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Effect of Timing and Pattern of Fuel Injection on Performance and Emissions of a Diesel Engine in the Low-Temperature Combustion Mode – An Experimental Investigation

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ABSTRACT

This paper deals with experimental investigations on a four-cylinder, four-stroke, direct injection, turbocharged engine to evaluate effects of fuel injection strategies, exhaust gas recirculation (EGR) and intake boost pressure on the low temperature combustion (LTC) mainly to reduce nitric oxides (NOx) and smoke emissions keeping a concern on indicated mean effective pressure (IMEP). First, single-pulse fuel injection strategy with various fuel injection timings is tried to find the best operating conditions. Then, five fuel injection pulse strategy with a variation in injection timings and fuel quantities of each pulse along with the usage of EGR and intake boost pressure is tried to further improve the performance. Finally, it is found that for a low NOx and smoke emissions with a good brake thermal efficiency, five-pulse injection with the last pulse at closer to top dead centre along with the use of EGR in the LTC mode is a suitable one.

Keywords: LTC; Single injection; Multiple injections; Combustion; EGR.

NOMENCLATURE

CO	carbon monoxide	LTC	low temperature combustion
EGR	exhaust gas recirculation	NOx	nitric oxide
HCCI	homogeneous charge compression Ignition	PM	particulate matter
		P	pressure
HC	hydrocarbon	Q	appearent heat release
HRR	heat release curve	TDC	top dead center
IMEP	indicated mean effective pressure	V	volume
K	polytropic index		

1. INTRODUCTION

Diesel engines play a vital role as the prime mover in the automotive industry. However, the main problems with them are high emissions of nitric oxides (NOx), smoke and particulate matter (PM) in spite of their high brake thermal efficiency. Earlier, conventional diesel engines used a single fuel injection near the top dead center (TDC), which led to high NOx and soot emissions. This was because the fuel accumulated during the ignition delay period burned rapidly which increased the in-cylinder temperature and pressure quickly followed by diffusion combustion in which most of the fuel injected after the ignition delay period was burned. Modern diesel engines use multiple fuel injections (Kook, 2004), in order to overcome this problem, through the use of electronically controlled common rail fuel injection systems. These systems allow variations in fuel injection timing, fuel quantity, fuel injection pressure and also allow multiple injections in one engine cycle (Zhao, 2007; Horibe, 2009 and Torregrosa, 2013). Injecting a small quantity of fuel very early during the compression stroke increases the ignition delay (Drallmeler *et al*., 2018) and the possibility of formation of a very lean homogeneous mixture leading towards homogeneous charge compression ignition (HCCI) combustion which helps reduce NOx and smoke emissions simultaneously (Itoli, 2006). Wanhua *et al.*, (2005) injected diesel in six pulses to form a homogeneous mixture. Ming and Rajkumar (2009) injected diesel with eight pulses per cycle at an indicated mean effective pressure (IMEP) of 3 bar in order to obtain a homogeneous mixture and achieve HCCI combustion. Diesel HCCI combustion was achieved by Swaminathan *et al.*, (2009) using three fuel injection pulses in a cycle. The main advantage of HCCI combustion is the simultaneous achievement of low levels of soot and NOx emissions (Thring, 1989). Lean mixtures result in low local temperatures and hence result in low NOx emissions, and the homogeneity of the mixture reduces soot emissions (Mohamed, 2013). Najt and Foster (1983) conducted experiments in a four-stroke engine with a mixture of iso-octane and n-heptane in HCCI mode. Sofianopoulos *et al.*, (2018) studied thermal strafication effects using chemical kinetics. They found that the combustion in HCCI engines was controlled by chemical kinetics and high combustion rates in HCCI engines could be used to get high brake thermal efficiency if combustion was made to occur at the proper phasing.

However, there are lots of challenges with HCCI combustion before they can be made commercially successful. First, is to control the combustion phasing and the occurrence of autoignition over a wide range of operating conditions. Second, is to reduce CO and HC emissions which are high due to the incomplete combustion of the lean homogeneous mixture. As of now, the HCCI combustion can be achieved only at low and mid-load conditions, because of the excessive rate of pressure rise which can arise during combustion at high loads. Other problems such as the formation of a homogeneous mixture at all the loads also exist Song-Chang *et al.*, (2001). Therefore, at high load conditions, conventional diesel combustion is still preferred (Haiyun *et al.*, 2007).

In this paper, a study has been undertaken to find the effects of various fuel injection strategies on the performance, emission, and combustion characteristics of a diesel engine operating in the LTC mode. Here, a single pulse injection and a five-pulse injection with modulation of the injection timings, EGR and boost pressure variations were tried in a four-stroke, four-cylinder diesel engine. The final aim is to come out with a fuel injection strategy which can improve the emission characteristics without much penalty on the performance of the engine. In the five-pulse fuel injection strategy, especially with the late injection, the level of mixture homogeneity will be less. Hence, the combustion, in this case, is referred to as the LTC mode (Pandian *et al.*, 2017).

2. EXPERIMENTAL SETUP AND PROCEDURE

2.1. Experimental Setup

A four-cylinder, four-stroke, direct-injection, and turbocharged (with variable geometry turbine) engine was used to conduct the experiments (Table 1). Also, two intercoolers were used to maintain the inlet air temperature at a constant temperature of 36°C.

Table 1 Specifications of the test engine

Type	Four-cylinder, four-stroke
Displacement volume	2200 cm^3
Maximum power	105 kW @ 4000 rev/min.
Maximum torque	320 Nm @1750-2750 rev/min.
Compression ratio	17.2:1
Fuel injection	Common rail system

The engine was coupled to an eddy current dynamometer with a closed-loop controller for the measurement of speed and load. In-cylinder pressure was measured by a piezoelectric pressure transducer (accuracy ± 0.07% of full-scale (250 bar)) mounted on the first cylinder. The position of the crankshaft was determined by using an angle-encoder connected to the engine. An exhaust gas analyzer (Model MEXA 7100 DEGR, Make-Horiba) was used to determine the levels of HC, CO, NOx, and CO_2 emissions. The specifications of the analyzer are given in Table 2.

Table 2 Exhaust gas sample analyzer systems

Analysers	Species	Unit	Make
NDIR	CO	PPM	Horiba
NDIR	CO_2	%	Horiba
Para magentic	O_2	%	Horiba
CLA	NOx	PPM	Horiba
FID	HC	PPM	Horiba
Variable sampling smoke meter	Smoke/ dry soot	FSN	AVL

Variable sampling smoke meter was used to measure the smoke levels. The temperatures of the inlet fuel, inlet air, cooling water, lubricant oil and exhaust gas were measured by using suitable sensors. The coolant water and lubricant oil temperatures were maintained at about 75±5°C and 85±5°C respectively by using conditioning systems. A data acquisition system was used to register the in-cylinder pressure.

The specifications of the fuel injection system used is given in the Table 3.

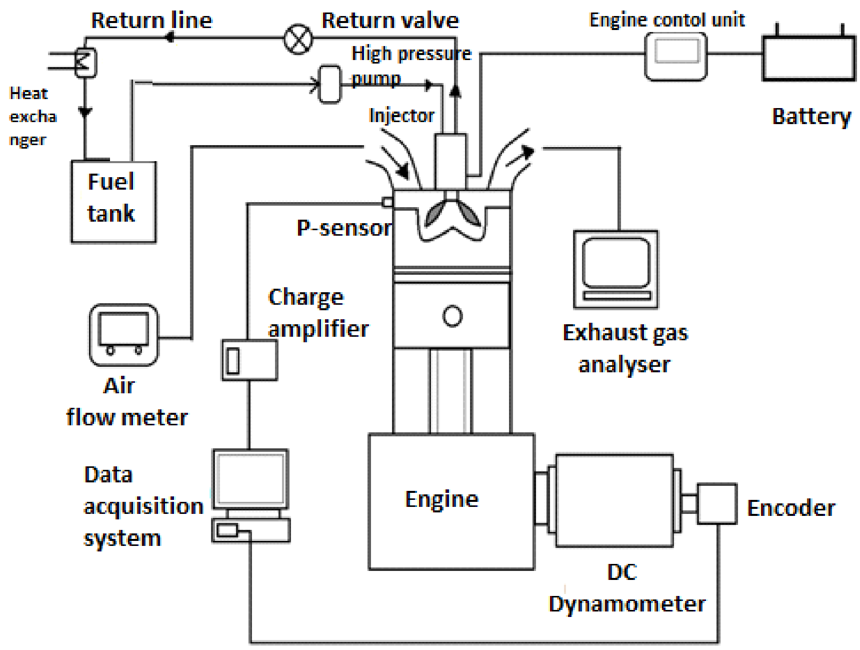

Fig. 1. Schematic of the experimental setup with instrumentation.

Table 3 Specifications of fuel injection system

Injector type	Solenoid
Nozzle type	Mini Sac
Number of holes	Six
Diameter of holes	0.142 mm
Cone angle of spray	153°
Maximum Injection pressure	1600 bar

Heat release rate and IMEP were calculated using commercial software with the data of 200 consecutive cycles. The following equation was used to calculate the apparent heat release rate.

$$\Delta Q = \left(\frac{k}{k-1}\right) p\Delta V + \left(\frac{1}{k-1}\right) V\Delta p \qquad (1)$$

where Q is the apparent heat release at a given crank angle, k is the polytropic index taken as 1.37, p is the pressure, and V is the volume (Brunt *et al.*, 1998)

The schematic diagram of the experimental setup is shown in Fig.1. Figure 2 shows a view of the engine setup used for testing. Figure 3 shows the modifications done to the intake air system with two intercoolers for maintaining the inlet air temperature. In the first phase of the experiments, a single fuel injection was used at the timing of 7.5° CA before TDC as in the conventional diesel mode. This timing was selected based on the best performance of the engine. Then, the fuel injection timing was advanced in steps of 10° CA up to 100° CA before TDC in order to achieve a large ignition delay and to shift into the LTC mode. In the second phase, five injections were used at 100°, 90°, 80°, 70° CA before TDC and at the TDC. The timing of the last injection pulse alone was varied after TDC in steps of 1° CA up to 10° CA after TDC. In the last phase of the experiments, the EGR and the boost pressure were applied. The following equation was used to calculate the EGR rate:

$$EGR\ rate = \frac{CO_{2\,inlet} - CO_{2\,ambient}}{CO_{2\ exhaust} - CO_{2\,ambient}} \qquad (2)$$

All the experiments were carried out at the IMEP of 3 bar. During the experiments, CO, HC, NOx and smoke emission levels, in-cylinder pressure data and exhaust gas temperature were measured. The test engine had a maximum torque in the engine speed range of 1750 to 2250 rev/min. Therefore, tests were conducted at the engine speed of 1800 rev/min., and at the common rail fuel pressure of 1200 bar.

2.2. The Strategy of the Experiments

This study mainly focuses on the effects of fuel injection strategies, EGR and boost pressure on the LTC. First, the single pulse fuel injection was tried with various fuel injection timings, at an IMEP of about 3 bar (this was maintained by adjusting the injected fuel quantity). In these conditions, combustion, performance and emission characteristics were studied. From this, the best operating conditions for the LTC mode was selected and compared with that of the conventional diesel combustion mode. Then, with the same quantity of the fuel as that of the single pulse injection (14 mg/ stroke), five injection pulses were used in order to study the influence on the performance, combustion, and emission characteristics of the engine. In this study, the first four injection pulses were set to occur at the injection timings of 100°, 90°, 80°, 70° CAs before TDC and the last pulse at

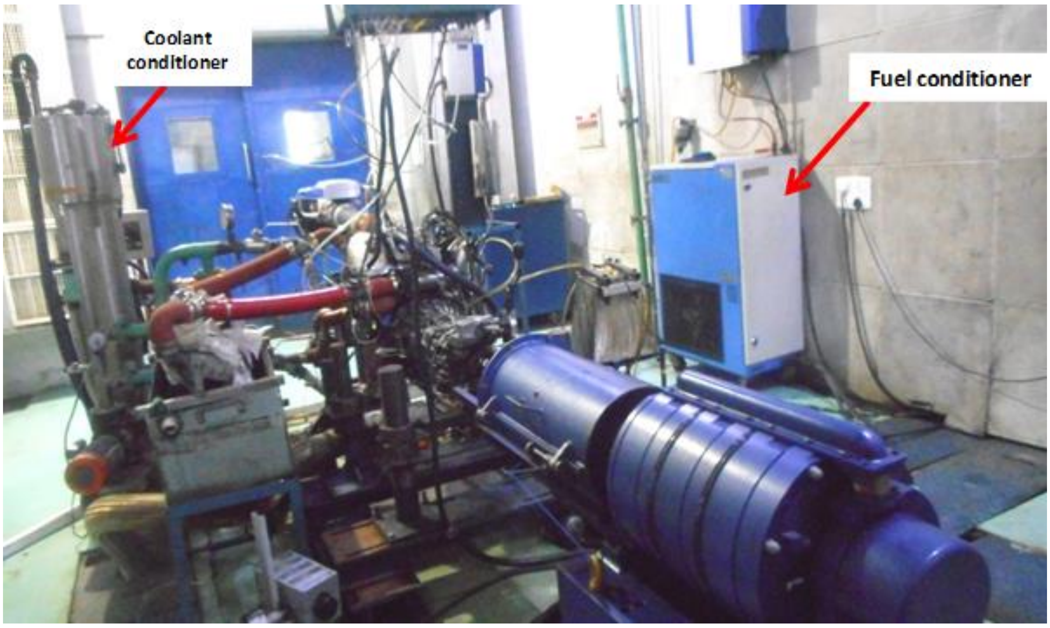

Fig. 2. Test engine setup with instrumentation.

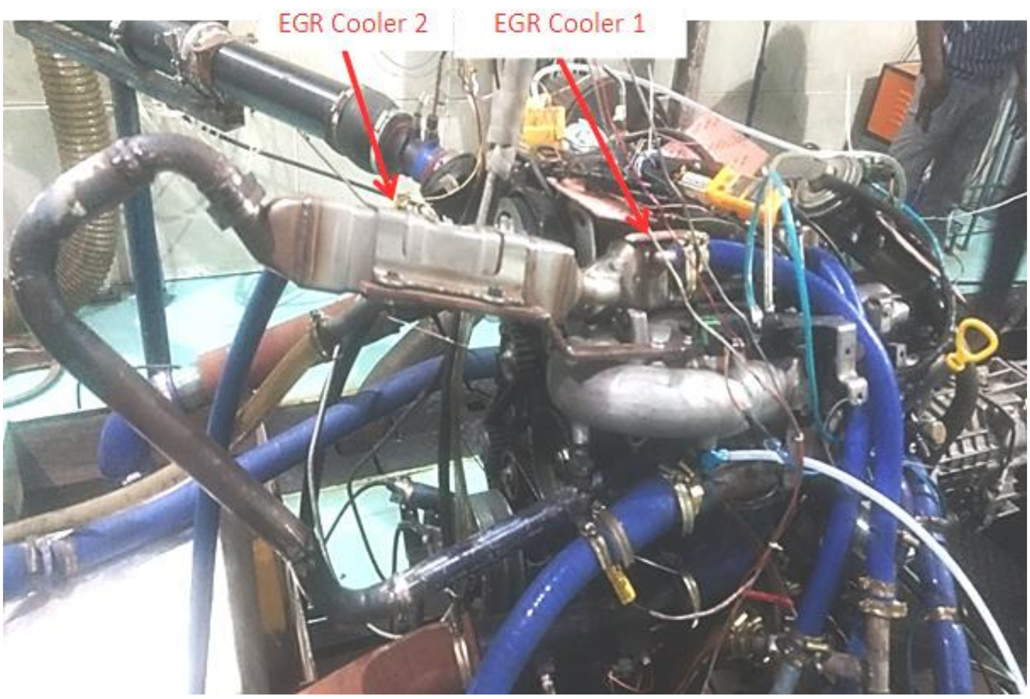

Fig. 3. Modification done to the intake air system with two EGR coolers.

TDC as per the recommendation of Mathivanan *et al*., (2016).

In their previous study, they observed high smoke, HC and CO emissions. Therefore, here, with five injection pulses, the influence of retarding the last pulse was studied as it would happen at the high charge temperature and help the oxidation of the combustion products. In addition, with five pulse injections, the effect of using EGR to delay the combustion to have proper combustion phasing in order to improve the brake thermal efficiency was also studied. Further, the effect of using high boost pressure along with the EGR, in order to raise the IMEP was also studied.

3. RESULTS AND DISCUSSION

Here, the results of performance and emissions of the engine in the diesel LTC mode with various fuel injection strategies are presented.

3.1. Diesel LTC Mode with Single Injection Pulse

Figure 4 shows variations of HRRs with the single pulse fuel injection at various fuel injection timings. From Fig.4, it is seen that at the fuel injection timing of 50° CA before TDC, the HRR is the maximum of about 290 kJ/CA among the fuel injection timings considered including that of the conventional CI mode. The HRR increases from about 233 kJ/CA at conventional CI injection timing to 290 kJ/CA at 50° CA before TDC. From Fig.4, it is seen that for the fuel injection timings beyond 50° CA before TDC, the HRR decreases. Also, there are two regions in the HRR curve viz., the cool flame and the main combustion (Pearlman *et al.*, 1999). The cool flame is identified by a small hump appearing before the main combustion (Haiyun *et al.*, 2007). In addition, the main combustion is delayed for the fuel injection timings beyond 50^0 CA before TDC. This is because as the fuel injection timing advances more time is available for the formation of the homogeneous mixture due to a longer ignition delay which results in slow premixed combustion. In addition, the early injection of fuel under low in-cylinder pressure and temperature conditions, more fuel hits the cylinder wall and leads to wall wetting which affects the vaporization of the fuel as well as the combustion. It may be noted that some of the fuel that hits the wall of the combustion chamber may not eventually participate in the combustion process at all as it may not be vaporized (Stefano *et al.*, 2018). From Fig.4, it is found that the peak HRR which occurs at about 12^0 CA before TDC in the case of the fuel injection timing 50° CA before TDC is higher by about 21% compared to that of the conventional CI mode which occurs at about 10^0 CA after TDC.

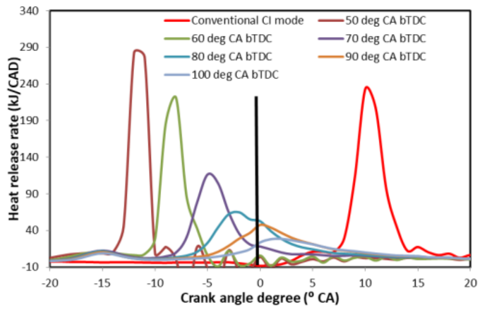

Fig.4. Variations of HRRs at various single pulse injection timings.

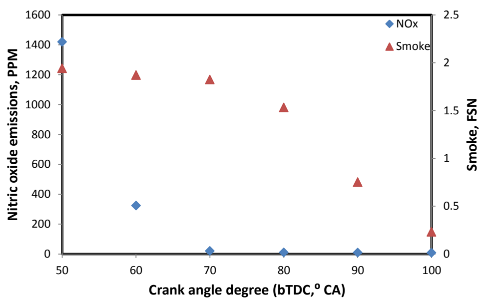

Fig. 5. Variations of NOx and smoke emissions at various single-pulse injection timings.

Figure 5 shows the variations of NOx emissions at various fuel injection timings with the single injection pulse. From Fig.5, it is seen that the NOx emissions are the highest at about 1418 PPM at the fuel injection timing of 50° CA before TDC (where the maximum heat release occurred). Also, the NOx emissions reduce significantly with the advancement of the fuel injection timing. The lowest emissions of NOx occur at the fuel injection timing of 100° CA before TDC, which is about 7 PPM. This is mainly because of the reduction in HRR and also the presence of a lean homogeneous mixture (thereby in-cylinder temperature reduces) with advanced fuel injection timings as shown in Fig.4. Wall wetting of the fuel also leads to less fuel participating in the main combustion process and thereby results in low NOx emissions.

Figure 5 also shows the variations of smoke emissions at various fuel injection timings with the single injection pulse. From Fig.5, it is seen that as the fuel injection timing is advanced, smoke emissions increase first (up to the fuel injection timing of 60° CA) and then decrease. However, the general trend is that they decrease with the advancement of the fuel injection timing. This is because the smoke formation is a strong function of the in-cylinder temperature and it requires a threshold in-cylinder temperature to form. As the in-cylinder temperature and pressure are low when the fuel injection advances, the smoke formation reduces. In addition, as the injection timing advances, the ignition delay increases and the mixture become more homogeneous before ignition which is also better from the smoke emission point of view (Kei *et al.*, 2018).

Figure 6 shows the variations of HC and CO emissions at various fuel injection timings with the single injection pulse. From Fig.6, it is seen that as the fuel injection timing advances, the HC emissions increase. This is because, when the fuel injection timing is advanced, the mixture is more homogeneous and also the fuel injection takes place when the in-cylinder temperature and pressure are lower. In addition, with the early fuel injection, the wall-impingement of the fuel also may take place (Nathan, 2007). All these factors lead to higher HC emissions. The lower in-cylinder temperatures under advanced fuel injection timings do not favour oxidation of HC.

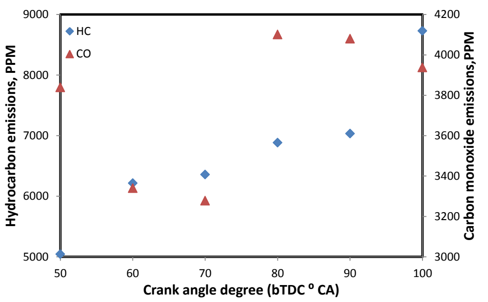

Fig. 6. Variations of HC and CO emissions at various single-pulse injection timings.

Figure 6 also shows the variations of CO emissions at various fuel injection timings with the single injection pulse. From Fig.6, it is seen that the CO emissions are in the range of 3200 PPM to 4100 PPM. Also, it is seen that the CO emissions decrease initially up to the fuel injection timing of 70° CA before TDC and then they increase. The fuel first oxidizes to CO and then into CO_2. The chemical equilibrium controls the conversion of CO into CO_2 at high temperature. Once the temperature drops below 1800 K, the conversion becomes kinetically controlled which is slower in nature (Pundir, 2011). From Fig.4, it is seen that after 70° CA before TDC, the heat release drops drastically, hence in-cylinder temperature decreases and leads to higher CO emissions. Also, from Fig.4, it can be seen that with early fuel injection (up to 70° CA before TDC), the entire heat release is rapid and occurs early. However, at retarded fuel injection timings, the heat release is slow and the temperature is low and thereby CO oxidation is inhibited (Cheng *et al*., 2013). This leads to higher CO emissions.

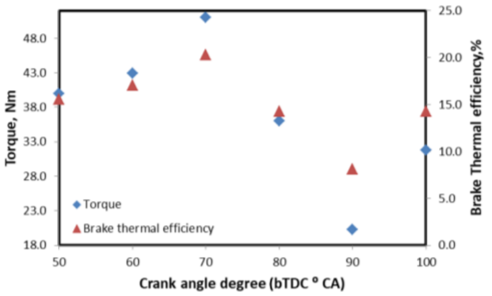

Fig.7. Variations of Torque and brake thermal efficiency at various single- pulse injection timings.

Figures 7 show variations of torque and brake thermal efficiency respectively, at various fuel injection timings with the single injection pulse. The trends of variations are the same in both the cases. When the brake thermal efficiency is more, the torque also is more. This is because the amount of fuel injected is the same in all the cases. From Fig.4, it is seen that at the injection timings of 50° and 60° CAs before TDC, the HRRs are quite high, but they are quite advanced, which will reduce the brake thermal efficiency. However, with the injection timings after 70° CA before TDC, the combustion occurs closer to TDC which results in better combustion phasing and may yield good brake thermal efficiency. However, the HRR is low as discussed earlier, which may reduce brake thermal efficiency. Thus, the maximum brake thermal efficiency occurs at the injection timing of 70° CA before TDC, where the HRR is high and combustion phasing also is proper.

From the above discussion, it is found that the single injection pulse at 70° CA before TDC is better among the various fuel injection timings considered regarding torque, brake thermal efficiency and emissions are concerned. It is also found that compared to that of the conventional diesel injection at 7.5° CA before TDC, both CO and HC emissions are higher by about 80%, and the torque and brake thermal efficiency are lower by about 25% and 40% respectively, and the NOx and smoke emissions are lesser by 96% and 18% respectively. Figure 8 compares the various parameters of the conventional CI and the LTC modes with a single injection pulse at 70° CA before TDC. Single injection after 70°CA before TDC is considered as LTC mode because the NOx emissions are significantly lower than that of the previous injection timings.

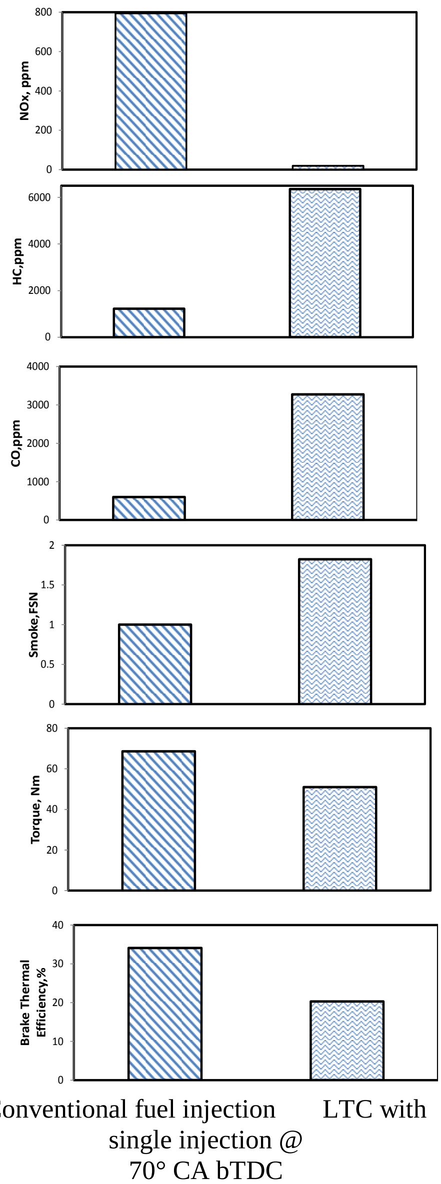

Fig. 8. Comparison of performance and emission characteristics of the conventional CI and the LTC modes with single pulse injection at 70° CA bTDC.

3.2. Analysis of Diesel LTC Mode with Five Pulse Fuel Injection

From the above experiments, it is found that with the single injection pulse i.e., with a large fuel quantity per pulse and early injection, the performance deteriorates because of the fuel spray impingement on the combustion chamber walls. It is also reported in the literature that the wall-impingement can be reduced in LTC mode through the use of split injection (Cheng *et al*., 2013). For the present study, for five injection pulses, the timings of 1000, 900, 800, 700 CAs before TDC and at TDC are used because these timings gave better performance as per Mathivanan *et al*., (2016) . The number of injections in the present case is also restricted to five because of system limitations. The problems found in the above work are high smoke, HC and CO emissions. Therefore, in the present study, an attempt has been made to retard the last of the five injection pulses, so that it could have a favourable effect on the post oxidation of soot, HC, and CO.

For the present study, the total fuel quantity/cycle (14 mg/stroke) is kept the same as earlier in the single pulse injection case but distributed among five injection pulses. The first four injection timings are kept at 100°, 90°, 80° and 70° CAs before TDC. The last injection pulse is varied from TDC to 10° after TDC.

However, from the previous study of Mathivanan *et al*., (2016) it is found that the fuel injection timings of 100°, 90°, 80°, 70° CA before TDC, with the fuel injection pulse durations of 373 µs for each, and the last injection pulse at the TDC with the pulse duration of 274 µs were the best combination from the point view of brake thermal efficiency and emissions at the IMEP of 3 bar . Table 4 and 5 show the various results of the previous study of Mathivanan *et al*., (2016) with their operating conditions and also the results of the present study with the single pulse fuel injection at 70° CA before TDC. From Table 4 and 5, it is seen that when the fuel was injected in five injection pulses instead of a single injection pulse, the HC, CO, and smoke emissions decreased because of the reduced wall wetting, but the NOx emissions increased marginally. Also, the brake thermal efficiency was higher with five injection pulses than that of the single injection pulse because of the better combustion phasing. It was also found that the last injection pulse played a vital role in the level of emissions. But, its timing was limited by the time available for mixing and therefore the fuel quantity in the last pulse had to be low (Wanhua *et al*., 2005). Hence, in the present study, the last injection pulse retarded beyond the TDC and the effect of it on the performance of the engine has been studied. Here, the last injection pulse timing is varied from the TDC to 10° CA after TDC in the step of 1° CA. The last injection pulse duration is kept as 274 µs and other four injection pulses are kept as 373 µs.

3.2.1. Effect of Last Fuel Injection Pulse Timing (after TDC)

Figure 9 shows the variations in the HRRs with the last injection pulse timings when five injection pulses are used. There are three peaks in the HRR curves. These are because of the cool flame, main combustion, and the post-combustion respectively. From Fig.9, it is found that as the timing of the last injection pulse retards, the occurrence of the cool flame and the main combustion advance. However, the occurrence of the post-combustion retards. The post-combustion occurs because of the last fuel injection pulse. The fuel in the last injection pulse is injected when the in-cylinder temperature is low. The probable reason for the advancement of the main combustion is that when the last injection pulse retards, the fuel in the last injection pulse burns very late during the expansion stroke. This results in a very little heat release during the cool flame as seen in Fig.9. During this period, certain radicals produced may enhance the reactivity during the cool flame and make it advance and hence the main combustion. However, more investigations are required to support this.

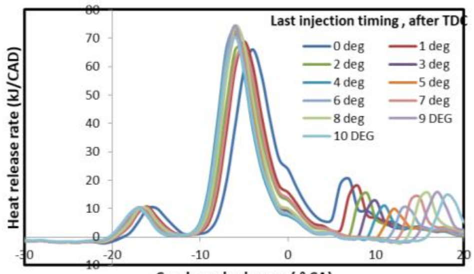

Fig. 9. Variations of HRRs with last injection pulse timings with five pulses injection.

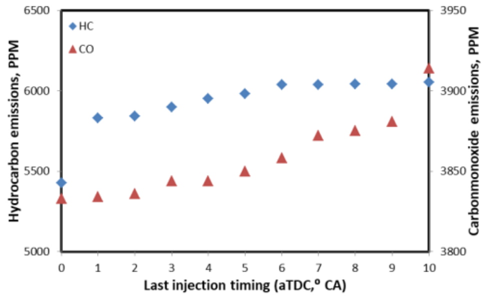

Fig. 10. Variation of HC and CO emissions with last injection pulse timing with five pulses injection.

Table 4 Comparision of Performance of the diesel LTC engine with single and five injection pulses

Injection pattern	Torque (Nm)	Brake thermal efficiency (%)
Five injection pulses	44.5	22
Single injection pulse	51	20.3

Table 5 Comparision of emission characteristics of the diesel LTC engine with single and five injection pulses

Injection pattern	HC (ppm)	NOx (ppm)	CO (ppm)	Smoke (FSN)
Five injection pulses	3743	26	2970	1.5
Single injection pulse	6357	19	3278	1.8

Figure 10 shows the HC and CO emissions with the various last injection pulse timings in the case of five injection pulses. From Fig.10, it is found that as the last injection pulse timing retards, the levels of the HC emissions increase. This is because when the last injection pulse retards, the in-cylinder temperature and pressure decrease because of the expansion and thus affect the post oxidation of HC and thereby the HC emissions are enhanced.

Figure 10 also shows the carbon monoxide emissions with the various last injection pulse timings in the case of five injection pulses. The fuel first burns into CO and then oxidizes to CO_2. The oxidation of CO depends upon the availability of oxygen in the cylinder and the in-cylinder temperature. Since the last injection pulse occurs during the expansion stroke, the in-cylinder temperature is low as mentioned earlier, which inhibits the CO oxidation. In addition, the oxidation of CO into CO_2 may be inhibited because of low in-cylinder temperature.

Figure 11 shows the NO_x emissions with the various last injection pulse timings in the case of five injection pulses. The NO_x formation is a function of in-cylinder temperature and availability of oxygen. As discussed above, the last injection pulse is set during the expansion stroke, while the in-cylinder temperature is low. and thereby further suppressing the formation of NO_x.

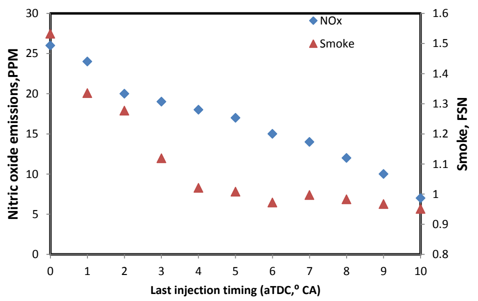

Fig. 11. Variation of NOx and smoke emissions with last injection pulse timing with five pulses Injection.

Figure 11 also shows the smoke emissions with the various last injection pulse timings in the case of five injection pulses. As the last injection pulse retards, the smoke emissions decrease. This is because of the low in-cylinder temperature as smoke formation needs a threshold of it.

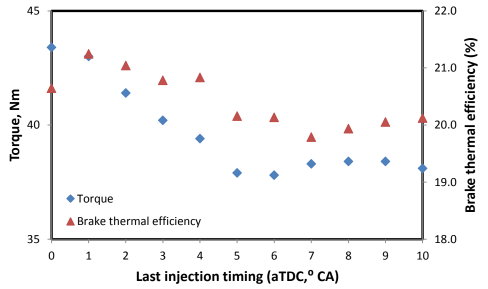

Fig. 12. Variation of Torque and Brake thermal efficiency with last injection pulse timing with five pulses injection.

Figure 12 showsthe variations of torque and brake thermal efficiency respectively with the various last injection pulse timings in the case of five injection pulses. From Fig.12, it is seen that as the last injection pulse timing retards, the torque and brake thermal efficiency decrease. It may be noted that under these conditions, the main combustion advances which affects the combustion phasing (combustion, on the whole, is too advanced) and thus lowers the brake thermal efficiency (Fig.9).

Generally, it is found that NOx and smoke emissions decrease, and the HC, CO, and fuel consumption increase when the last injection pulse is retarded beyond TDC.

3.2.2. Analysis of Diesel LTC with Multiple Fuel Injection Pulses and EGR

From the above discussion, it is found that the last injection pulse beyond TDC resulted in lower NOx and smoke emissions, however, the HC, CO and fuel consumption increased. Therefore, for the present study, the timing of the last injection pulse is set to occur at TDC (Mathivanan *et al.*, 2016). But, with these conditions, the combustion phasing is too advanced which may reduce the brake thermal efficiency (Garcia *et al.*, 2009). Therefore, in order to get the proper combustion phasing, cold EGR is used along with the last injection pulse kept at TDC (Andre, 2012; Cody *et al.*, 2016).

Here, the EGR levels up to 40% in steps of 10% in a combination of five injection pulses were tried. The timings of first four injections were maintained at 100°, 90°, 80°, 70° CA before TDC, along with pulse durations of 373 µs for each. The last injection pulse is kept at TDC along with a duration of 274 µs. The IMEP at these conditions was closer to 3 bar.

Figure 13 shows the variations of heat release rates with the EGR and multiple fuel injection pulses. From Fig.13, it is found that as the EGR rate increases, the combustion retards, but the maximum heat release rate decreases. This is because of the reduction in oxygen concentration in the charge due to the addition of the EGR. However, the retarded combustion process resulted in an increase in thermal efficiency.

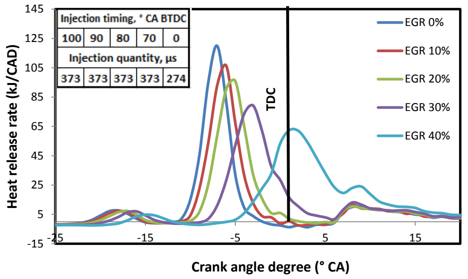

Fig. 13. Variations of HRRs with EGR rate with five pulses injection.

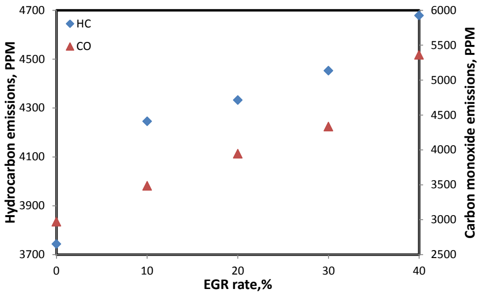

Fig. 14. Variations of HC and CO emissions with EGR rate with five pulses injection.

Figure 14 shows variations of HC and CO emissions respectively with the EGR and the multiple fuel injections. From Fig.14, it is seen that as the EGR rate increases, the HC and CO emissions also increase. This is mainly because of the reduction in the concentration of oxygen by the dilution because of the EGR. In addition, the use of EGR also lowers the temperatures after combustion.

Figure 15 shows the variations of NOx and smoke emissions with the EGR and the multiple fuel injections.

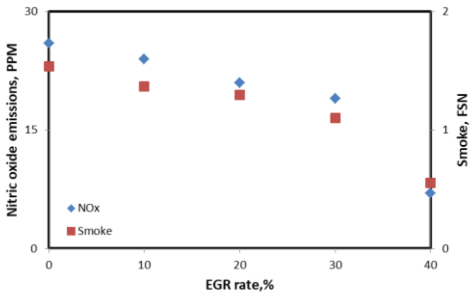

Fig. 15. Variations of NOx and smoke emissions with EGR rate in five pulses injection.

From Fig.15, it is seen that as the EGR rate increases, the NOx emissions reduce further. This is mainly because of the reduction in the in-cylinder temperature and the concentration of the oxygen in the mixture at higher EGR rates.

Figure 15 also shows the variations of smoke emissions with the EGR in five pulses injections. From Fig.15, it is seen that as the EGR rate increases, the smoke emissions reduce. This is mainly because of the formation of a more homogenous mixture at higher EGR rates. At higher EGR rates, because of the dilution, ignition delay increases, which helps form a homogeneous mixture. This is also the main advantage of the LTC in which simultaneous reduction of both NOx and smoke emissions are possible.

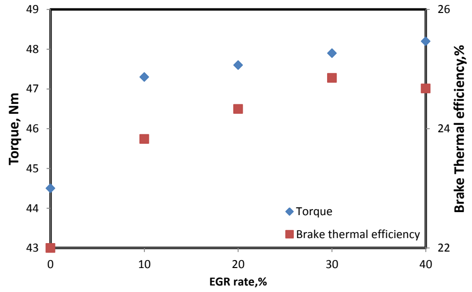

Fig. 16. Variations of torque and Brake thermal efficiency with EGR rate with five pulses injection.

Figure 16 shows the variations of torque with the EGR and the multiple fuel injections. From Fig.19, it is seen that as the EGR rate increases, torque increases. This is mainly because of the proper combustion phasing of the main combustion (Asad, 2015).

Figure 16 also shows the variations of brake thermal efficiency with EGR and multiple fuel injections. From Fig.16, it is seen that as the EGR rate increases, the thermal efficiency increases. This is again mainly because of the increase in the torque at higher EGR rates.

3.3. Analysis of Diesel LTC with Multiple Fuel Injections along with EGR and Boost Pressure

The EGR decreases the equivalence ratio of the mixture and thereby reduces NOx and smoke emissions. As the EGR level increases, it lowers the local oxygen concentration also. This leads to a reduction of oxidation of CO to CO_2. When the EGR is used, the amount of the fresh charge inducted into the cylinder reduces (thereby oxygen concentration reduces) which helps reduce the NOx emissions. Hence the engine was boosted to increase the oxygen supply so as to reduce CO emissions. In this case, the boost pressure of 1.1 bar and 1.2 bar along with the EGR were tried using five injection pulses. Air from the turbo-compressor was passed through an intercooler and the exhaust gas was passed through an EGR cooler. The mixture of the fresh air and the EGR was kept at a constant temperature of 36°C by adjusting the cooling water flow rate of the EGR cooler. When the EGR was used without boosting, there was an increase in the CO emissions by about 62% than that of the engine with five pulse injection. When the boosting was done along with the EGR, there was a reduction of CO emissions by about 50% than that of the engine with five pulse injection. The combustion phasing got retarded with the increase in EGR level for the given boost pressure as shown in the Figs.17 and 18.

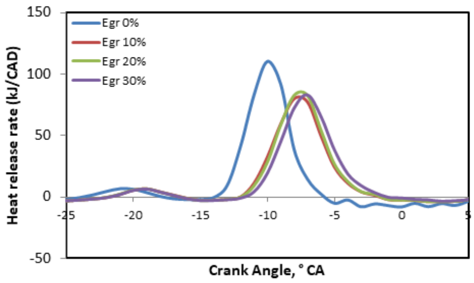

Fig. 17. Variations of HRRs with the EGR rate at the boost pressure of 1.1 bar.

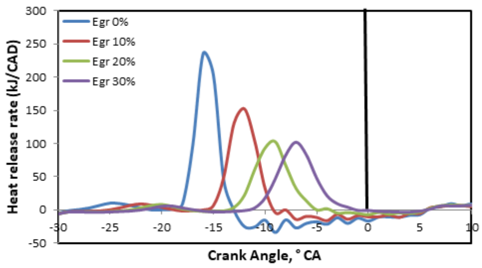

Fig. 18. Variations of HRRs with EGR rate at the boost pressure of 1.2 bar.

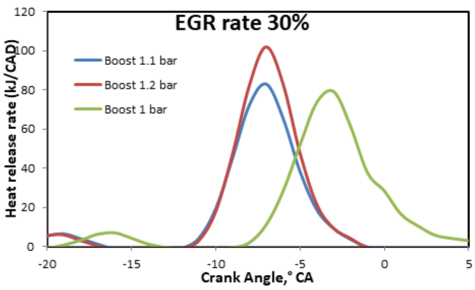

Fig. 19. Variations of HRR with boost pressures at the EGR rate of 30%.

However, for the given EGR level, as the boost pressure increases, the combustion advances as shown in Fig.19 at the EGR rate of 30%. As the boost pressure increases from 1 bar to 1.1 bar and 1.2 bar, the availability of oxygen increases which increases the combustion rate and thus early combustion takes place. This also increases the heat release rate. As more airflow takes place, accordingly more fuel consumption and this increases the torque. However, the brake thermal efficiency of the engine reduces by about 20% than that of without boost pressure due to improper combustion phasing.

4. CONCLUSIONS

In this study, experimental investigations were carried out on a diesel CI engine in the LTC mode with various fuel injection strategies along with variations in the EGR rate and the boost pressure to evaluate their effects on combustion, performance and emission characteristics. From the analysis of the results, the following conclusions are drawn:

- For the single pulse injection with the fuel injection timing of 70° CA before TDC, the NOx and smoke emissions reduced by about 96% and 18% respectively, and both the HC and CO emissions increase by about 4 times than that of the conventional CI mode.
- For the single pulse injection with the fuel injection timing of 100° CA before TDC, the NOx and smoke emissions reduced by about 99% and 77% respectively, and the HC and CO emissions increased by about 6 and 5 times respectively than that of the conventional CI mode.
- For the five pulses injection case with the timings of 100°, 90°, 80°, 70° and 0° CAs before TDC, the HC and CO emissions reduced by about 40% and 10% respectively compared to that of the single pulse injection at 70° CA before TDC.
- For the five pulses injection case with the timings of 100°, 90°, 80°, 70° and 0° CAs before TDC, the brake thermal efficiency increased 22% from 20% of the single pulse injection case.
- For the five injection pulses, with 40% EGR, the brake thermal efficiency increased to 25% from 22% of that of no EGR. However, the HC and CO emissions increased by about 27% and 80% respectively compared to those of without EGR.
- For the five-pulse injection strategy, with 30% EGR rate and the boost pressure of 1.2 bar, the brake thermal efficiency reduced by about 20% compared to that of without boost pressure. However, the HC and CO emissions reduce by about 25% and 20% respectively.

Finally, it is concluded that for low NOx and smoke emissions with good brake thermal efficiency, the multiple pulse injection with the last pulse closer to TDC along with the use of EGR is suitable for LTC mode of operation.

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