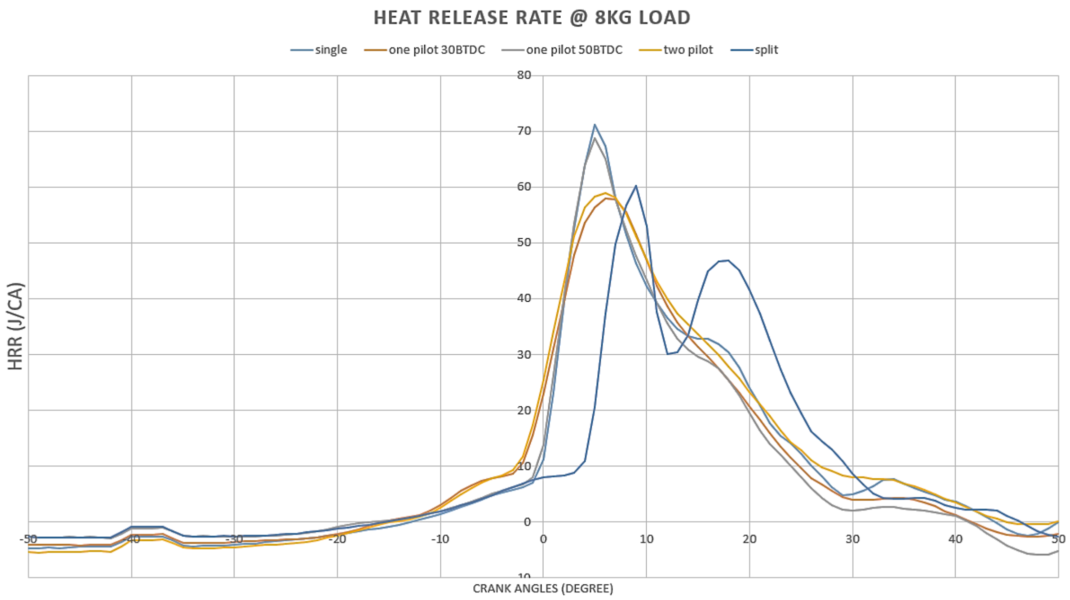


Figure 22: HRR data for 4 kg load with different Multi-injection techniques.

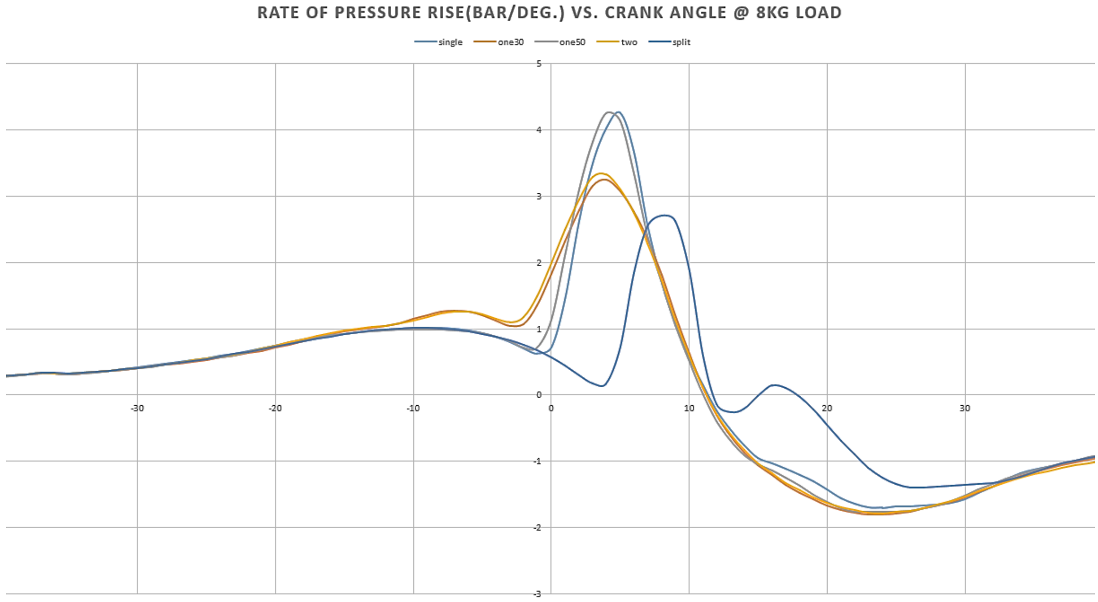


Figure 23: Pressure rise rate for 8kg load with different injection strategies.

Figures 21, 22 and 23 represents the combustion pressure, HRR and pressure rise rate curves for different injection strategies. It is evident from all three curves that due to high temperature with high loading condition there is further reduction in the ignition delay for all the strategies. In Fig. 22 and 23 the combustion of pilot injection becomes prominent at high load with 30 BTDC one pilot injection and double pilot injection. The bump in the HRR and pressure rise rate curves prior to main injection represents the pilot injection combustion due to which there is an increment in pressure and temperature which significantly shortens the ignition delay period which results in smoother pressure rise and lower peaks of pressure rise rate. Due to higher temperatures at high load there is a slight reduction in delay period of early pilot injection resulting in slightly lesser HRR and pressure rise rate than single injection.

	4kg load	**8kg load**
Single injection	9	7
One pilot @ 30 BTDC	7	6
One pilot @ 50 BTDC	9	7
Two pilot injection	7	6
Split injection	9	8

Table 5: Ignition Delay (in CA) for different injection strategies under different loading.

Table 5 shows the ignition delay values with the multiple injection strategies under different loadings. The injection delay is significantly reduced in one pilot at 30 BTDC and the double pilot injection strategies for the same reason as for the reduction in HRR and pressure rise rate. The delay period of early pilot injection is almost same as of single injection because of the accumulation of fuel inside the combustion chamber and split injection strategy also shows more delay like single injection due to retarded injection timing which results in lesser combustion pressure.

4.2.2 Effects on Exhaust Emissions and IMEP

NOx emissions are shown in Fig. 24. The trend observed under both the loading conditions are almost same. As expected the split injection strategy shows the minimum levels out of all other multiple injection strategies due to the drastic decrease in the combustion pressure because of the retardation in SOI timing to limit combustion noise under both loading conditions. With the same main injection timing of 9 BTDC, double pilot injection shows the best improvement in NOx emissions and one pilot at 30 BTDC remains close to it. Due to the smoother pressure rise rate in both the strategies premixed combustion phase is quite limited which also decreases the HRR values producing lesser combustion temperature than single and early pilot injections. Pilot fuel in early pilot injection does not combusts prior to the main injection to reduce the ignition delay but prepares a better air fuel mixture than the single injection resulting in slightly reduced premixed combustion and ultimately reduced NOx levels.

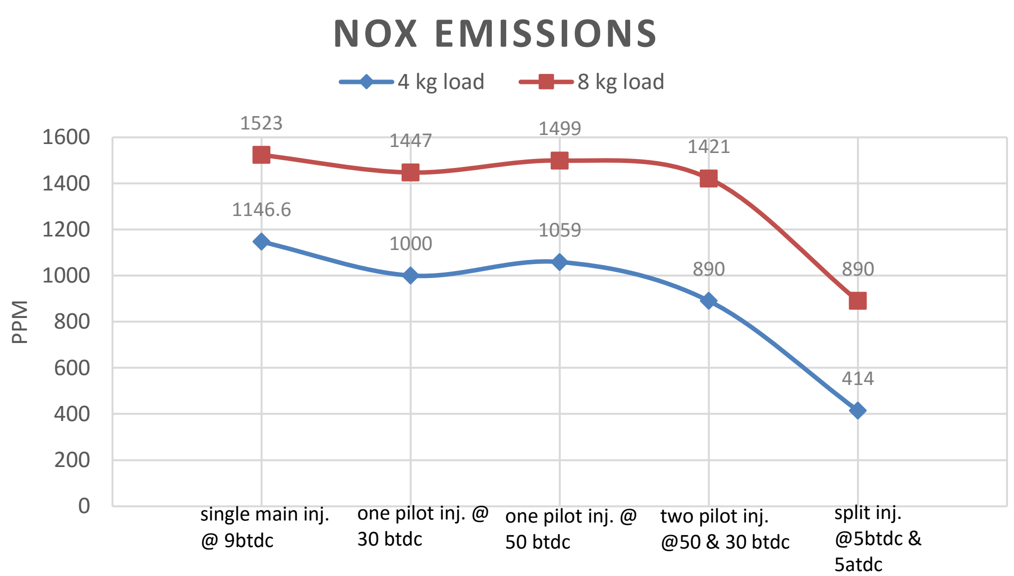


Figure 24: NOx emissions with different injection strategies.

HC emissions originate in the regions where excessively diluted air prevents the combustion process form either starting or going to completion. The early pilot injection shows the highest levels of HC emissions under both loading conditions which can be observed from Fig. 25. The reason behind more HC emissions with early injection is the sticking of the pilot fuel to cylinder walls and the flame quenching due to which that stuck quantity remains partially burned and comes out in the engine's exhaust.

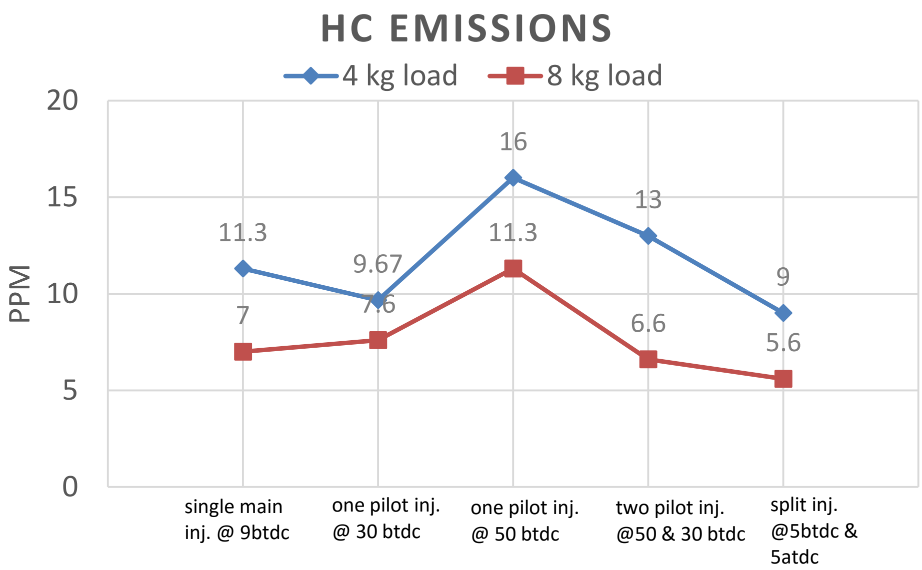


Figure 25: HC emissions with different injection strategies.

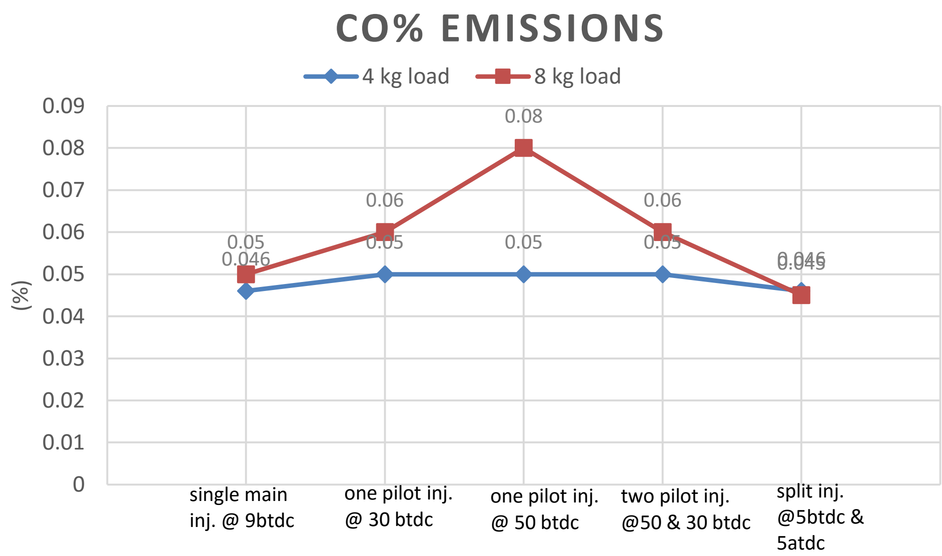


Figure 26: CO emissions with different injection strategies.

The CO emissions are shown in Fig. 26. More CO formations were observed in multiple injection case compared to single injection because the lower combustion temperature during the multi-stage combustion event induces the low CO oxidation rate.

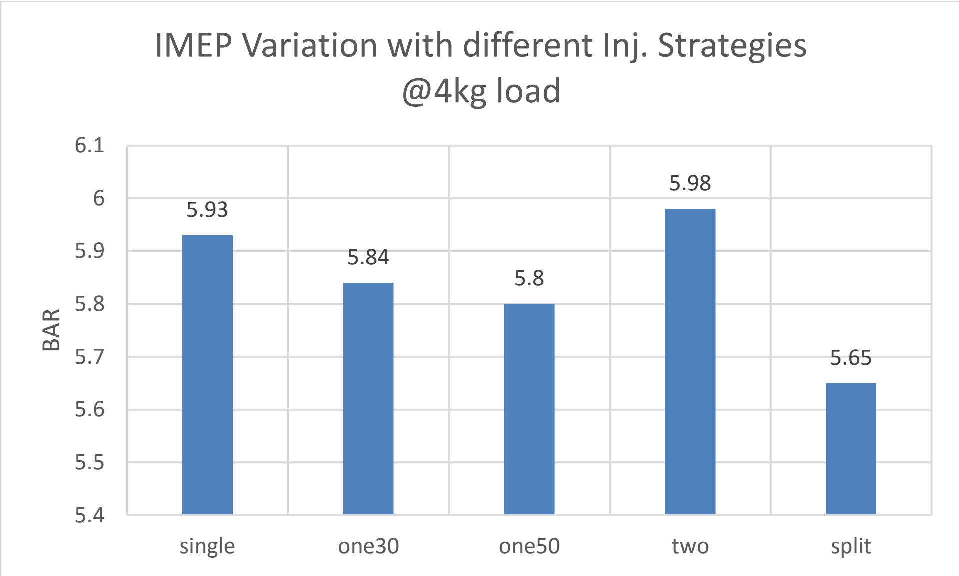


Figure 27: IMEP variation with different injection strategies at 4kg load.

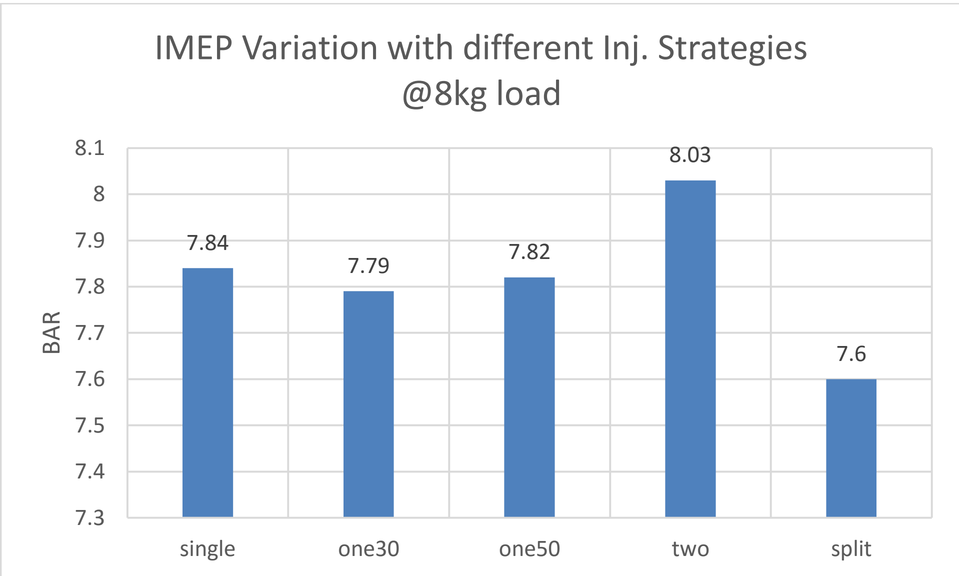


Figure 28: IMEP variation with different injection strategies at 8kg load.

The IMEP values are described in Fig. 27 and 28. The higher IMEP values indicate the better combustion efficiency which is defined as the ratio of the fuel chemical energy and actual heat release during combustion. Therefore, it can be concluded that two pilot injection strategies improve combustion efficiency of low compression ratio engine and this improved combustion efficiency originated from low heat radiation of the flame which might reduce cooling loss through the combustion chamber wall. The IMEP values increased 0.8% and 2.4% with 4kg and 8kg load respectively. Split injection technique has been most disadvantageous in this parameter.

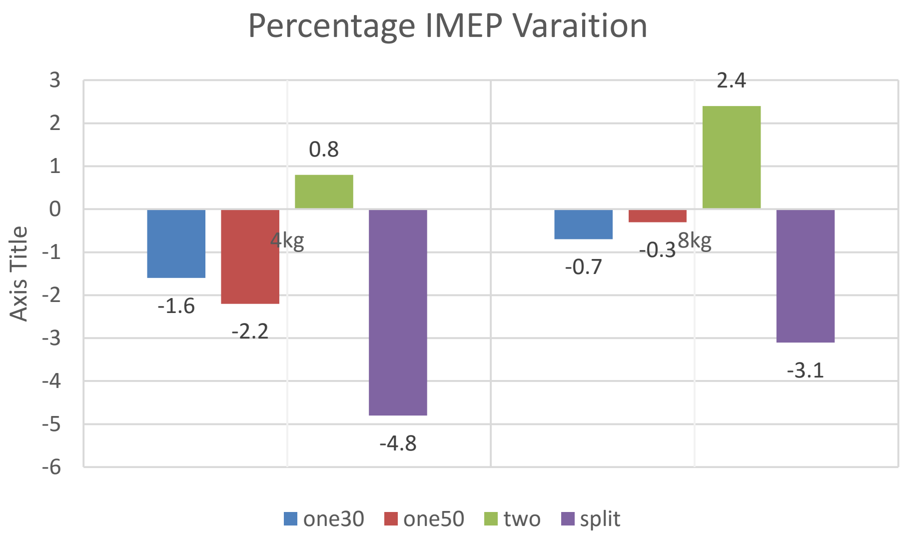


Figure 29: Percentage variation in IMEP.

4.2.3 Availability Analysis

Availability is not a conserved property; availability is destroyed by irreversibility in any process the system undergoes. The change in availability of any system undergoing any process where work, heat, and mass transfer across the system boundary occur can be written as,

$$\Delta A = A_{in} - A_{out} - A_{destroyed}$$

When availability destruction occurs, the potential for the system to do useful mechanical work is permanently decreased. Thus to make a proper evaluation of the processes occurring within an engine system, both energy and availability must be considered concurrently [16].

The mathematical equations[14] used for the calculations of Availability associated with various processes are as following:

- A_{in}= $(1.033 * m_f * LHV)/3600$;
- $A_{transfer\ to\ cooling\ water} = (m_w/3600) * \{(c_{pw} * (T_{wo} - T_{wi}) + (T_0 * \left(c_{pw} * \ln\left({}^{T_{wi}}/_{T_{wo}}\right)\right))\}$;

- $A_{loss\ in\ exhaust\ gases} = Q_{ex} + [(m_{ex}/3600) * T_0 * \{\left(c_{pex} * ln\left({T_0}/{T_{exo}}\right)\right) - (R_{ex} * \ln({P_0}/{P_{exo}})\}];$

- $A_{destroyed} = A_{in} - (A_{shaft} + A_{transfer\ to\ cooling\ water} + A_{loss\ in\ exhaust\ gases});$

- $A_{efficiency} = \left\{1 - {A_{destroyed}}/{A_{in}}\right\} * 100.$

The availability calculated in various processes with the equations gives us an account of how much available energy is lost in those processes and how much we were able to extract from the engine operation. This data provides to us the capability try to minimise the exergy destructions in combustion process. The availability input and variation with different injection strategies under different loading conditions is shown in Fig. 31 and 33.

In the present research engine we don't have the facility to control the fuel injection quantity per cycle, so the fuel injected per cycle is different for different injection strategies. Now, to have a bench mark and to compare the availability associated with processes in different injection strategies the availability input and distribution is converted into the percentage form for better understanding. Figure 32 and 34 represents the availability input and distribution for varying fuel input in percentage under different loading conditions.

The exergetic efficiency gives us the second law quantification of the quality of the combustion process under different injection strategies. Figure 35 shows the exergetic efficiency calculated with the above equations for different injection strategies.

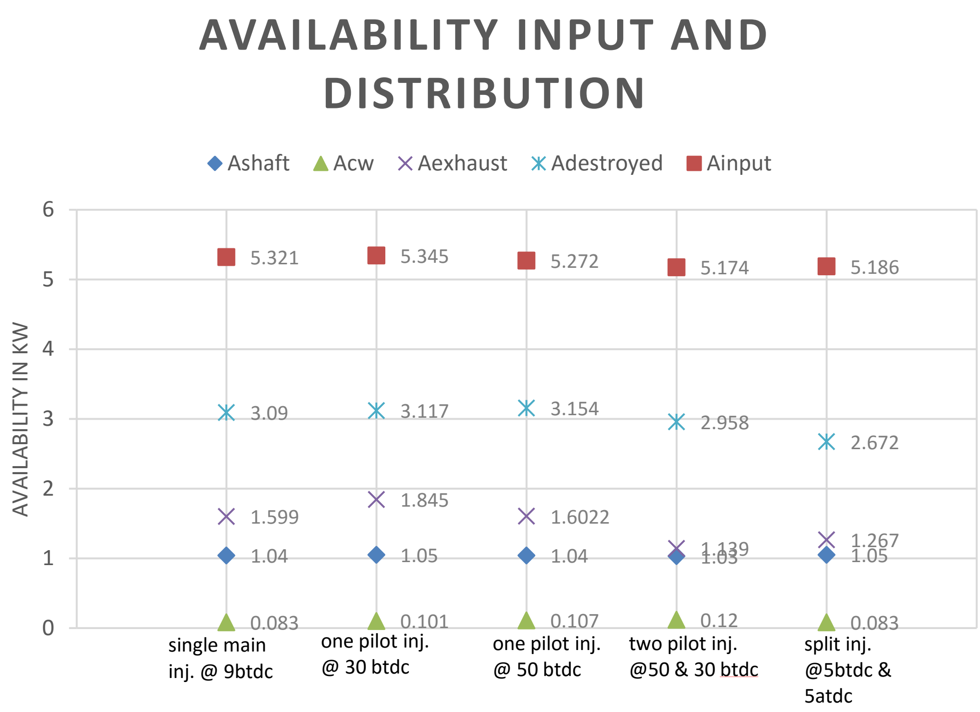


Figure 30: Availability input and distribution under 4kg load

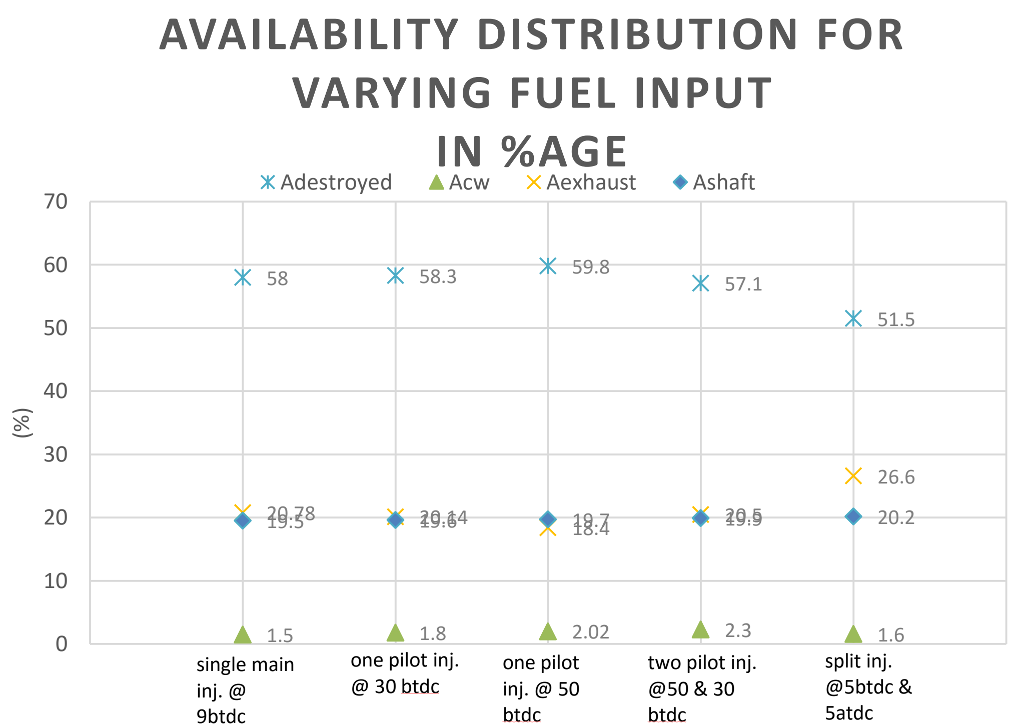


Figure 31: Availability distribution for varying fuel input in %age under 4 kg load

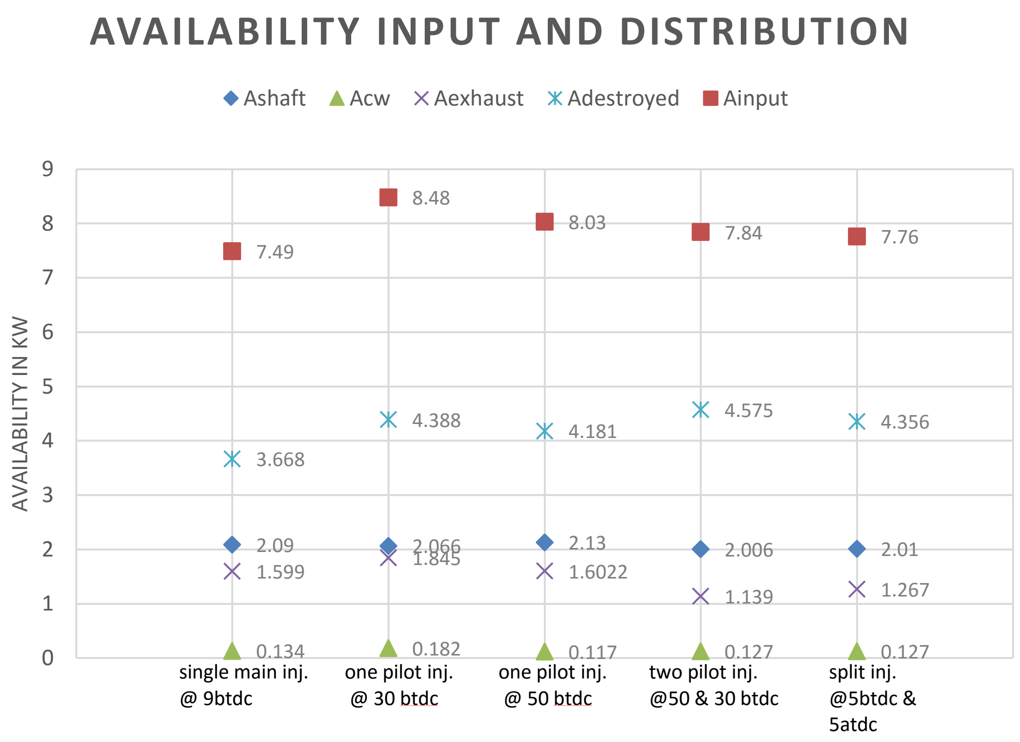


Figure 32: Availability input and distribution under 8 kg load

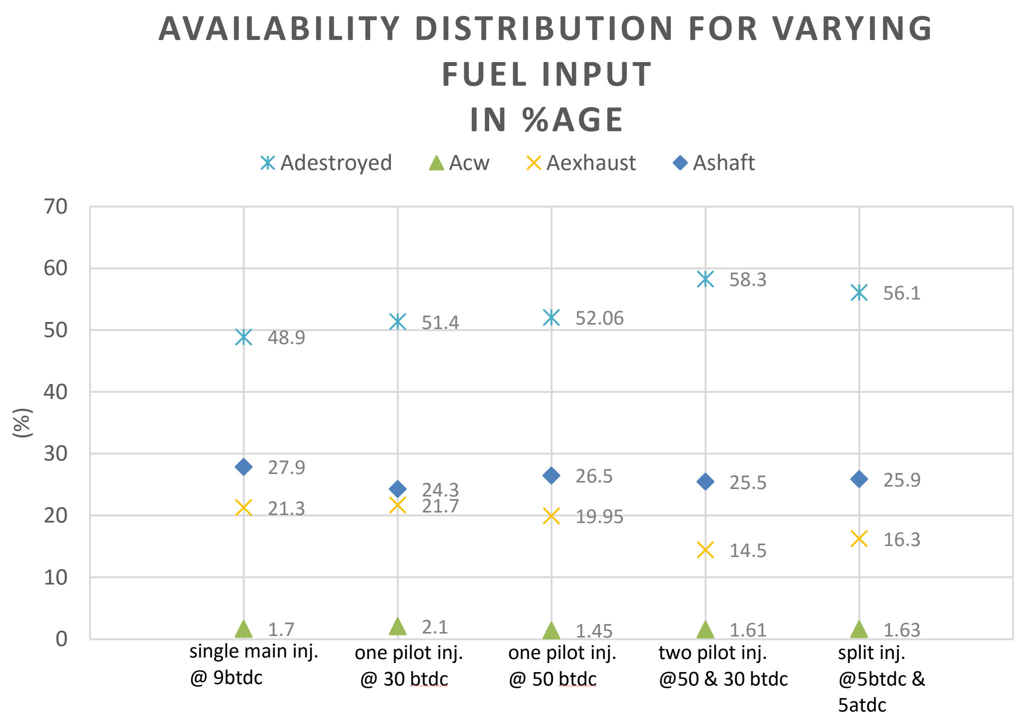


Figure 33: Availability distribution for varying fuel input in %age under 8 kg load

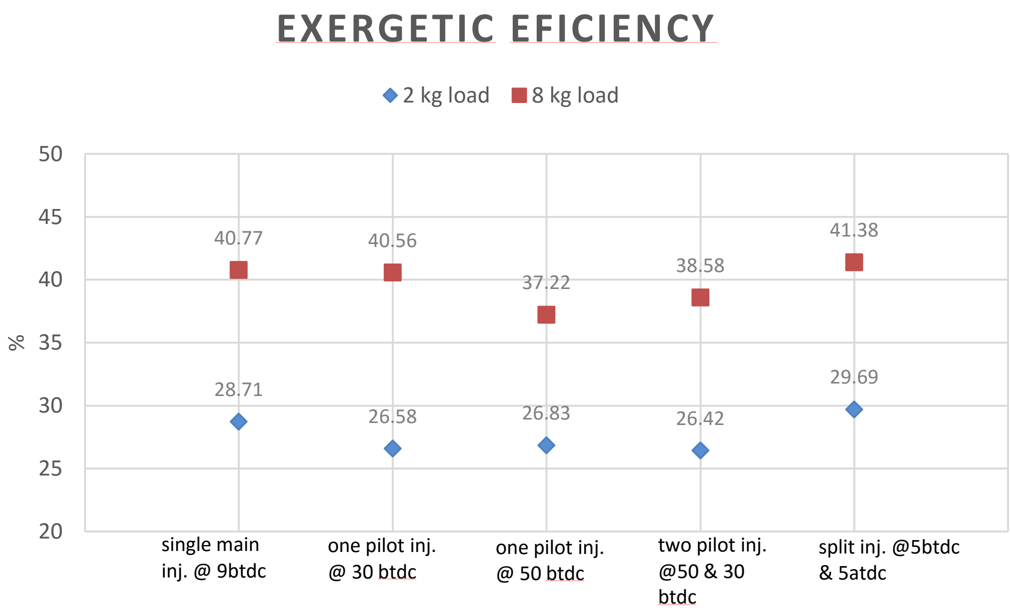


Figure 34: Exergetic Efficiency with different injection strategies

5 Conclusions

In this study, the impact of multiple injection strategies on the capability of exhaust emissions reduction were investigated in a CRDI diesel engine while maintaining a satisfactory engine performance. The conclusions of this work are as follows:

1. Pressure rise rate with two pilot injections is the fastest because of the combustion of two injections with interval prior to the main injection which results in enhanced combustion of main injection and lesser premixed combustion phase.

2. Split injection shows the minimum values for the combustion pressure, HRR. The reason behind is the retardation in the SOI timing to limit the noise produced due to discontinuous combustion.

3. The IMEP with double pilot injection increased 0.8% and 2.4% with medium and high loads. The splitting of early pilot into two pilot injections results in lesser fuel sticking to the cylinder walls and more fuel burning prior to main injection which improves the combustion efficiency.

4. More CO and HC emissions with early pilot injection because most of the fuel gets accumulated in the combustion chamber and some part gets stick to the cylinder walls which increases CO and HC emissions.

5. Minimum NOx levels were obtained by split injection strategies but the engine performance also went down. Split injection resulted in more combustion noise which was limited by retarding the SOI timings.

6. Among the pilot injections, double pilot is the most promising strategy for NOx reduction. Under medium and high loads 22.23% and 6.7% reductions in NOx levels was recorded.

6 Future scope

Combination of other emissions reduction techniques with multiple injection strategies:

1. Exhaust gas Recirculation.
2. Employing alternate fuels such as biodiesels, alcohols etc. and their blends with diesel fuels.

7 References

[1] A. K. Agarwal, “Biofuels (alcohols and biodiesel) applications as fuels for internal combustion engines,” *Prog. Energy Combust. Sci.*, vol. 33, no. 3, pp. 233–271, Jun. 2007.

[2] M. Karabektas and M. Hosoz, “Performance and emission characteristics of a diesel engine using isobutanol–diesel fuel blends,” *Renew. Energy*, vol. 34, no. 6, pp. 1554–1559, Jun. 2009.

[3] J. Yan, D. Lee, S. Chou, U. Desideri, H. Li, B. Choi, Y. K. Kim, G. Jung, C. Lee, X. Jiang, and I. Choi, “International Conference on Applied Energy, ICAE2014Effect of Diesel Fuel Blend with Biobutanol on the Emission of Turbocharged CRDI Diesel Engine,” *Energy Procedia*, vol. 61, pp. 2145–2148, Jan. 2014.

[4] D. C. Rakopoulos, C. D. Rakopoulos, R. G. Papagiannakis, and D. C. Kyritsis, “Combustion heat release analysis of ethanol or n-butanol diesel fuel blends in heavy-duty DI diesel engine,” *Fuel*, vol. 90, no. 5, pp. 1855–1867, May 2011.

[5] D. C. Rakopoulos, C. D. Rakopoulos, E. G. Giakoumis, A. M. Dimaratos, and D. C. Kyritsis, “Effects of butanol–diesel fuel blends on the performance and emissions of a high-speed DI diesel engine,” *Energy Convers. Manag.*, vol. 51, no. 10, pp. 1989–1997, Oct. 2010.

[6] M. Senthil Kumar, A. Ramesh, and B. Nagalingam, “An experimental comparison of methods to use methanol and Jatropha oil in a compression ignition engine,” *Biomass Bioenergy*, vol. 25, no. 3, pp. 309–318, Sep. 2003.

[7] H. Raheman and A. G. Phadatare, “Diesel engine emissions and performance from blends of karanja methyl ester and diesel,” *Biomass Bioenergy*, vol. 27, no. 4, pp. 393–397, Oct. 2004.

[8] E. İleri and G. Koçar, “Effects of antioxidant additives on engine performance and exhaust emissions of a diesel engine fueled with canola oil methyl ester–diesel blend,” *Energy Convers. Manag.*, vol. 76, pp. 145–154, Dec. 2013.

[9] V. Ganesan, *Internal Combustion Engines*. New Delhi: McGraw Hill Education, 2012.

[10] Y. Hotta, M. Inayoshi, and K. Nakakita, “Achieving Lower Exhaust Emissions and Better Performance in an HSDI Diesel Engine with Multiple Injection,” *SAE Tech. Pap. 2005-01-0928*, 2005.

[11] H. K. Suh, “Investigations of multiple injection strategies for the improvement of combustion and exhaust emissions characteristics in a low compression ratio (CR) engine,” *Appl. Energy*, vol. 88, no. 12, pp. 5013–5019, Dec. 2011.

[12] M. Yao, H. Wang, Z. Zheng, and Y. Yue, "Experimental study of n-butanol additive and multi-injection on HD diesel engine performance and emissions," *Fuel*, vol. 89, no. 9, pp. 2191–2201, Sep. 2010.

[13] M. Y. Kim, S. H. Yoon, K. H. Park, and C. S. Lee, "Effect of Multiple Injection Strategies on the Emission Characteristics of Dimethyl Ether (DME)-Fueled Compression Ignition Engine," *Energy Fuels*, vol. 21, no. 5, pp. 2673–2681, Sep. 2007.

[14] B. B. Sahoo, U. K. Saha, N. Sahoo, and P. Prusty, "Analysis of Throttle Opening Variation Impact on a Diesel Engine Performance Using Second Law of Thermodynamics," pp. 703–710, Jan. 2009.

[15] S. Ismail and P. S. Mehta, "Evaluation of the effects of fuel and combustion-related processes on exergetic efficiency," *Fuel*, vol. 90, no. 5, pp. 1818–1825, May 2011.

[16] J. Heywood, *Internal Combustion Engine Fundamentals*. McGraw-Hill Education, 1988.