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Effect of Injection Timing, Combustion Chamber Shapes and Nozzle Geometry on the Diesel Engine Performance

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Abstract
Diesel engines are becoming more important for transport and power generation applications as they have high thermal efficiency and lower HC and CO emissions compared to gasoline engines. Also this can handle wide variety of fuel types; hence they can be predominantly used for transportation and power generation applications. Thus, now it has become imperative to improve their fuel consumption and emission characteristics. In this direction several researchers have carried out attempts to improve the thermal efficiency and emission characteristics of compression ignition (CI) engines without having to sacrifice their fuel consumption. In diesel engines, air-fuel mixing are quite stringent. Hence, this needs to be addressed and beside required fuel-air mixing improvements in CI engines is possible through changes in injection and combustion chamber geometry. In view of this experimental investigations were carried out to study the effect of injection timing, nozzle and combustion chamber geometry on the performance of diesel engine. By the current study it is concluded that advancing the injection timing appropriately and adopting suitable combination of injector nozzle and combustion chamber configuration would greatly enhance the engine performance, besides reducing emission levels to a great extent.

Key Word and Phrases:
Direct Injection Diesel Engines, In-Cylinder Pressure, Heat Release Rate, Nozzle and Combustion Chamber, Emission.

1. Introduction

Diesel engines are widely used as prime movers in transport industry, for power generation and used quiet extensively in agricultural operations due to their high efficiency compared to their counter parts petrol engines and other types of prime movers like gas turbines. Diesel engines are widely employed for the above mentioned applications due to their simple and sturdy mechanical structure, low fuel cost, high reliability and durability, and high power to weight ratio as reported by Lin and Wang [1]. Hence, due to its unique combination of energy efficiency, power, reliability, and durability, diesel technology plays a vital role in automotive and power generation applications. The field of application is increasing steadily. The technological advancement in century, the awareness of the pollution effects with ever stringent emission norms has made the engine research highly professional, challenging, competitive and rewarding as well. The diesel engine as compared to the petrol-powered engine of the equal volume of combustion chamber has advantage of higher torque. The main reasons for the emission reductions since last decade are fuel efficiency improvements, mainly in diesel-fueled vehicles, and a shift in fleet composition from petrol to diesel passenger cars. For this reason, advanced clean diesel technology must remain a viable option for light-duty vehicles. The increased cost and scarcity of hydrocarbon fuels have made the study of internal combustion engines more and more attractive and the complexities in the process study made the work all the more challenging. The conventional diesel combustion structure combines premixed flame combustion and diffusion flame combustion. It has been recognized that this combustion structure is one of the major cause of NOx and soot emissions.Diesel engine, fuel and emissions control technology manufacturers are working with

EPA, state and regional governments and environmental organizations in a collaborative effort to reduce diesel-related emissions. For diesel engines a general strategy of lowering soot and NOx emission is more complicated. This is due to the interrelation between the heterogeneous mixture formation and self-ignition of the diesel combustion process. On the one hand, long ignition delay times lead to a more premixed combustion with low soot emission due to lean combustion. In addition, this process amplifies NOx formation because of increasing combustion temperature. This soot-NOx trade-off is one of the major problems in diesel combustion development. Diesel vehicles have been regarded as one of the major air pollution sources particularly in metropolitan cities [2]. Particulate matter (PM) or smoke and NOx emission are the most significant pollutants emitted from diesel engines [3].

Injection Timing and Injection Pressure

In the diesel engine, combustion and emissions characteristics are greatly influenced by the quality of atomization and in particular by the fuel-air mixture present in the combustion chamber. Injection pressure and injection timing are the most important parameters affecting the diesel engine performance and the fuel injection pressures of diesel engines have continued to increase over recent years with implementation of CRDI and unit injection systems.

Several researchers have conducted engine tests on compression ignition (CI) engine with different fuels at different injection pressures and injection timings [4]. Advancing or retarding the injection timings change the position of the piston and cylinder pressure and temperature inside the combustion chamber. Retarded injection timings showed significant reduction in diesel NOx and biodiesel NOx [5]-[6]. Cylinder pressures and temperatures gradually decrease when injection timings are retarded [7]. The effect of fuel injection timing with waste cooking oil as an alternative to diesel in direct injection diesel engines has been investigated and that an advanced injection timing of 40^{o}bTDC has shown better efficiency, reduced CO and higher NOx emissions has been reported [8]. It has been reported that retarded injection timing and increased injection pressure has favoured improved performance with Honge biodiesel fuelled diesel engine [9].

Similarly various investigators have also performed experiments on CI engine with different fuel combinations at different injection pressures. Better performance, higher peak cylinder pressure and temperature were observed at increased injection pressure [7]-[11]. Effect of injection timing, injection pressure on the cotton seed oil methyl ester fuelled diesel engine has been reported in the literature. Retarding injection timing and increasing injector opening pressure upto 230 bar has been found to improve engine performance [12]

Combustion Chamber Shapes

Combustion chamber of an engine plays a major role during the combustion of wide variety of fuels used. In this context, many researchers performed both experimental and simulation studies on the use of various combustion chambers [13]-[14]. In re-entrant combustion chamber intensification of swirl and turbulence were reported to be higher when compared to cylindrical chambers which lead to more efficient combustion causing higher NOx emissions and lesser soot and HC emissions [15]. Montajir et al. [16] studied the effect of combustion chamber geometry on fuel spray behavior and found that a re-entrant type combustion chamber with round lip and round bottom corners provides better air and fuel distribution than a simple cylindrical combustion chamber. Experimental study to optimize the combination of injection timing and combustion chamber geometry to achieve higher performance and lower emissions from biodiesel fueled diesel engine has been reported. Toroidal re-entrant combustion chamber and retarded injection timing has been found to improve brake thermal efficiency and reduced brake specific fuel consumption [17]. Improvement in air entrainment with increased swirl and injection pressure were reported [18] –[19]. Prasad et al. [20] studied in-cylinder air motion in a number of combustion chamber geometries and identified a geometry which produced the highest in-cylinder swirl and turbulence kinetic energy around the compression top dead centre (TDC). Three dimensional CFD simulations involving flow and combustion chemistry were used to study effect of swirl induced by re-entrant piston bowl geometries on pollutant emissions of a single cylinder diesel engine fitted with a hemispherical piston bowl and an injector with finite sac volume. The optimal geometry of the re-entrant piston bowl geometry was confirmed by the detailed combustion simulations and emission predictions used. Optimum combustion chamber geometry of the engine showed better performance and emission levels. Suitable combustion geometry of bowl shape helps to increase squish area and proper mixing of gaseous fuel with air [13], [21]. Designing the combustion chamber with narrow and deep and with a shallow reentrance had a low protuberance on the cylinder axis and the spray oriented towards the bowl entrance reduced the NOx emission levels to the maximum extent [14], [17], [22]. Influence of combustion chamber geometry on pongamia oil methyl ester and its blend (B20) fuelled diesel engine were investigated [14] in which toroidal re-entrant and shallow depth re-entrant combustion chambers were used. Toroidal reentrant combustion chamber resulted into higher brake thermal efficiency, higher NOx and reduced emissions of particulates, CO, UBHC. Lower ignition delay, higher peak pressure with B20 were also obtained when compared to baseline hemispherical and shallow depth reentrant combustion chambers [14].

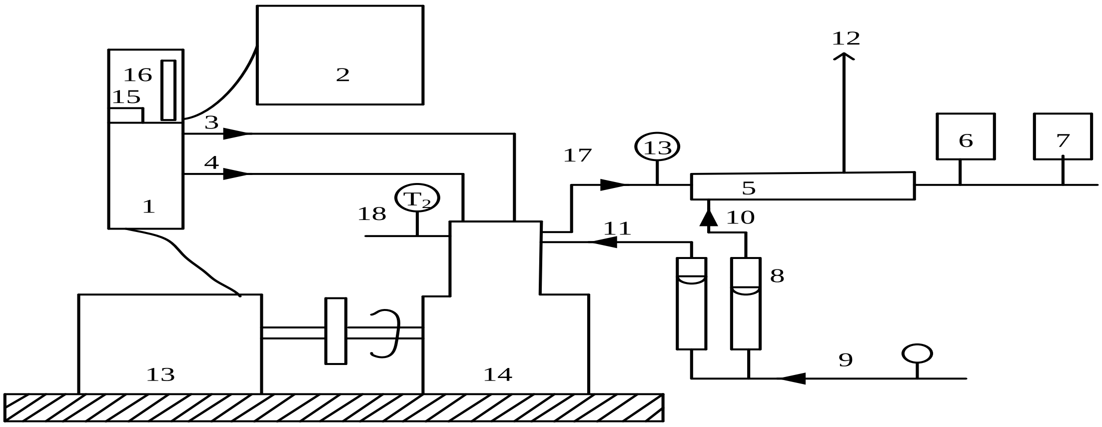
Injection Strategies: Nozzle Geometry

The behaviour of fuel once it is injected in to the combustion chamber and its subsequent interaction with air is important. This is in turn helps to achieve homogeneous air-fuel mixture available for combustion. It is well known that nozzle geometry and cavitations strongly affect evaporation and atomization processes of fuel. Suitable changes in the in-cylinder flow field results in varying combustion quality. The performance and emission characteristics of compression ignition engines are largely governed by fuel atomization and spray processes which in turn are strongly influenced by the flow dynamics inside injector nozzle. Modern diesel engines use micro-orifices having various orifice designs and affect engine performance to a great extent. Effects of dynamic factors on injector flow, spray combustion and emissions have been investigated by various researchers [23]-[24]. Experimental studies involving the effects of nozzle orifice geometry on global injection and spray behaviour has been reported [25]-[27]. Several biofuels have been used as alternative to diesel for diesel engine applications [28].

From the literature survey it follows that very limited work has been done to investigate the combined effect of injection parameters, combustion chamber shapes and injector nozzle geometry on the performance, combustion and emission characteristics of diesel fuelled engines. In this context, experimental investigations were carried out on a single cylinder four stroke direct injection diesel engine operated on fossil diesel with different injection parameters, combustion chamber shapes and injectors suitably adopted for this work.

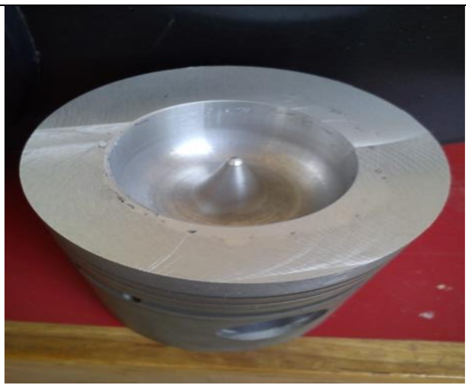
2. Experimental setup

Experiments were conducted on a Kirloskar TV1 type, four stroke, single cylinder, water-cooled diesel engine test rig fuelled diesel. Figure 1 shows the line diagram of the test rig used. Eddy current dynamometer was used for loading the engine. The fuel flow rate was measured on the volumetric basis using a burette and stopwatch. The engine was operated at a rated constant speed of 1500 rev/min. The emission characteristics were measured by using HARTRIDGE smoke meter and five gas analyzer during the steady state operation. Different combustion chamber shapes of cylindrical (CCC), trapezoidal (TrCC), shallow depth (SDCC) and toroidal (TCC) were used apart from the hemispherical shape provided in the existing engine. Figures 2 (a), (b), (c), (d) and (e) shows the different combustion chamber shapes used in the study. To study the effect of number of nozzle holes, different injectors with 3, 4, 5 holes each of 0.3 mm were selected. Further to study the effect of orifice size a 5 hole injector with 0.2, 0.25 apart from 0.3 mm were also selected. Fig. 3 shows the various injectors used in the study. Finally the results obtained with diesel operation were analyzed. Properties of the HS diesel used are shown in Table 1. The specification of the compression ignition (CI) engine is given in Table 2.



1- Control Panel, 2 - Computer system, 3 - Diesel flow line, 14 - Air flow line, 5 – Calorimeter, 6 - Exhaust gas analyzer, 7 - Smoke meter, 8 - Rota meter, 9, 11- Inlet water temperature, 10 - Calorimeter inlet water temperature,12 - Calorimeter outlet water temperature, 13 – Dynamometer, 14 - CI Engine, 15 - Speed measurement,16 - Burette for fuel measurement, 17 - Exhaust gas outlet, 18 - Outlet water temperature, T1- Inlet water temperature, T2 - Outlet water temperature, T3 - Exhaust gas temperature.

Fig. 1 Experimental set up

	
(a) Hemispherical	(b) Cylindrical
	
(c) Toroidal	(d) Trapezoidal

(e) Shallow depth

Fig. 2 Combustion chamber shapes

3 hole	4 hole	5 hole

Fig. 3 Injectors with different number of nozzle holes

Table 1 Properties of fuels tested

Sl. No.	**Properties**	**Diesel**
1	Chemical Formula	$C_{13}H_{24}$
2	Density (kg/m^3)	840
3	Calorific value (kJ/kg)	43,000
4	Viscosity at 40^{o}C (cSt)	2-5
5	Flashpoint (^{o}C)	75
6	Cetane Number	45-55
7	Carbon Residue (%)	0.1
8	Cloud point	-2
9	Pour point	-5
10	Carbon residue	0.13
11	Molecular weight	181
12	Auto ignition temperature (^{o}C)	260
13	Ash content % by mass	0.57
14	Oxidation stability	High
15	Sulphur Content	High

Table 2 Specifications of the engine

Sl No	Parameters	Specification
1	Type of engine	Kirloskar make Single cylinder four stroke direct injection diesel engine
2	Nozzle opening pressure	200 to 205 bar
3	Rated power	5.2 KW (7 HP) @1500 RPM
4	Cylinder diameter (Bore)	87.5 mm
5	Stroke length	110 mm
6	Compression ratio	17.5 : 1

3. Results and Discussions

In this section effects of injection timing, combustion chamber configurations, and nozzle geometry on the performance of diesel engine are presented in following sections.

3.1 Effect of Injection Timing on the Performance of Diesel Fuelled Engine: (Optimization of Injection Timing for Diesel Fuel)

Studies on the diesel engine performance were conducted at three injection timings of 19^o, 23^o and 27^o bTDC. Compression ratio of 17.5, injector opening pressure of 205 bar (20.5 MPa) was maintained. An injector of three holes each having 0.3 mm diameter orifice was selected for the study.

In the first phase of this work, studies on basic performance, emission and combustion characteristics of a single cylinder four stroke compression ignition engine fuelled with diesel was carried out using three different injection timings. At the rated speed of 1500 rev/min, variable load tests were conducted at three injection timings selected and at a fixed injection pressure of 205 bar. For each load, air flow rate, fuel flow rate, exhaust gas temperature, HC, CO, smoke and NOx emissions were recorded. Based on the results, optimum injection timing was determined for the diesel fuel tested.

3.1.1 Brake Thermal Efficiency

The effect of injection timing on brake thermal efficiency for single fuel operation with diesel at three injection timings is shown in Fig. 4. Highest brake thermal efficiency of 25.5% is obtained for 80% load with diesel at a static injection timing of 23^o BTDC. The brake thermal efficiency at 19^o and 27^o BTDC is found to be 24.62 and 24.82% respectively for diesel. However by retarding or advancing the injection timing by 4^oCA there was no noticeable improvement in brake thermal efficiency. The optimum injection timing was found to be 23^oBTDC, which also matched with the manufacturer's specification.

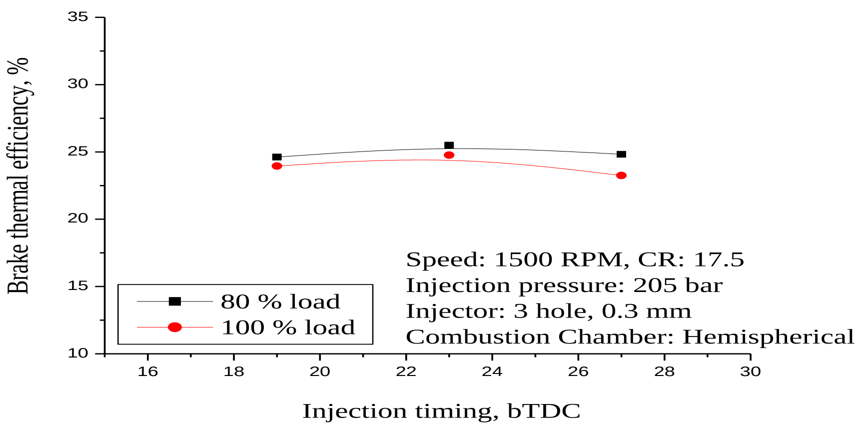


Fig. 4 Effect of injection timing on brake thermal efficiency

3.1.2 Emission Characteristics

Smoke Opacity

Formation of smoke is basically a process of conversion of molecules of hydrocarbon fuels into particles of soot. The effect of injection timing on smoke emission for diesel operation is shown in Fig. 5. The smoke emission with diesel generally increases as the injection timing is retarded. The reasons for this could be incomplete combustion prevailing inside combustion chamber associated with incorrect air-fuel ratio and improper mixing of the two. It is seen that with diesel fuel the smoke level falls when the injection timing is advanced to 23^o BTDC from 19^o BTDC. When the injection is advanced to 23^o BTDC from 19^o BTDC, fuel injection occurs at lower temperature and pressure in the cylinder. This results in an increase in the ignition delay and hence a significant portion of the injected fuel burns in the premixed mode. Hence, this results in lower smoke. However, with the further increase in injection timing to 27^o BTDC the smoke level is observed to increase due to fall in brake thermal efficiency, which leads to increased fuel input at a given power output.

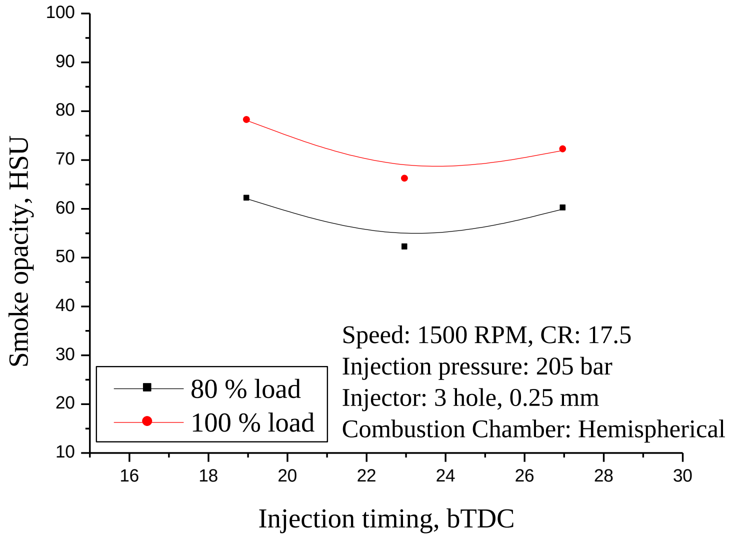


Fig. 5 Effect of injection timing on smoke

HC and CO Emissions

Figure 6 and 7 shows effect of injection timing on HC and CO emissions for diesel engine when using fossil diesel fuel. Hydrocarbon emissions in diesel engines are caused due to lean mixture during delay period and under mixing of fuel leaving fuel injector nozzle at lower velocity. The general trend of varying HC and CO emissions were observed for diesel at three injection timings considered. This may be attributed to decreased combustion efficiency available with these timings. Lowest HC levels are found at injection timing of 23^o BTDC.

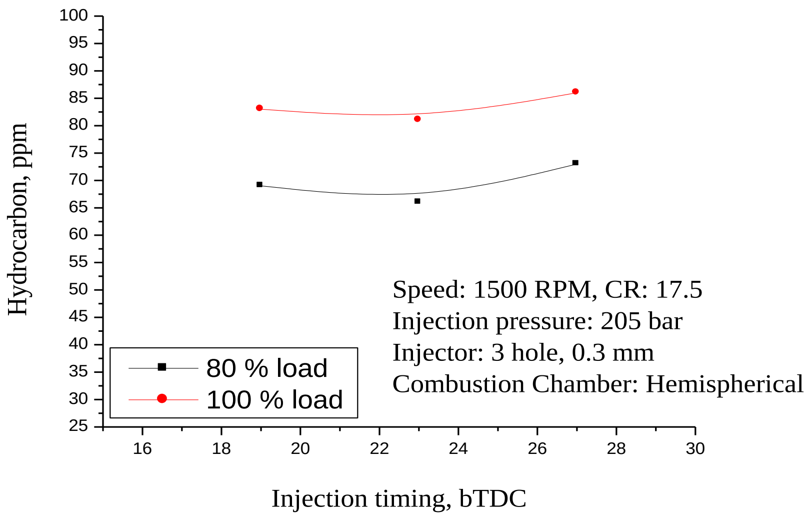


Fig. 6 Effect of injection timing on HC emissions

Carbon monoxide is a toxic by-product and is a clear indication of incomplete combustion of the pre-mixed mixture. Similar trends were observed with CO emissions as well compared to HC emissions at injection timing of 23°BTDC. Lowest CO levels were found at the optimum injection timing of 23°BTDC.

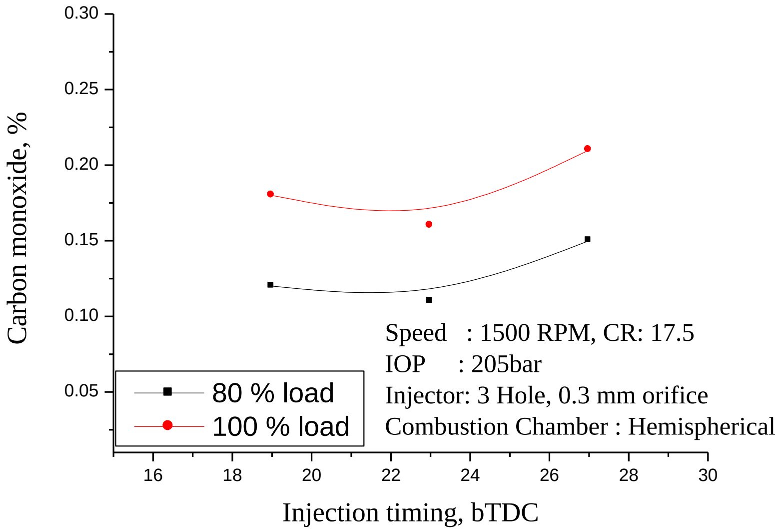


Fig. 7 Effect of injection timing on CO emissions

NOx Emissions

The effect of injection timing on emission of nitrogen oxides for diesel is shown in Fig. 8. In general retarded injection results in substantial reduction in nitrogen oxide emissions. As the injection timing is retarded, the combustion process gets retarded. Nitrogen oxide concentration levels are lower as peak temperature is lower. Nitrogen oxide levels are higher at the injection timings of 23° and 27° BTDC as they lead to a sharp premixed heat release due to higher ignition delay. From these results the best injection timing was taken as 23° BTDC.

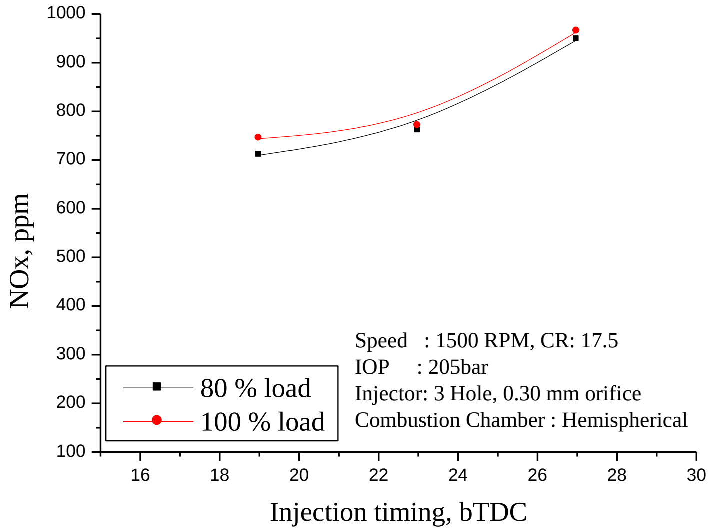


Fig. 8 Effect of injection timing on NOx emissions

3.2 Effect of Combustion Chamber Shapes on the Performance of Diesel Fuelled Engine: (Optimization of Combustion Chamber Shapes for Diesel Fuel)

In the second phase of this work, studies on basic performance, emission and combustion characteristics of a single cylinder four stroke compression ignition engine fuelled with diesel was carried out using four different combustion chamber shapes. At the rated speed of 1500 rev/min, variable load tests were conducted at injection timings of 23^{o}BTDC and at fixed injection pressure of 205 bar. For each load, air flow rate, fuel flow rate, exhaust gas temperature, HC, CO, CO_2, smoke and nitric oxide emissions were recorded. Based on the results optimum combustion chamber shapes was selected for the diesel engine.

3.2.1 Brake Thermal Efficiency:

Figure 9 shows the variation of brake thermal efficiency (BTE) with different combustion chamber configurations adopted. BTE for diesel was varying with different combustion chambers over the entire load range. Different mixture formation occurring in the individual combustion chamber shapes leads to varying combustion behaviour and hence varying BTE is obtained. TCC resulted in better performance compared to other combustion chambers. It may be due to the fact that, the TCC prevents the flame from spreading over to the squish region resulting in better mixture formation of biodiesel-air combinations, as a result of better air motion and lowers exhaust soot by increasing swirl and tumble.

Based on the results, it is observed that the TCC has an ability to direct the flow field inside the sub volume at all engine loads and therefore substantial differences in the mixing process may not be present.

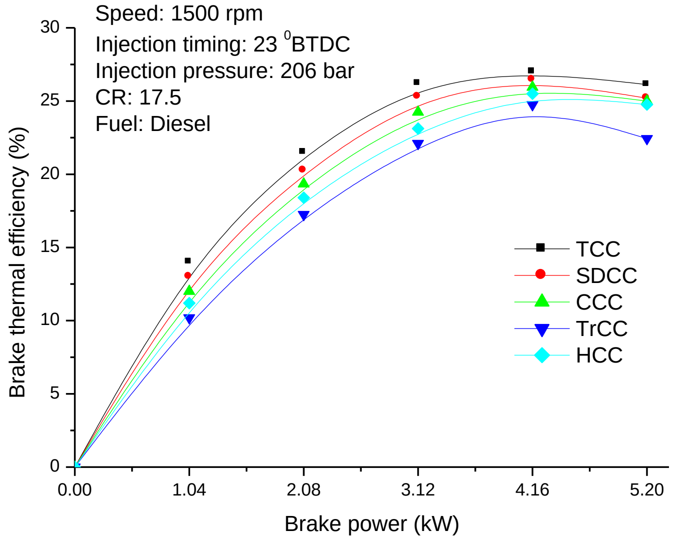


Fig. 9 Variation of BTE with BP for different combustion chamber shapes

3.2.2 Emission Characteristics

Smoke Opacity

Figure 10 shows variation of smoke opacity with brake power for different combustion chamber configurations adopted. Diesel being common fuel used type of combustion chamber configuration affects the smoke opacity over the entire load range. This may be attributed to improper fuel-air mixing occurring inside the individual combustion chamber shapes adopted. However, TCC gives lower smoke emission levels compared to other combustion chambers. It may be due to the fact that, the air-fuel mixing prevailing inside combustion chamber and higher turbulence resulted in better combustion and oxidation of the soot particles which further leads to reduction in the smoke emission levels.

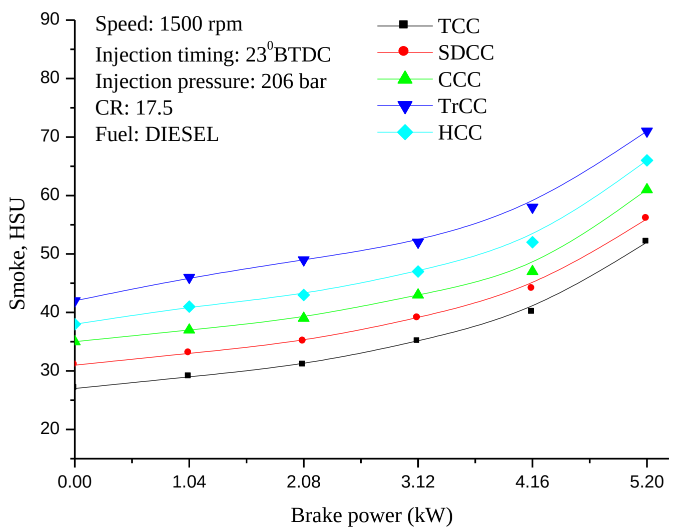


Fig. 10 Variation of Smoke opacity with BP

HC and CO Emissions

Figure 11 and 12 shows the variation of hydrocarbon (HC) and carbon monoxide (CO) emission levels for diesel. Both HC and CO emission levels were observed with the configurations of combustion chambers used. Incomplete combustion and associated lower BTE with respective configuration is responsible for this observed trend. However, TCC resulted in lower HC and CO emission levels compared to other combustion chamber shapes. Marginally comparable performance was observed with SDCC as well. It could be due to higher turbulence and comparatively higher temperature prevailing inside the combustion chamber that resulted into minimum heat losses and better oxidation of HC and CO and hence reduced emission levels. However, other combustion chambers may marginally contribute to the proper mixing of fuel combinations.

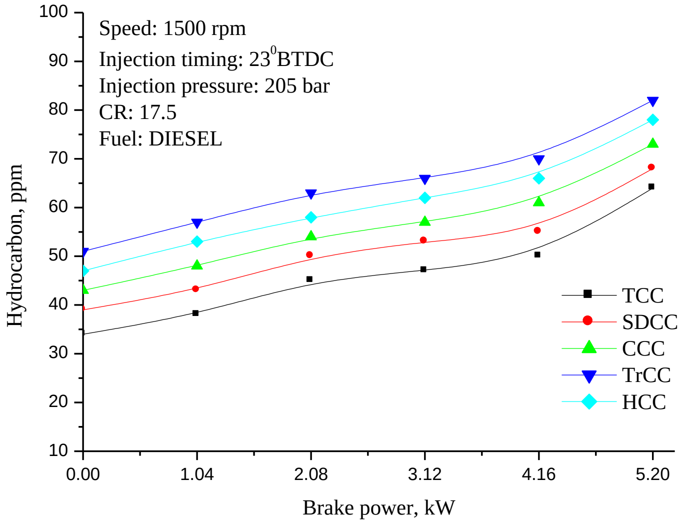


Fig. 11 Variation of HC with BP

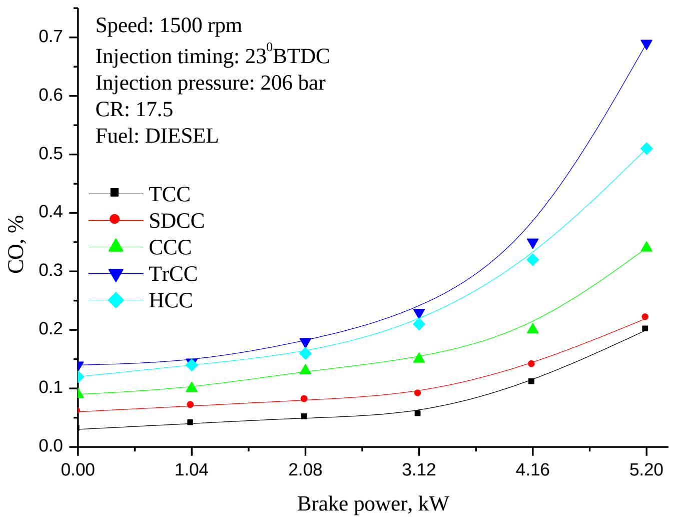


Fig. 12 Variation of CO with BP

NOx Emissions

Higher NOx emission levels were with diesel for the configurations adopted over the entire load range (Fig. 13). Higher heat release rates during premixed combustion phase observed with TCC and SDCC led to higher NO_x formations compared to other combustion chamber shapes used. This could be due to slightly better combustion prevailing in combustion chamber due to more homogeneous mixing and larger part of combustion occurring just before top dead centre.

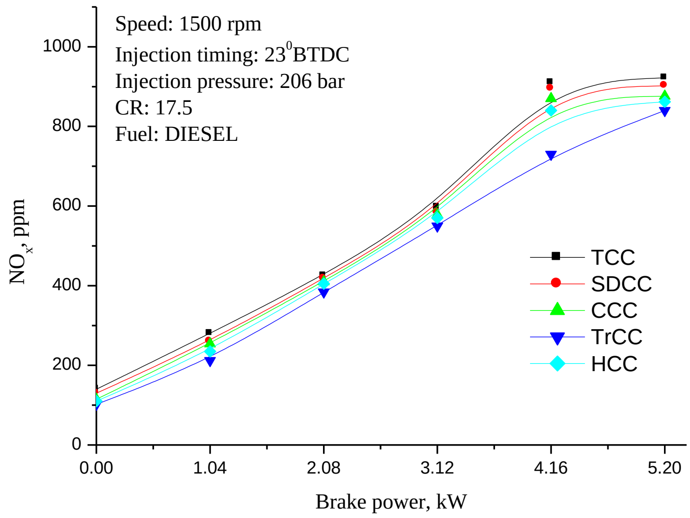


Fig. 13 Variation of NOx with BP

3.3 Effect of Nozzle Geometry on the Performance of Diesel Fuelled Engines: (Optimization of Nozzle Holes and Nozzle Size for Diesel Fuel)

In the third phase of this work, studies on basic performance, emission and combustion characteristics of a single cylinder four stroke compression ignition engine fuelled with diesel was carried out using injectors with different number of holes and hole sizes. Nozzle geometry with 3, 4, and 5 holes each having 0.3 mm orifice was selected for the study. At the rated speed of 1500 rev/min, variable load tests were conducted at injection timing 23°BTDC and at fixed injection pressure of 205 bar. For each load, air flow rate, fuel flow rate, exhaust gas temperature, HC, CO, CO_2, smoke and nitric oxide emissions were recorded. Based on the results optimum injector nozzle geometry was selected for the diesel engine.

3.3.1 Effect of nozzle geometry on Brake Thermal Efficiency:

The effect of brake power on brake thermal efficiency with different nozzle geometry is shown in Fig. 14. At fixed IOP the highest brake thermal efficiency occurred for a nozzle geometry with 5 hole having an orifice diameter of 0.3 mm. This is because atomization, spray characteristics and mixing with air were better, resulting in improved combustion. The BTE is found to be 28. 5% at 80% load and its maximum value obtained with 5-hole nozzle at an IOP of 205 bar. The BTE reported for 3-hole and 4-hole nozzles were 24.80% and 27.56% at 205 bar respectively. Increased number of holes has not much effect on ignition delay, but the fuel-air mixing rate increases and hence the observed results favoured the 5 hole injector.

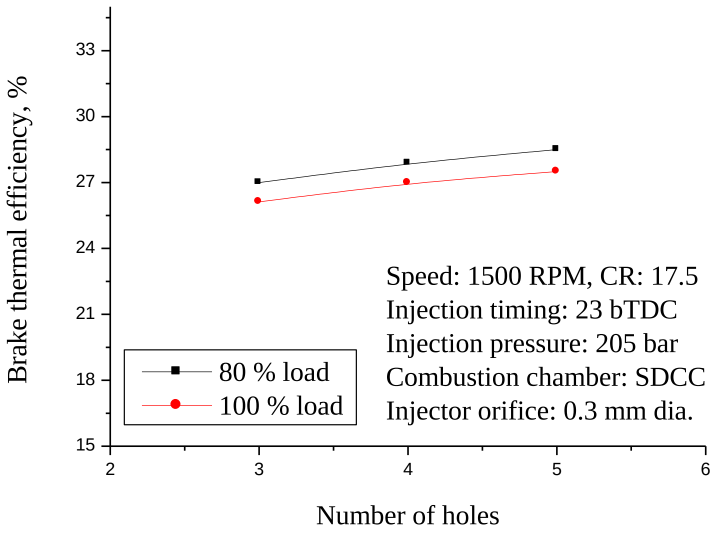


Fig. 14 Effect of nozzle holes on BTE

3.3.2 Emission Characteristics

Smoke Opacity

Figure 15 shows variation of smoke opacity with brake power for different injector nozzle holes adopted. Injector nozzle orifice being constant (0.3 mm) increased number of holes affected the smoke opacity levels over the entire load range. 5 hole injector provided better air-fuel mixture combinations and hence reduced smoke was observed. Improper fuel-air mixing occurring with other injector holes (3 and 4) resulted in poor combustion which increased smoke levels.

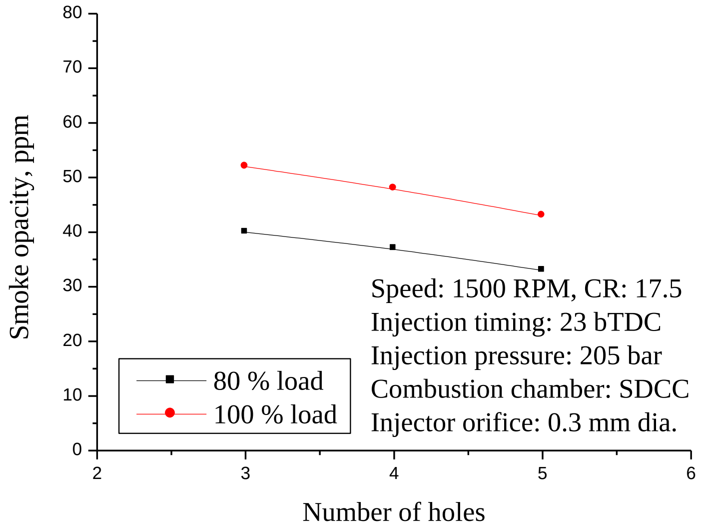


Fig. 15 Effect of nozzle holes on Smoke opacity

HC Emisssion

Figures 16 show the effect of brake power on HC emission using different nozzle geometry at fixed IOP. A significant drop in HC emission is observed with 5-hole nozzle geometry because of better combustion. Enhanced atomization will also lead to a lower ignition delay. An improvement in the spray, will lead to a lower physical delay. HC emissions are found to be lower for 5-hole compared to 3 and 4-hole nozzle geometry respectively. Unburnt hydrocarbons were lesser for 5-hole nozzle and are due to improved atomization and proper combustion of diesel.

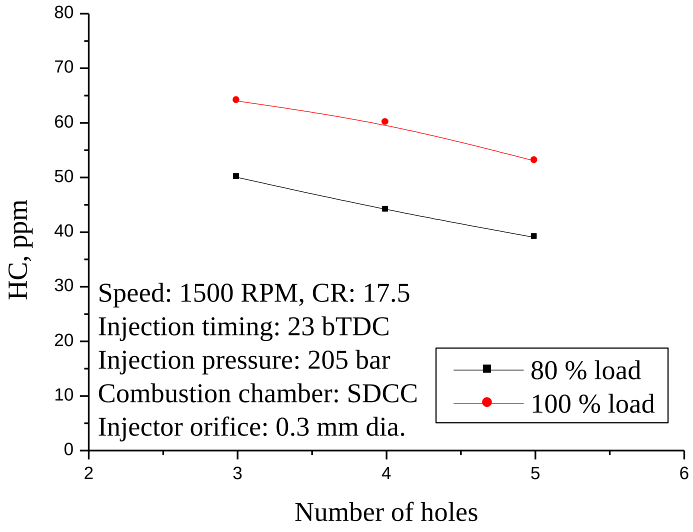


Fig. 16 Effect of nozzle holes on HC Emission

CO Emisssion

Fig. 17 shows effect of brake power on CO emission. Observed trends for CO emissions were similar to HC emissions, with lower CO emissions occurring with 5 hole injector compared to 3 and 4 hole injectors.

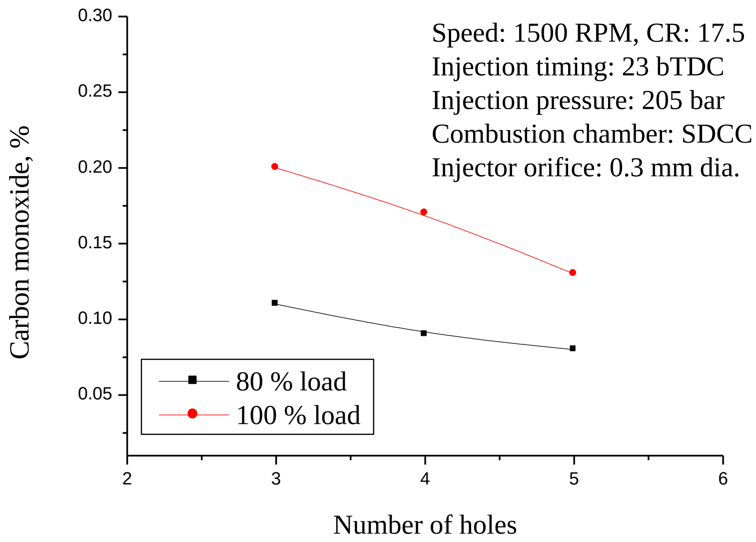


Fig. 17 Effect of nozzle holes on CO Emission

NO_x Emission

NOx emissions increases with the increase in the number of injector holes due to faster combustion and higher temperatures reached in the cycle as shown in Fig. 18.

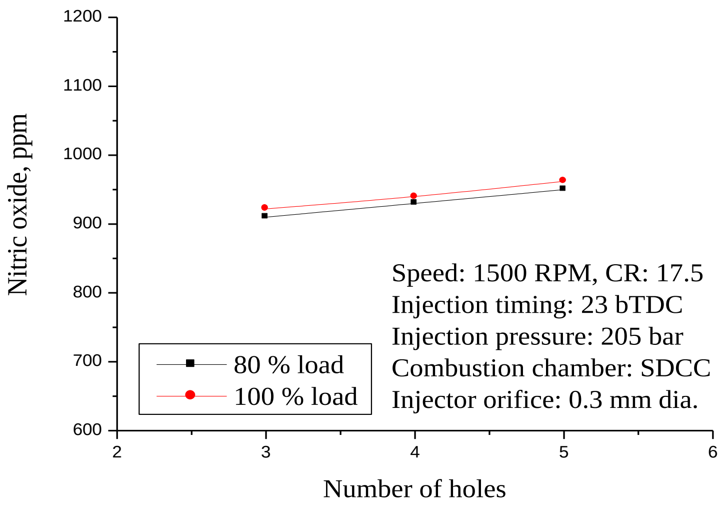


Fig. 18 Effect of nozzle holes on NO_x Emission

3.4 Effect of Size of Injector Nozzle Holes on the Performance of Diesel Fuelled Engines: (Optimization of Nozzle Holes and Nozzle Size for Diesel Fuel)

This section deals with the performance of diesel engine using a 5-hole injector with varying nozzle hole sizes. In order to study the effect of nozzle orifice on the engine performance the orifice diameter was varied from 0.2-0.3 mm in a 5-hole injector.

3.4.1 Brake Thermal Efficiency

Fig. 19 shows variation of BTE for the diesel injected with a 5-hole injector with its orifice size decreased from 0.3 to 0.2 mm. Decreasing the size of holes from 0.3 to 0.2 mm, ensured better mixing of air and fuel inside the combustion chamber and further leads to better combustion and hence increased BTE is found.

However with less than 0.2 mm the above effect is nullified as fuel droplets move faster than air associated with poor mixing further resulting in inferior engine performance.

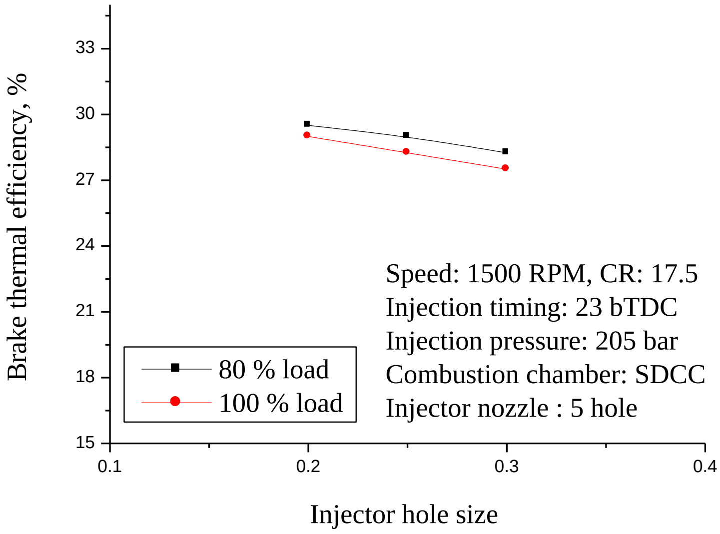


Fig. 19 Effect of injector nozzle orifice size on BTE

3.4.2 Emission Characteristics

Smoke Opacity

Fig. 20 explains the effect of injector nozzle orifice size on the variation of smoke opacity behaviour. Decreased hole size ensures improved air fuel mixing which results in enhanced combustion with improved BTE. This results in reduced smoke opacity levels.

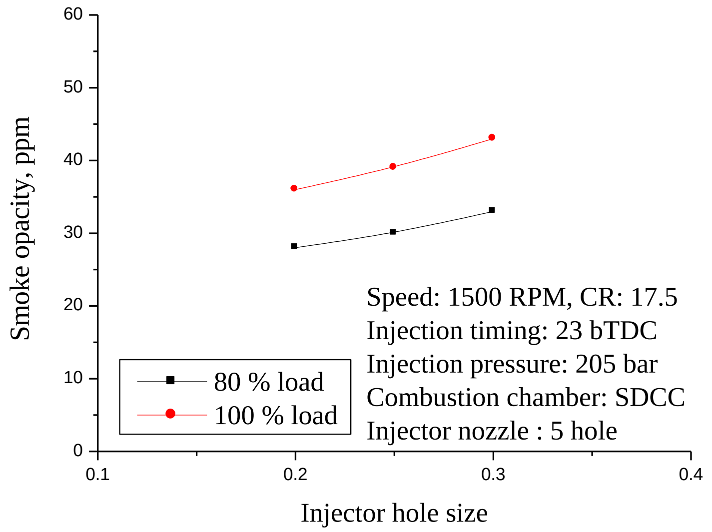


Fig. 20 Effect of injector nozzle orifice size on smoke opacity

HC and CO Emissions

Fig. 21 and 22 shows the effect of injector nozzle orifice size on the variation of HC and CO emissions with fixed 5-hole injector having varying orifice size from 0.3 to 0.2 mm. It may be noted that higher HC and CO from exhaust are the direct result of incomplete combustion. HC and CO emissions were found to be lower for decreasing injector hole sizes as the wall impingement with diesel is less compared to that with larger hole size. However higher hole size injectors leads to deposition of fuel on the combustion chamber walls. Hence higher HC and CO emissions were found to be more with 0.3 mm hole injector.

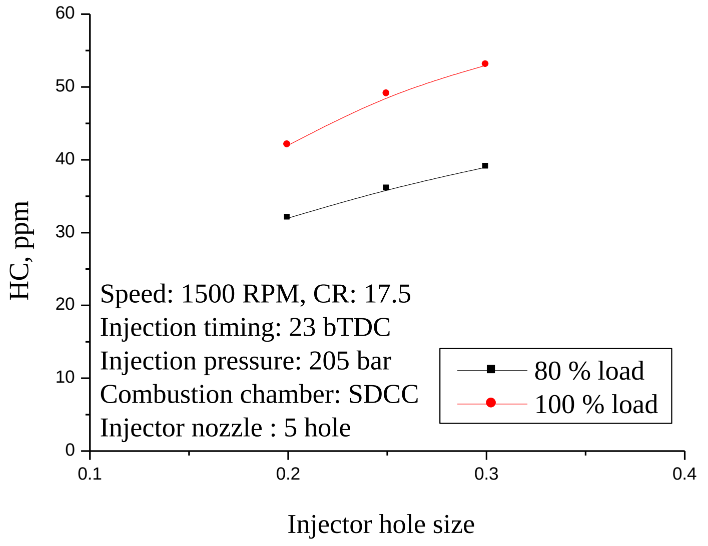


Fig. 21 Effect of injector nozzle orifice size on HC emission

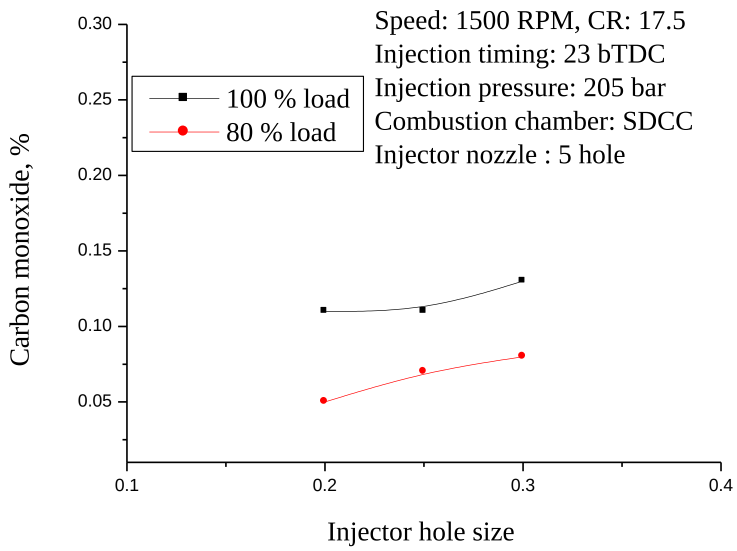


Fig. 22 Effect of injector nozzle orifice size on CO emission

NO_x Emission

Fig. 23 shows the variation of NO_x with 0.2, 0.25 and 0.3mm size of a 4 hole injector for diesel. The variations in NO_x follow changes in adiabatic flame temperature. The reason for increased NO_x with increased size of holes could be due to better combustion prevailing inside the engine cylinder and more heat released during premixed combustion. These effects also vary with spray pattern of liquid fuels, suggesting that reaction zone stoichiometry and post combustion mixing are also influenced by fuel composition. Reducing the injector orifice size further lowers NO_x emission with a 5 hole injector.

Figure 23 is a bar chart showing variation of NO_x with 0.2mm, 0.25mm and 0.3mm hole size injectors for injected diesels. The variations in NO_x followed due to changes in adiabatic flame temperature inside the engine cylinder. The reason for decreased NO_x with increased size of hole could be due to submissive combustion prevailing inside the engine cylinder and comparatively less heat released during premixed combustion in turn it leads to low exhaust gas temperature. These effects also vary with spray pattern of liquid fuels, reaction zone stoichiometry, post combustion mixing and fuel composition also influenced by NO_x formation. The highest NO_x emission found for diesel of 963 ppm with 0.3 mm orifice and this may be due to increased adiabatic flame temperature associated with its combustion.

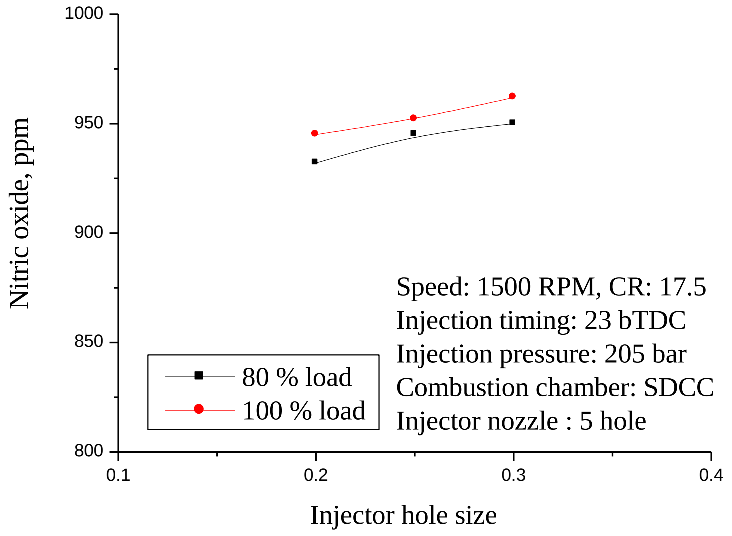


Fig. 23 Effect of injector nozzle orifice size on NOx emission

Combustion Parameters

In this section effect of injector nozzle geometry viz nozzle orifice size on combustion parameters like in-cylinder pressure variation and heat release rate are presented in the form of graphs. The detailed study with remarks is mentioned for diesel fuelled engine.

Cylinder Pressure

The cylinder pressure crank angle history is obtained for 100 cycles for the diesels at 80 % load injected with 5-hole injector having 0.2, 0.25 and 0.3 mm size orifice and the average pressure variation with crank angle is shown in Figure 24. The peak pressure depends on the combustion rate and on how much fuel is taking part in rapid combustion period. The uncontrolled combustion phase is governed by the ignition delay period and by the mixture preparation during the delay period. Therefore, mixture preparation during the ignition delay period is responsible for the variations of peak pressure and maximum rate of pressure rise. Injector with orifice size of 0.2 mm showed higher peak pressure compared to higher orifice sizes.

It is observed that, at the same brake power the IMEP is comparatively lower for the operation with 0.25 and 0.30 mm nozzle hole diameter compared to 0.20 mm diameter nozzle hole as the peak pressure and rate of pressure rise is lower with higher orifice injectors. Therefore diesel operation with 0.2 mm nozzle hole diameter results in higher peak pressure as shown in Figure 24 showing faster combustion and rapid pressure increase due to better fuel spray atomization and mixing. It could also be attributed to combined effect of longer ignition delay due to improper spray pattern and mixing rate caused by the use of 0.25 and 0.3 mm diameter nozzle hole size. Higher second peak during the diffusion burning phase was observed for these orifice sizes compared to 0.2 mm nozzle hole diameter. Poor quality of mixture formation and air entrainment and fuel air mixing rates could also be responsible for this trend.

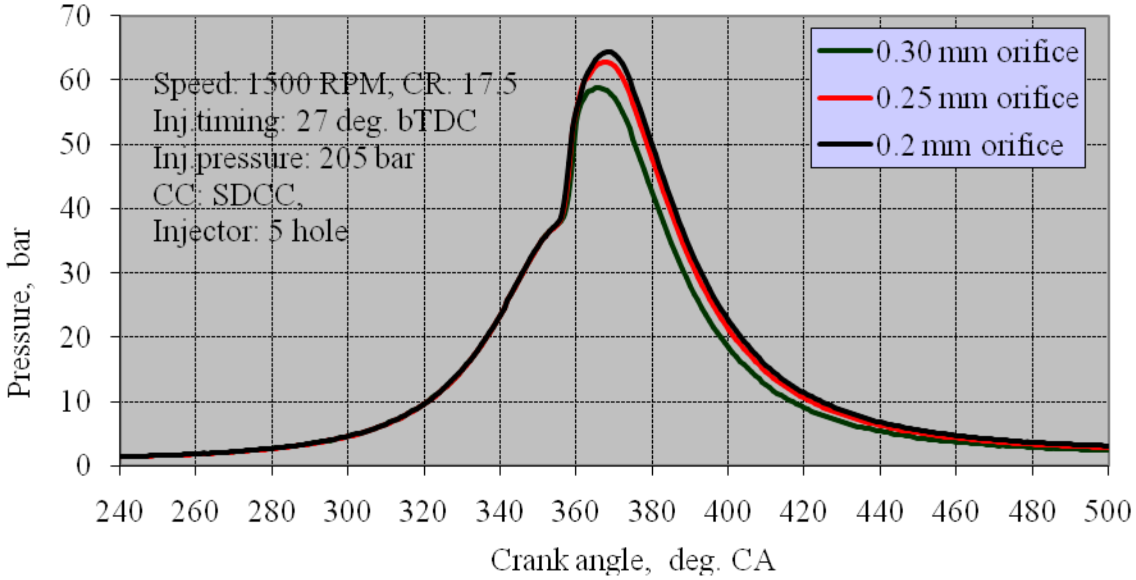


Fig. 24 Variation of pressure with crank angle for diesel using 5 hole injector of varying orifice diameters

Heat Release Rate

Figure 25 shows the variation of heat release rate for diesel injected with 5-hole injector having 0.2, 0.25 and 0.3 mm size orifice. The premixed burning phase associated with a high heat release rate is significant with diesel operation with 0.2 mm orifice which is responsible for higher peak pressure and higher rates of pressure rise. This is the reason for the higher brake thermal efficiency of diesel with this injector. The diffusion-burning phase indicated under the second peak is greater for 0.2 mm compared to 0.25 and 0.3 mm respectively. Significantly higher combustion rates during the later stages with these injectors leads to high exhaust gas temperatures and lower brake thermal efficiency.

The injector of 0.2mm orifice showed higher heat release rate when compared to that of 0.25 and 0.3 mm orifices. Fuel being common the size of injector orifice was obviously responsible for this behaviour. Higher heat release rate was observed for 0.2 mm nozzle hole size showing a more rapid combustion and concentrated heat release process, while flatter and broader heat release shapes were observed for higher nozzle sizes. Lower heat release rate resulted in lower pressure-rise rate, which benefits noise reduction.This is due to the result of higher second peak obtained with 0.25 and 0.3 mm nozzle in the diffusion combustion phase compared to 0.2 mm nozzle operation.

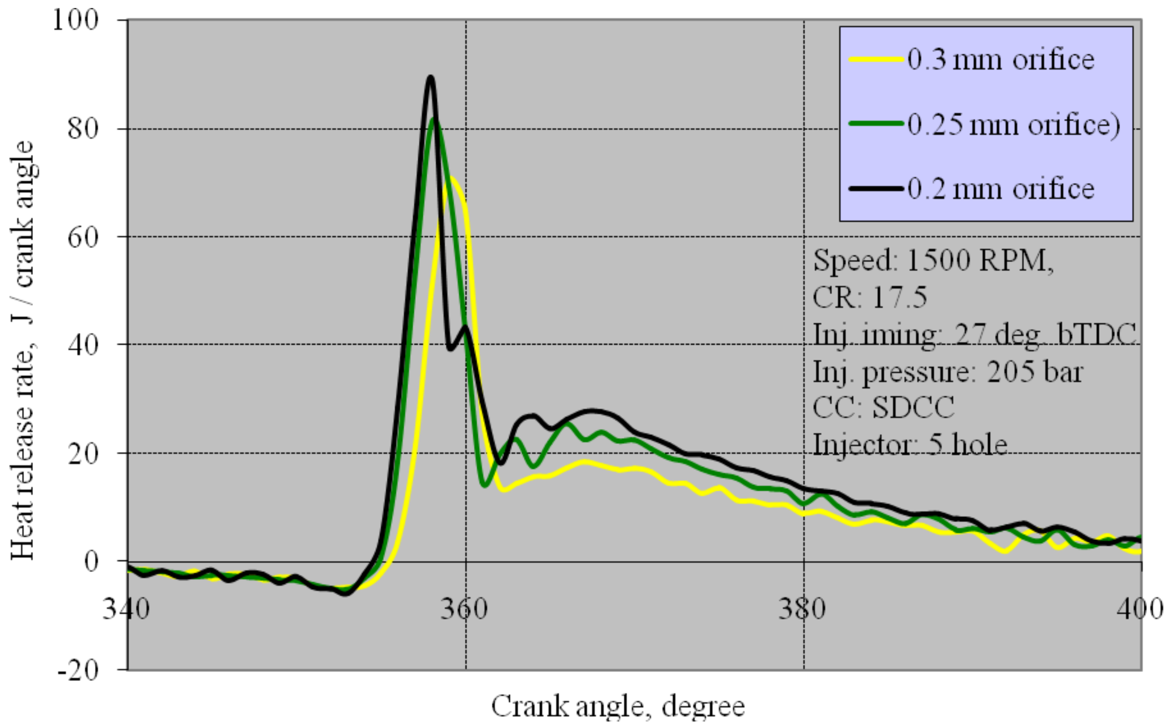


Fig. 25 Variation of heat release rate with crank angle for diesel using 5 hole injector of varying orifice diameters

Conclusions

From the exhaustive study it is observed that the performance of the diesel engine varies with injection timing and showed improvement in brake thermal efficiency when injection timing was advanced. Furthermore there was considerable decrease in smoke, HC and CO emissions with advanced injection timings but NO_x emissions increased. The following conclusion presented for optimized injection timing of 23° bTDC. Use of appropriate combustion chamber (SDCC) enhanced further the engine performance with better air fuel mixing ability. Increased injector holes were found suitable for operating CI engine using fossil fuel. Further reducing the size of 5 hole injector from existing 0.3 to 0.2 mm provided better combustion of fossil diesel with enhanced performance as well. Use of 5 hole injector with optimized injection timing of 23°bTDC, combustion chamber shape (SDCC), optimized injector orifice size resulted in overall improvement in the brake thermal efficiency and reduced emissions compared to engine operation under manufacturer specified conditions. The future work on diesel engines could involve diesel fuel modification with use of alcohol fuels such as ethanol, DEE (Diethyl Ether), DME (Dimethyl Ether). Such high volatile fuels can be either mixed with diesel fuel to form blended fuels or can be directly injected into the intake manifold with ECU (Electronic Control Unit) control for optimum engine operation in terms of improved engine performance and acceptable emission norms.

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