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ABSTRACT

The pursuit of new combustion concepts or modes is ongoing to meet future emissions regulations and to eliminate or at least to minimize the burden of aftertreatment systems. In this research, Premixed Low Temperature Diesel Combustion (PLTDC) was developed using a single-cylinder engine to achieve low NOx and soot emissions while maintaining fuel efficiency. Operating conditions considered were 1500 rpm, 3 bar and 6 bar IMEP. The effects of injection timing, injection pressure, swirl ratio, EGR rate, and multiple injection strategies on the combustion process have been investigated.

The results show that low NOx and soot emissions can be obtained at both operating conditions without sacrificing the fuel efficiency.

Low NOx and soot emissions are achieved through minimization of peak temperatures during the combustion process and homogenization of in-cylinder air-fuel mixture.

INTRODUCTION

Conventional diesel combustion is composed of a rich premixed combustion phase followed by mixing-limited diffusion combustion phase after compression ignition of fuel injected into the cylinder. It is well known that the ignition delay and the amount of fuel premixed before ignition has a strong impact on engine-out emissions.

Several different investigations have successfully demonstrated the capability of premixed combustion to reduce both NOx and soot emissions simultaneously. First, homogeneous charge compression ignition (HCCI) combustion has been widely studied as a premixed combustion technology [1, 2]. It is necessary to inject fuel at a fairly early time in the initial stages of the compression stroke. A very long ignition delay period is required to form a lean and uniform (homogeneous) mixture. This combustion achieves a simultaneous reduction in NOx and soot emissions primarily by premixing the fuel and air to overall lean conditions. However, HCCI combustion still has difficulties of control of combustion phasing and unacceptably high HC and CO emissions due to fuel wall impingement.

In order to overcome fuel-wall wetting problems, Iwabuchi et al. [3] developed premixed compression-ignited (PCI) combustion using an impinged-spray nozzle which has a low penetration, high dispersion and high injection rate. As a result, an extremely low NOx emission level was realized but fuel efficiency was deteriorated slightly. They also found that engine operating range possible with PCI combustion was found to be limited to part-load conditions, and moreover, PCI combustion was found to still generate high hydrocarbon (HC) emissions. As an alternative approach, Walter et al. [4, 5] suggested narrow angle direct injection (NADI) technology. They applied HCCI combustion to part load and obtained near zero particulate and NOx emissions while maintaining very good fuel efficiency close to EURO III Diesel engine. But these strategies require substantial hardware modification of engines such as changes in fuel injection equipment (FIE) and combustion chamber geometries. These changes can lead to less desirable performance at high-loads, when conventional diesel combustion operating modes are required.

One approach to address this problem involves retarding start-of-injection (SOI) timing to near top dead center (TDC). Main advantages of this combustion are two-fold. One is that combustion phasing is determined by the fuel injection event, and the other is that it may not require modification of the production diesel fuel injection equipment or combustion chamber geometries. As a result, conventional combustion mode will be applied at high loads without any sacrifice.

Many researchers have worked to develop new combustion mode using near-TDC injection technology. This combustion will not be pure homogeneous combustion but highly premixed combustion. Shimazaki et al. [6] investigated the characteristics of premixed diesel combustion (PDC) with fuel injection timings near TDC. They achieved a simultaneous reduction in NOx, soot, and BSFC using commercial diesel fuel. They used high EGR in order to achieve enough mixing time to reduce the amount of fuel-rich mixture. They found that mixture formation depends heavily on "high turbulent mixing rates," which were obtained by high injection pressures with small-hole nozzles and near-TDC injection. It was also pointed out that low total hydrocarbon (THC) emissions can be obtained through near-TDC injection due to a mixture that does not disperse into the squish region.

Kanda et al. [7, 8] also developed premixed charge compression ignition (PCCI) combustion with two different hardware configurations. One is early injection strategy into shallow-dish piston bowl combined with a narrow nozzle angle setting, and the other is with fuel injection near TDC into modified re-entrant piston bowl combined with conventional nozzle angle. It was found that regardless of the fuel injection timing, increasing EGR reduced NOx emissions. However, fuel consumption, soot, HC and CO emissions obtained at fuel injection near-TDC was superior to those obtained at earlier injection due to the reduction in the fuel wall-film formation. They also found that a slightly modified re-entrant combustion bowl shape provided better performance for both PCCI combustion operation and conventional combustion operation in terms of exhaust gas characteristics, fuel economy, and combustion stability.

Kimura et al. [9, 10] retarded injection timing further and developed after-TDC injection strategies to achieve simultaneous low NOx and soot emissions over a low-to-moderate speed and load range. They conducted experimental investigations of, so-called MK combustion, using a single and multi-cylinder DI diesel engine. It was concluded that in order to reduce NOx and soot emissions at the same time, the ignition delay should exceed the injection duration to ensure that all of the fuel has enough time to pre-mix. To shorten the injection duration, high injection pressures and larger hole nozzle were adopted. Furthermore, a high swirl ratio is employed to suppress the formation of unburned hydrocarbons and soluble organic fraction (SOF) of the particulate, although a small NOx penalty was observed.

In the premixed combustion concept, NOx formation is maintained low by implementing high rates of cooled EGR, late injection timings, and lower compression ratios. Each of these factors also extends the ignition delay, allowing a greater time for fuel-air premixing – which is thought to be the dominant factor suppressing soot formation.

On the other hand, there is another approach to reduce NOx and soot emissions at the same time. It is well known that NOx is formed at low equivalence ratios and in high temperature regions, while soot is formed at high equivalence ratios and in a specific temperature region. Thus, in order to avoid NOx and soot formation region at the same time, combustion temperature must be low enough. The reductions in flame temperature due to EGR lead to a reduction in NOx emissions because of increased specific heat capacity and a decreased oxygen concentration at the flame region [11, 12] and also lead to a decrease in soot emissions [13]. For these reasons, Low Temperature Combustion (LTC) has received a lot of interest.

Akihama et al. [13] developed so-called "Smokeless Rich Combustion" and obtained near-zero NOx and soot emissions under near stoichiometric and even rich operating conditions. They also proposed a quantitative ϕ - T map by performing 0-D calculations using a detailed soot formation model. Through this analysis, it was concluded that combustion temperature with high EGR is low enough to suppress soot formation in smokeless rich combustion.

Bianchi et al. [14] numerically investigated the possibility to extend low-temperature combustion developed for low-load conditions to medium-high loads. They found that it is very difficult because the maximum allowable EGR rate diminishes and greater chemical energy is released due to the large amount of fuel injected. They concluded that improvements in mixture formation are of particular importance to reduce soot formation at medium load.

Miles et al. [15] identified the rate-limiting physical processes dominating various phases of a low-temperature, late-injection combustion event by examining the effects of EGR rate, injection timing, engine speed, injection pressure, swirl ratio, and intake temperature on the heat release rates, engine emissions, and spatial distributions of combustion luminosity. They found that the low-temperature combustion regime more closely resembles a standard, two-stage diesel combustion process rather than a fully premixed (homogeneous or otherwise) process.

Aceves et al. [16] have developed a model of the diesel fuel injection process for application to analysis of low temperature non-sooting combustion. The model predicts chemical composition and soot precursors, and is applied at conditions that result in low temperature non-sooting combustion. They found that soot precursors are low at the low EGR cases. Precursors then increase rapidly as the EGR fraction is increased, reaching a maximum and then decreasing rapidly to near zero as the mixture approaches stoichiometric conditions. It was also pointed out through a parametric analysis that increasing the gas temperature and reducing the rate of mixing tend to increase the equivalence ratio at the time of ignition, considerably increasing the soot precursor production.

Fang et al. [17] visualized the whole cycle of low temperature combustion process using a high-speed digital camera by imaging natural flame luminosity in an optically accessible single cylinder small-bore High-Speed Direct-Injection (HSDI) diesel engine. Results show that low temperature combustion is feasible in an HSDI diesel engine with a higher injection pressure, a higher EGR rate, or later injection timing with little penalty in power output.

In the current research, premixed low temperature combustion was developed using a single cylinder engine by combining the advantages of premixed combustion and low-temperature combustion. The main objective of this study is to reduce NOx and soot emissions simultaneously while maintaining the diesel engine's good fuel efficiency, HC and CO emissions.

EXPERIMENTAL SETUP

The test engine used in the present study is a specially designed single-cylinder diesel HYDRA engine. The specifications of the engine are shown in Table 1. Aluminum cylinder head, cylinder block, cylinder liner, crankshaft, connecting rod assembly, and piston were re-designed for 200 bar maximum cylinder pressure. Inlet and exhaust cam timing can be manually adjusted for VVA applications.

Engine Type	4 valve DOHC diesel
Bore x Stroke	84.0 x 90.0 mm
Compression Ratio	15.00:1
Displacement	499 cm^3
Max. Cylinder Pressure	200 bar
Piston Geometry	re-entrant bowl (BMW M57)
Intake Ports	1 swirl, 1 tumble
Swirl ratio (at IVC)	1.0 ~ 4.0

Table 1. The Engine Specifications

The injection system used for this study was a DELPHI common-rail injection system. The injector was an electro-hydraulically controlled solenoid injector. Cylinder head was design-protected for 17mm and 19mm injectors, and injectors could be interchanged without removing the engine cover. The specifications of the common rail injection system are summarized in Table 2.

Injector type	Solenoid Injector
Injection pressure	15 ~ 160 MPa
Max. number of injections	5
Number of nozzle holes	6
Flow Rate	0.755, 0.84 l/min
Included spray angle	150°

Table 2 Common rail system specifications

The fuel used during the experiments was 2007 low sulfur diesel certification fuel. The cetane number of the fuel was 43.

A schematic diagram of the engine laboratory setup is shown in Fig. 1. The intake and exhaust systems allowed simulated turbo-charged operation with any combination of intake and exhaust pressures. The intake air temperature was controlled using a 6kW Chromalox air heater. EGR was controlled by a V-cut ball valve located between the exhaust and intake surge tanks. EGR was cooled by a water-cooled heat exchanger and the temperature was controlled to maintain intake air temperature. Two ten-gallon pressure tanks were used as the intake and exhaust surge tanks in order to lessen pressure fluctuations and to mix the EGR well with the fresh intake air.

EGR was defined as follows:

$$EGR = \frac{(CO_2)_{\text{int}} - (CO_2)_{atm}}{(CO_2)_{exh} - (CO_2)_{atm}} \times 100$$

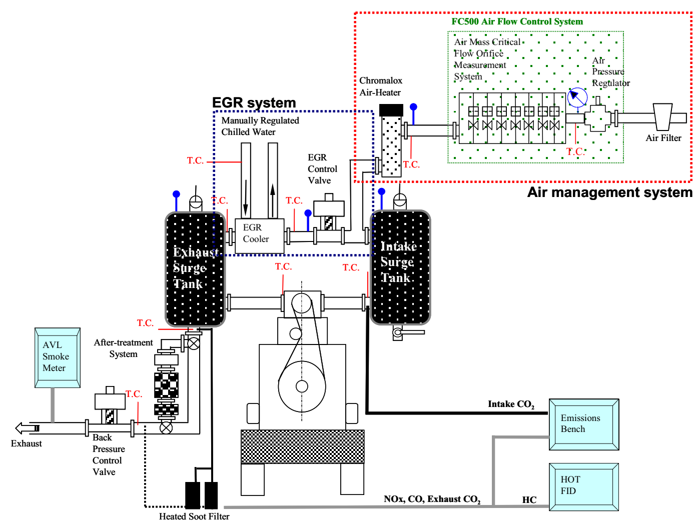


Fig.1: The Schematic Diagram of the engine setup

A DSP ACAP system was used to measure cylinder pressure at 1 crank angle degree (CAD) increments. Cylinder pressure was averaged for 100 cycles and analyzed to calculate the apparent heat release rate using the First Law of Thermodynamics [18].

The emissions data recorded during the experiments were comprised of both gaseous and particulate emissions. The gaseous emissions, including NOx, HC, CO, Intake and Exhaust CO_2, were sampled at the exhaust surge tank and measured with a HORIBA emissions bench. The samples passed through heated soot filter, heated sample lines, and a sample conditioning unit. With the use of the heated filter and line, the sample was kept above 190°C to help prevent emissions species from condensing. Exhaust smoke levels were sampled with an AVL 415S smoke meter.

RESULTS AND DISCUSSION

Premixed low-temperature diesel combustion was developed at two operating conditions, which are 1500 rpm, 3 bar IMEP and 1500 rpm, 6 bar IMEP. In order to achieve low NOx and soot emissions, different strategy was applied at each operating condition.

1500 rpm, 3 bar IMEP - Baseline test was performed at 1500 rpm, 3 bar IMEP operating condition. After performing injection timing sweep test around TDC, best efficiency point was chosen as a baseline. Fig.2 and Table 3 show the results of baseline. Baseline shows typical low-load emissions characteristics of diesel combustion which is high NOx and low soot. In this low load condition low soot emissions can be obtained due to high oxidation rate.

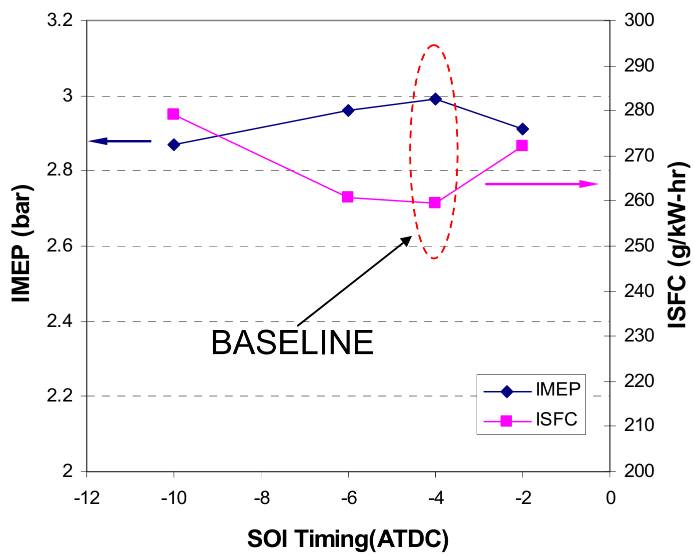


Fig.2: Baseline results obtained at 1500 rpm, 3 bar IMEP

Parameters	Values
Injection pressure	700 bar
Boost Pressure	1.1 bar
Swirl Ratio	1.0
EGR rate	0 %
Intake Temp.	53 °C
NOx [ppm]	631
Soot [FSN]	0.027
HC [ppm-C3]	68.7
CO [%]	0.1
MPRR [bar/deg]	8.1
ISFC [g/kW-hr]	259.5

Table 3: Baseline characteristics

As a next step, the effect of advanced injection timing was investigated. Injection timing was varied from 2° BTDC to 34° BTDC. All other control parameters are the same as that shown in Table 3. Emissions results are presented in Fig.3.

Beyond 32° BTDC combustion characteristics deteriorated suddenly because spray plume reaches outside of the piston bowl. Shown in Fig.3, as SOI timing was advanced, NOx, soot, HC and CO emissions show different patterns. NOx emissions initially increase due to temperature increase, but decrease after 20° BTDC. On the other hand, soot emissions maintain low until 15° BTDC, and increase rapidly up to 26° BTDC, then start to decrease beyond that timing. HC and CO emissions initially decreased but increased beyond 15° BTDC.

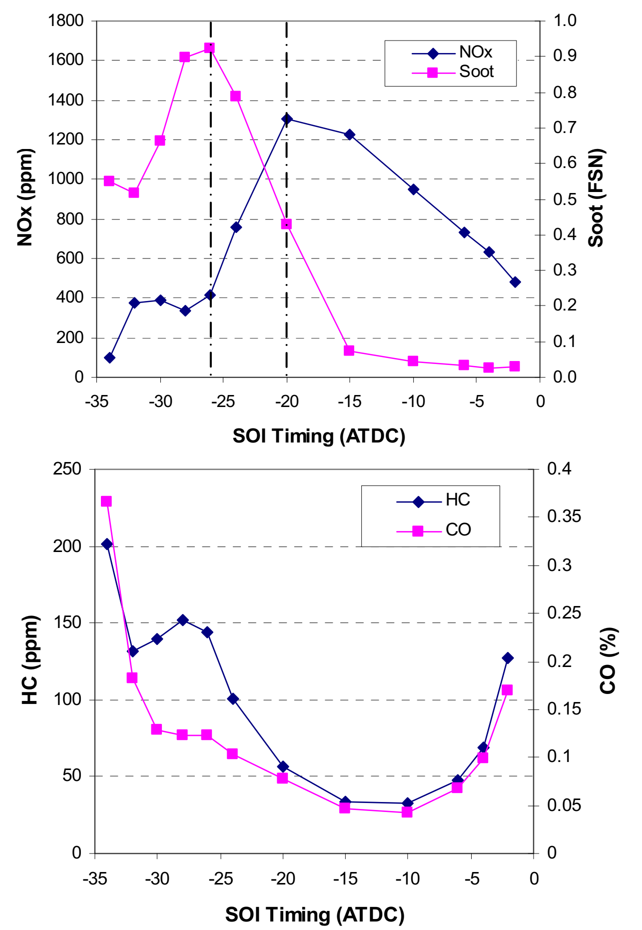


Fig. 3: The effect of advanced injection timing

To explain this characteristic, NOx and soot emissions were compared with ignition delay. In this research, ignition delay was defined as the difference between SOI timing and 10% burn time. Comparison results are shown in Fig. 4.

As SOI timing is advanced, ignition delay decreases until 10° BTDC, and after that it keeps increasing. However, when ignition delay reaches beyond a certain limit indicated with green dot-dashed line in Fig. 4, NOx and soot emissions start to decrease. Soot emissions require longer ignition delay compared to NOx emissions. Therefore, long ignition delay helps to reduce NOx and soot emissions simultaneously. Increased ignition delay provides more mixing time and results in more uniform in-cylinder air-fuel distribution (more homogeneous).

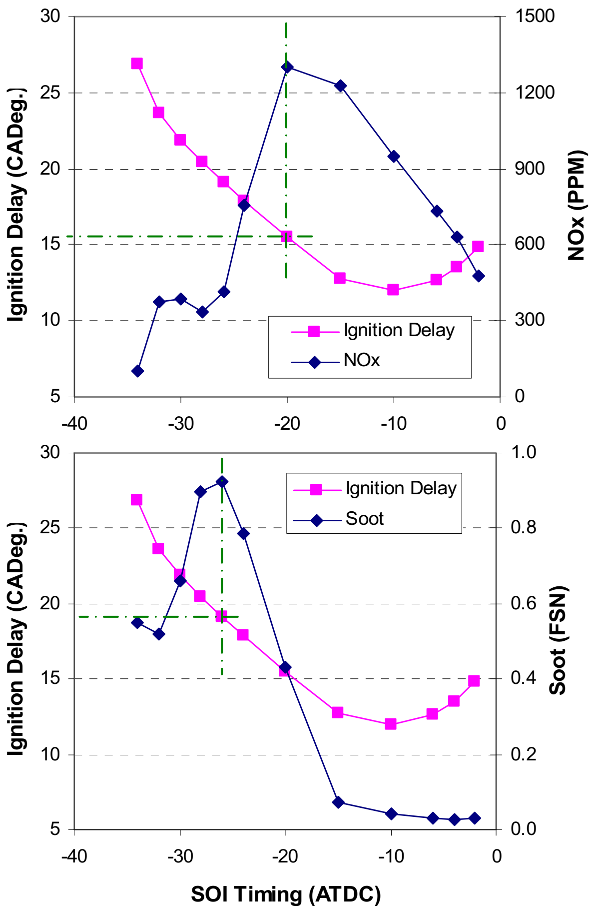


Fig.4: Comparison results of ignition delay versus NOx and soot emissions

Next, the parameters to have an influence on mixing intensity were explored. Swirl ratio was increased from 1.0 to 4.0 to examine to effect of swirl ratio. Again all other control parameters are the same as that presented in Table 3. Fig.5 shows the results of emissions and ignition delay.

As can be seen in Fig.5, ignition delay remained unchanged even though increased swirl which means total mixing time might not be changed. But huge soot emissions reduction is achieved especially at early injection timing, which indicates that enhanced mixing intensity through increased swirl ratio helps to reduce local rich regions of in-cylinder. Interestingly, other emissions such as NOx, CO emissions were not severely influenced as shown in the graph because temperature is a dominant factor for NOx emissions and wall effects are more principal one for CO emissions.

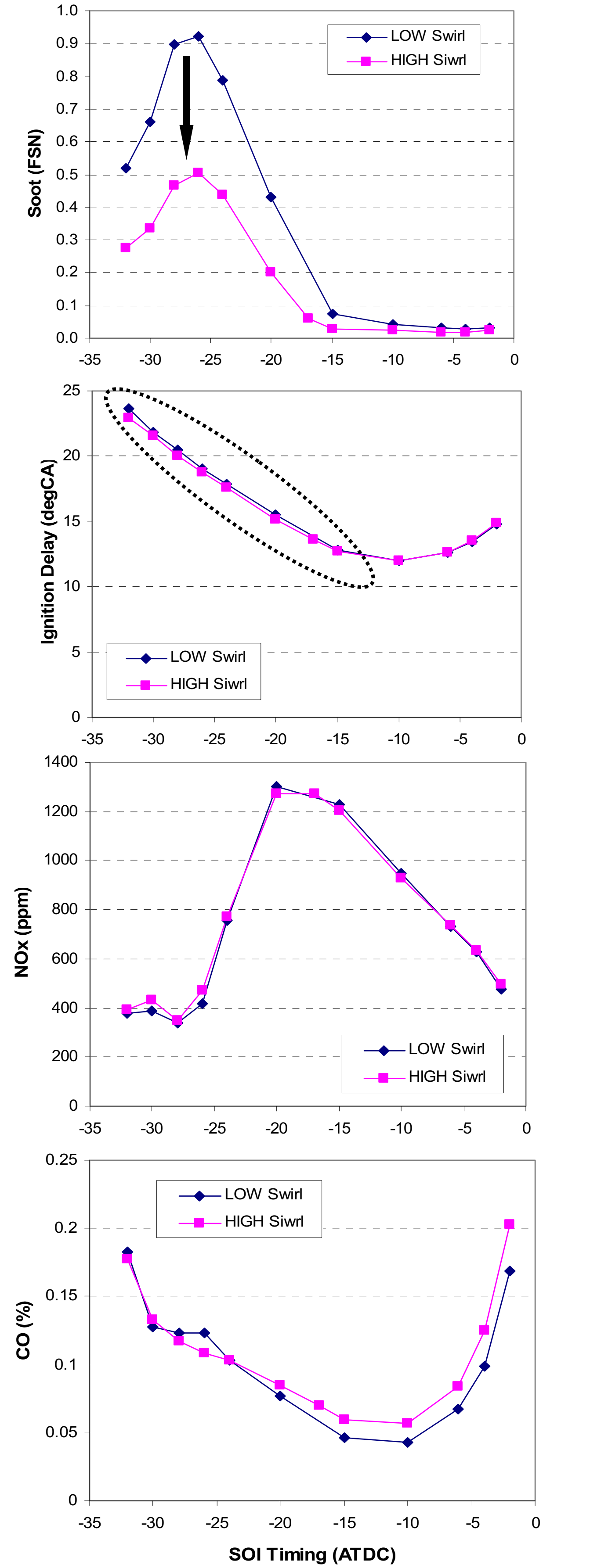


Fig. 5: The effect of swirl ratio on emissions and ignition delay

Another parameter to relate with mixing intensity is injection pressure. Swirl ratio was maintained at 4.0 and other control parameters such as boost pressure, intake air temperature were kept same as those shown in Table 3. To investigate the effect of injection pressure, three different injection pressures (700 bar, 1100 bar, 1500 bar) were tested and the results are shown in Fig.6.

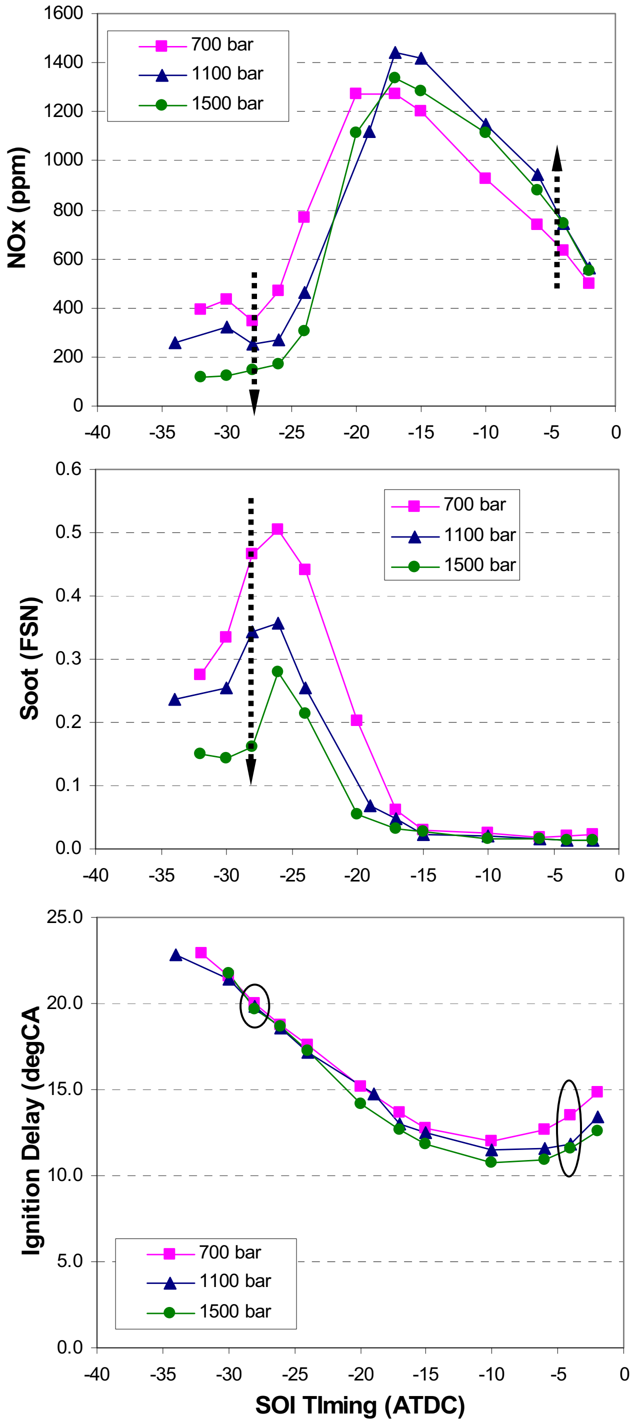


Fig.6: The effect of injection pressures on combustion and emissions.

For the case of near-TDC injection, as injection pressure increased, ignition delay was decreased and NOx emissions were increased, which is typical behavior of conventional diesel combustion. There is little effect on soot emissions. On the other hand, for the case of early injection, even though ignition delay remained unchanged regardless of injection pressure, NOx and soot emissions were decreased simultaneously. Therefore, it is verified that increased mixing time as well as increased mixing intensity is very effective to reduce soot emissions.

Through the investigation on the effect of injection pressure, it is found that not only soot emissions but also NOx emissions can be reduced. To understand the reason for the reduction in NOx emissions at early injection with increasing injection pressure, combustion characteristics were compared and are shown in Fig.7 and Fig.8.

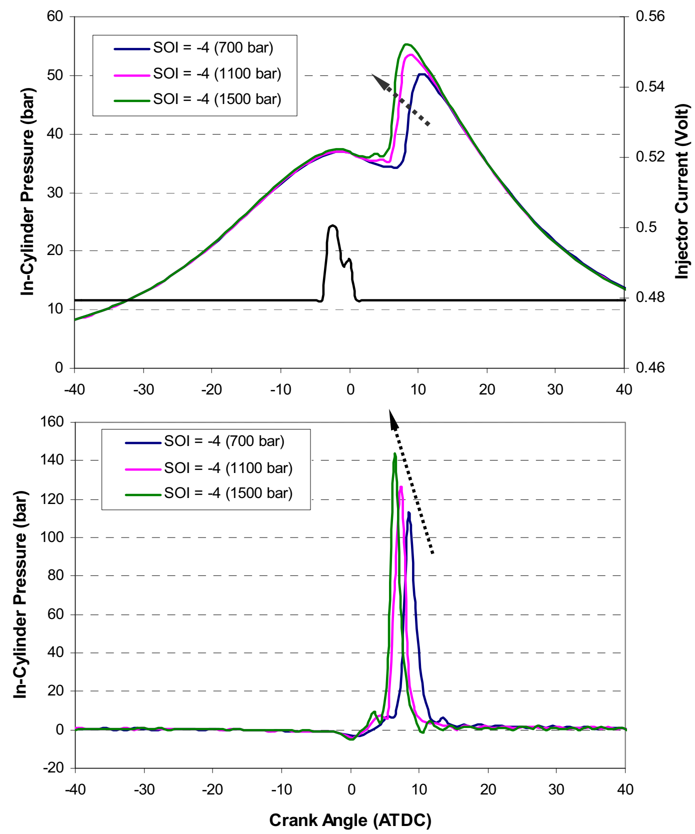


Fig.7: Combustion characteristics obtained at near-TDC injection (SOI = 4° BTDC)

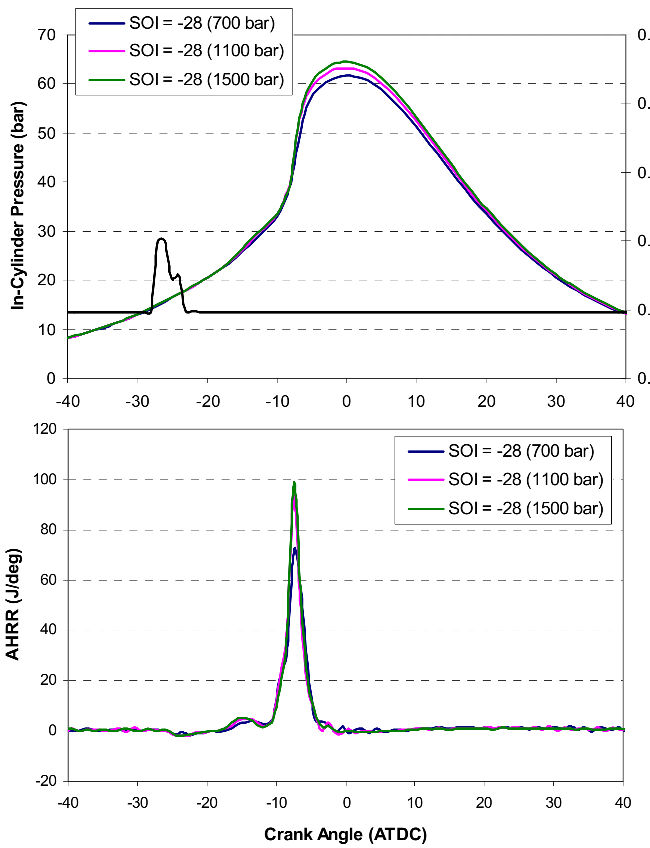


Fig.8: Combustion characteristics obtained at early injection (SOI = 28° BTDC)

For the case of near-TDC injection, as injection pressure increases, combustion phasing was advanced, ignition delay was decreased, and peak of heat release rate was increased. That's why higher NOx with increasing injection pressures was obtained (see Fig. 7).

On the contrary, for the case of early injection, even though injection pressure increases, combustion phasing and ignition delay remained unchanged. However, due to the mixing intensity enhancement, overall leaner air–fuel distribution inside cylinder can be achieved at higher injection pressures, so lower NOx emissions can be obtained (see Fig. 8).

Note that increase mixing intensity by swirl ratio change does not influence NOx emissions, but additional enhancement of mixing intensity through injection pressure increase leads to reduction in NOx emissions. Therefore, in order to reduce NOx emissions by increasing mixing intensity, much higher mixing intensity is required compared to that needed to decrease soot emissions. As mentioned earlier, mixing time point of view, mixing time required to reduce soot emissions is longer than that needed to decrease NOx emissions. Both longer mixing time and enhanced mixing intensity are crucial parameters to reduce NOx and soot emissions simultaneously.

Another important factor to influence NOx and soot emissions is temperature. It is well known EGR is a key parameter to control combustion temperature. To reduce in-cylinder combustion temperature, cooled EGR was employed at 1500 bar of injection pressure, 28° BTDC of injection timing, 4.0 of swirl ratio. Fig.9 shows the pressure trace, injector current, and apparent heat release rate obtained and calculated at different EGR level. Fuel consumption, ignition delay and maximum pressure rise rate (MPRR) are presented in Fig.10. Emissions results are plotted in Fig.11.

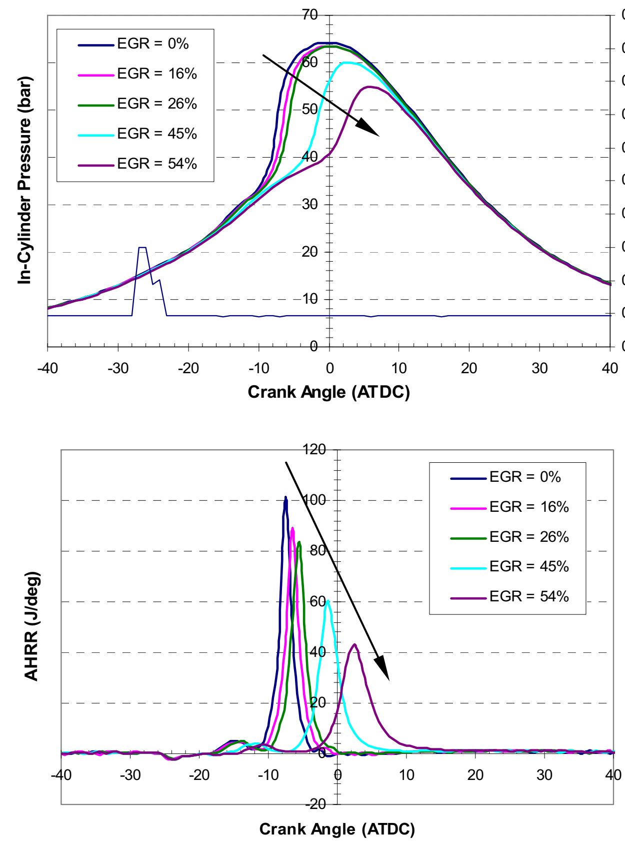


Fig.9: The effect of cooled EGR on combustion characteristics

As EGR rate increases, combustion phasing is retarded and more gradual heat release (more smooth combustion) is obtained. Therefore, ignition delay is further extended and pressure rise rate is decreased (lower combustion noise). Due to the extended ignition delay and high cooled EGR rate, more premixed in-cylinder condition and lower combustion temperature were achieved, which enable to reduce NOx and soot emissions at the same time (see Fig. 11).

HC and CO emissions are increased as EGR rate increases because of decreased oxidation rates. But compared to typical HCCI combustion [1, 2], HC and CO emission levels are still acceptable ranges.

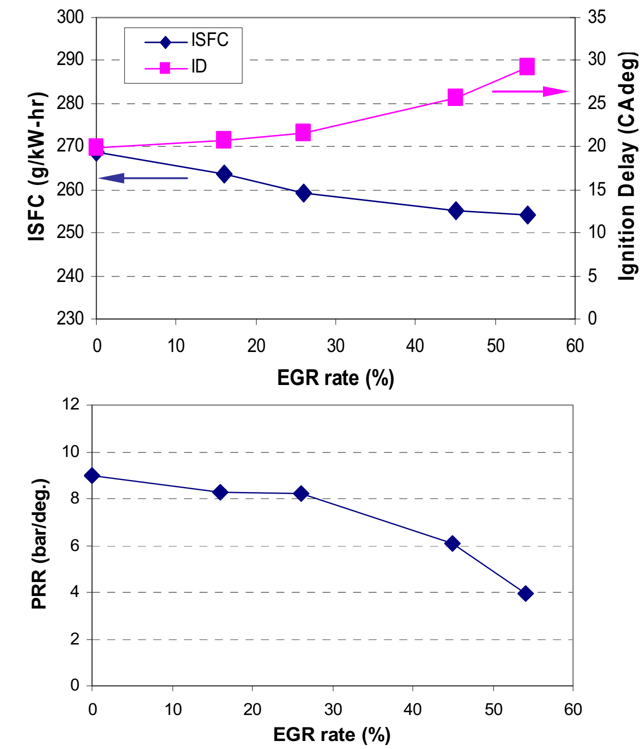


Fig.10: The effect of cooled EGR on fuel consumption, ignition delay and pressure rise rate

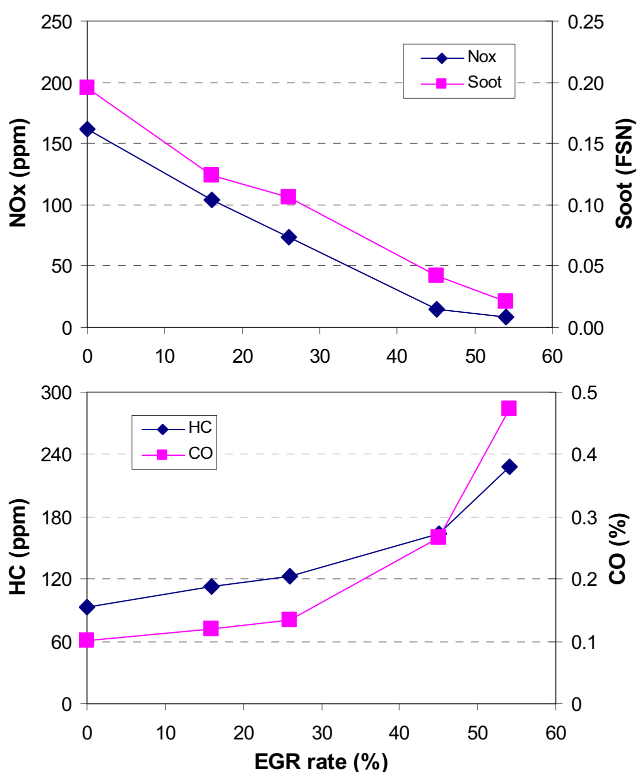


Fig.11: The effect of cooled EGR on emissions

The comparison results of baseline (conventional diesel combustion) and premixed low-temperature diesel combustion are summarized in Table 4. Compared to the baseline which shows the characteristics of conventional diesel combustion, ignition delay is extended by greater than factor of two. Also, overall A/F ratio becomes rich because of EGR replacing fresh air. Near-zero NOx and soot emissions were obtained with improved fuel efficiency. Lower combustion noise was expected as well.

	Conv.	PLTDC	Gain / Loss
NOx [ppm]	631	9	- 99 %
Soot [FSN]	0.027	0.021	- 20 %
HC [ppm-C3]	68.7	228.7	~ + 3 times
CO [%]	0.10	0.47	~ + 5 times
MPRR [bar/deg]	8.10	3.97	- 50 %
Ignition Delay [CA]	13.5	29.2	> + 2 times
A/F Ratio	52.0	24.9	
ISFC [g/kW-hr]	259.5	254.1	- 2 %

Table 4: Comparison results between baseline (conventional) and PLTDC

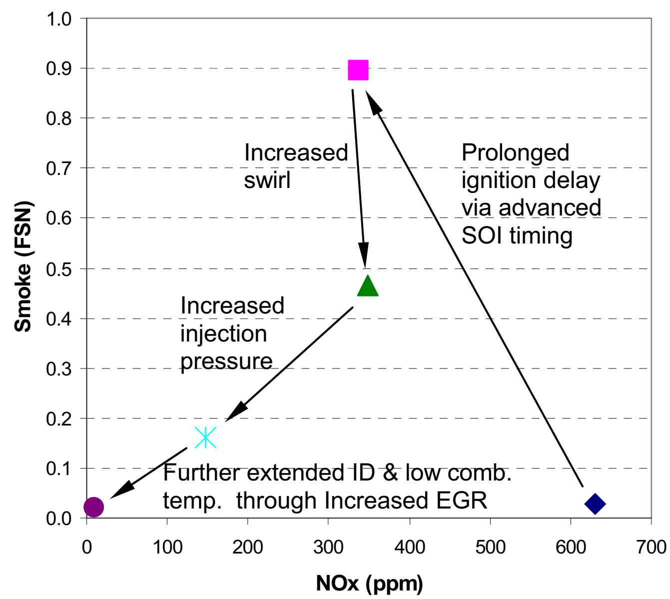


Fig.12: The path to reach premixed low-temperature diesel combustion at 1500 rpm, 3 bar IMEP condition

Fig.12 summarizes a path how to achieve premixed low-temperature diesel combustion. Starting with the baseline, first, ignition delay was prolonged via advanced injection timing. Due to lean effect by longer mixing time, NOx emissions decreased but soot emissions increased. In order to reduce soot emissions while maintaining NOx emissions level, swirl ratio and injection pressure were increased. NOx and soot emissions were decreased simultaneously due to enhanced mixing intensity. Finally, ignition delay was further extended and combustion temperature was reduced through increased cooled EGR rate. In a conclusion, mixing enhancement (mixing time and mixing intensity) and low combustion temperature are the main keys to reduce NOx and soot emissions simultaneously.

1500 rpm, 6 bar IMEP - In this section, the results obtained at 1500 rpm, 6 bar IMEP condition are presented. As for 3 bar IMEP condition, after performing injection timing sweep test around TDC at the condition shown in Table 5, best efficiency point was chosen as a baseline. Fig.2 and Table 3 show the results of baseline. Fig.14 and Table 5 show the results of the baseline. The baseline shows the features of a typical diesel engine with high NOx and soot, and low CO and HC emissions. Compared to 3 bar IMEP condition, NOx increased by factor of two, soot increased by factor of ten, and very high maximum pressure rise rate (MPRR) was measured.

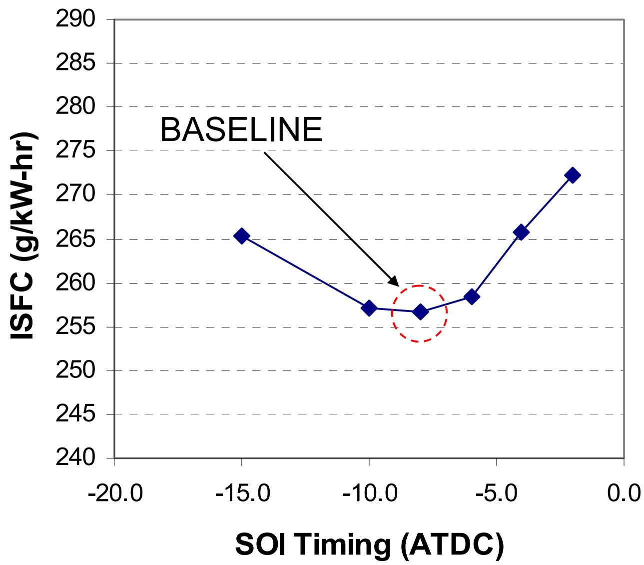
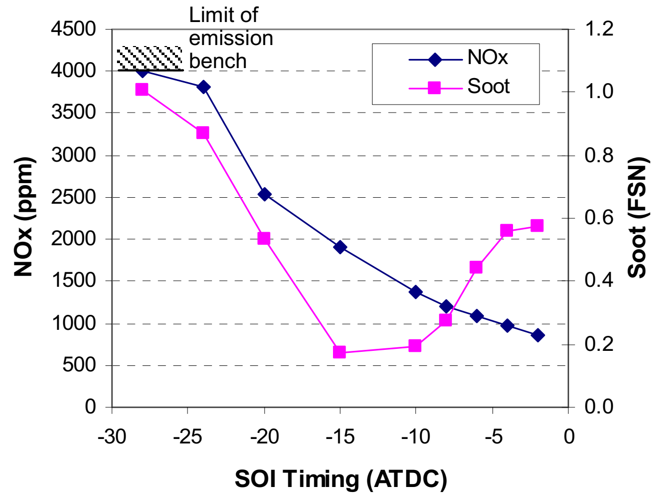


Fig.14: Baseline results obtained at 1500 rpm, 6 bar IMEP

Parameters	Values
Injection pressure	700 bar
Boost Pressure	1.1 bar
Swirl Ratio	1.0
EGR rate	0 %
Intake Temp.	55 °C
NOx [ppm]	1208
Soot [FSN]	0.275
HC [ppm-C3]	37.1
CO [%]	0.17
MPRR [bar/deg]	19.4
ISFC [g/kW-hr]	256.7

Table 5: Baseline characteristics

Again, the effect of early injection timing was investigated. The results are shown in Fig.15. As injection timing is advanced, NOx and soot emissions keep increasing. Ignition delay shows similar pattern to that of 3 bar IMEP condition, but it is less than that of 3 bar case. Overall, there is no advantage of advanced injection timing in terms of NOx and soot emissions. Since total fueling rate was doubled to achieve desired load condition, more mixing time is expected to obtain the premixed combustion condition. However, ignition delay is decreased instead, so worse NOx and soot emissions were obtained.



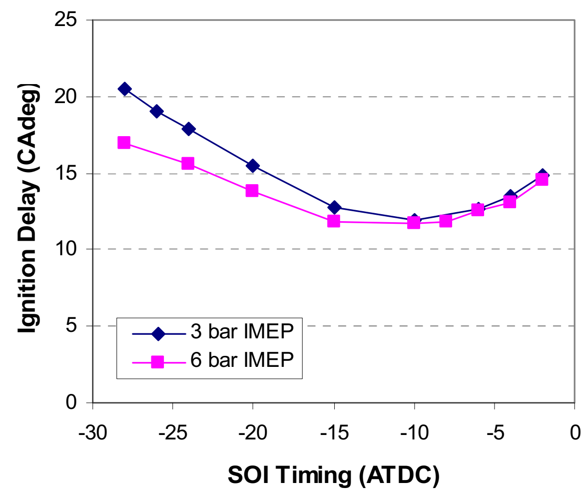


Fig.15: The effect of injection timing on Emissions and ignition delay

Enough mixing time could not be obtained by early injection. As a next step, to boost up the mixing rate, swirl ratio was increased. NOx and soot emissions results are summarized in Fig.16. As swirl increases, NOx and soot emissions show a typical trade-off pattern, which is increased NOx and decreased soot emissions. This means overall mixing rate (time and intensity) is still not enough to influence engine-out emissions even though the application of advanced injection timing and high swirl ratio.

Next, EGR was employed at high injection pressure of 1500 bar to reduce NOx and soot emissions at the same time like 3 bar IMEP condition.

Fig. 17 shows the results of NOx and soot emissions measurement as a function of EGR. Low NOx emissions can be achieved, but soot emissions were increased exponentially. As EGR increases, decrease in in-cylinder temperature leads to NOx emissions reduction, but that temperature is not low enough to suppress soot generation due to increased chemical energy released from the fuel. Also deficit of oxygen concentration leads to increase in soot emissions.

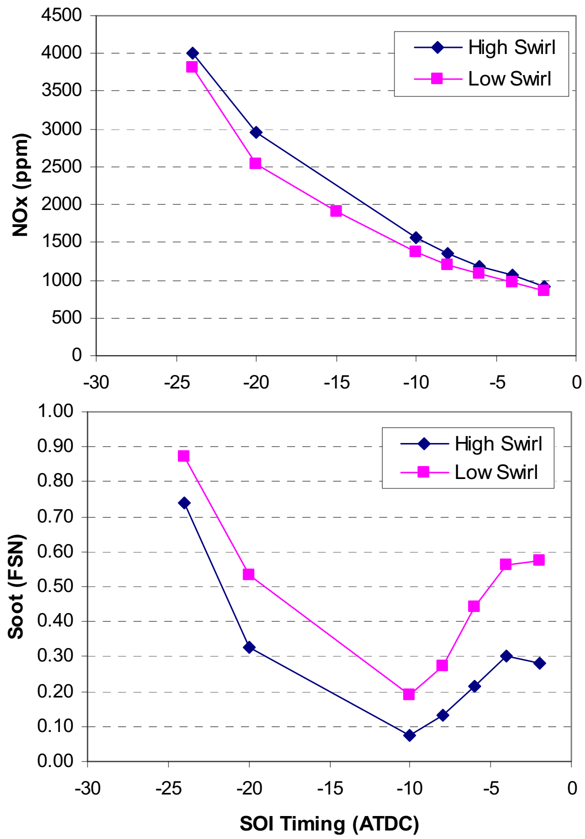


Fig.16: The effect of swirl ratio on emissions

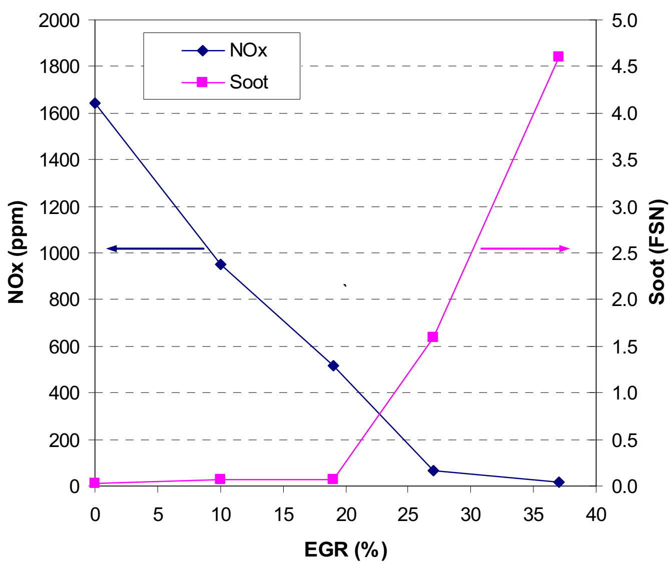


Fig.17: The effect of EGR ratio on NOx and soot emissions at 1500 bar injection pressure

So far, there is limited benefit of single injection strategy with early injection, high swirl and high EGR, moreover still high soot emissions and combustion noise are expected. Hence split injection strategies were applied and investigated to make uniform air-fuel distribution inside the cylinder.

First, to examine the effect of injection dwell, dwell was changed from 500 μs to 3000 μs with fueling level of 1 mg/cycle for first injection. Injection dwell is defined as the time duration between SOI of the first injection and SOI of the second injection (see at lower left corner in Fig. 18). Main SOI timing was fixed at 8° BTDC. To maintain engine load condition, main injection duration was adjusted. The effect of injection dwell on emissions and maximum pressure rise rate is presented in Fig.18 and Fig.19 respectively. Typical NOx-soot trade-off was obtained along with a variation of dwell. Also, lower maximum pressure rise rate was obtained regardless of dwell compared to single injection. Overall, lower NOx and lower combustion noise can be obtained through the split injection, although soot emissions were increased compared to single injection.

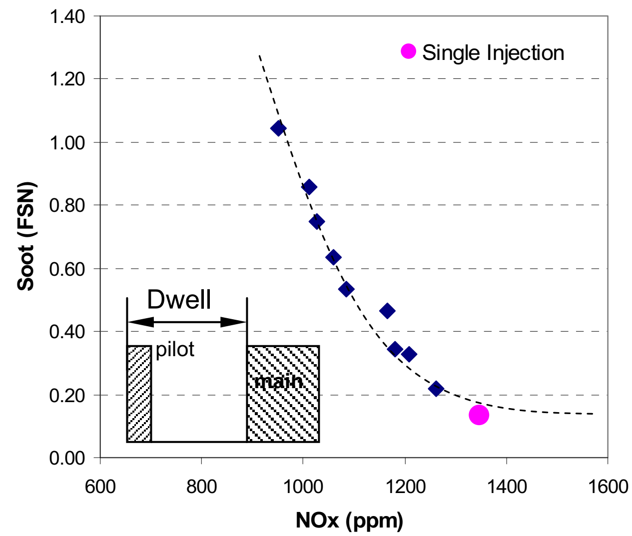


Fig.18: The effect of dwell on emissions with 1mg/cycle first injection

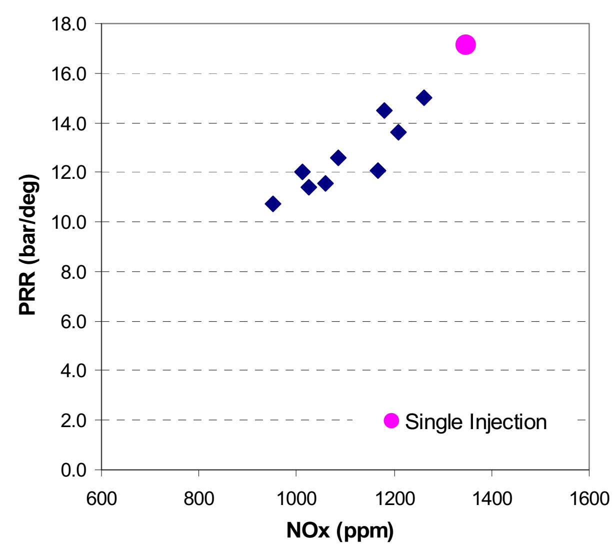


Fig.19: The effect of dwell on maximum pressure rise rate

Next, the effect of the amount of the first injection was investigated. The amount of the first injection was varied from 0.5 mg/inj to 10 mg/inj with injection dwell of 2200 µs. Main SOI timing was 8° BTDC. Again main injection duration was varied to maintain engine load condition. The effect of the amount of the first injection is shown in Fig.20 and Fig.21 respectively.

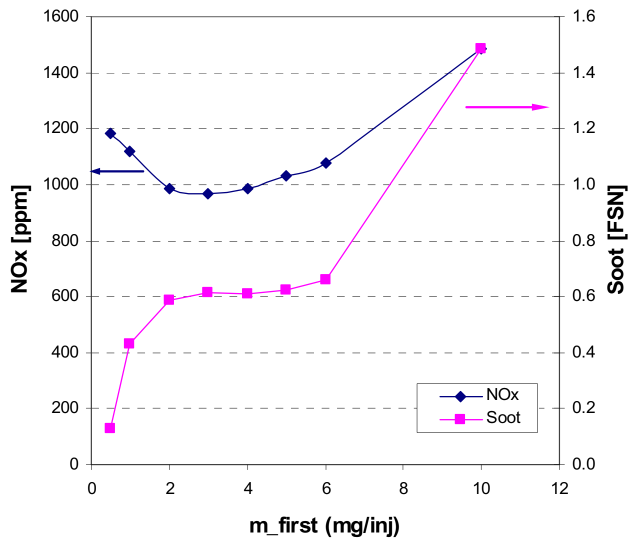


Fig.20: The effect of the amount of the first injection on emissions with fixed dwell = 2200 µs

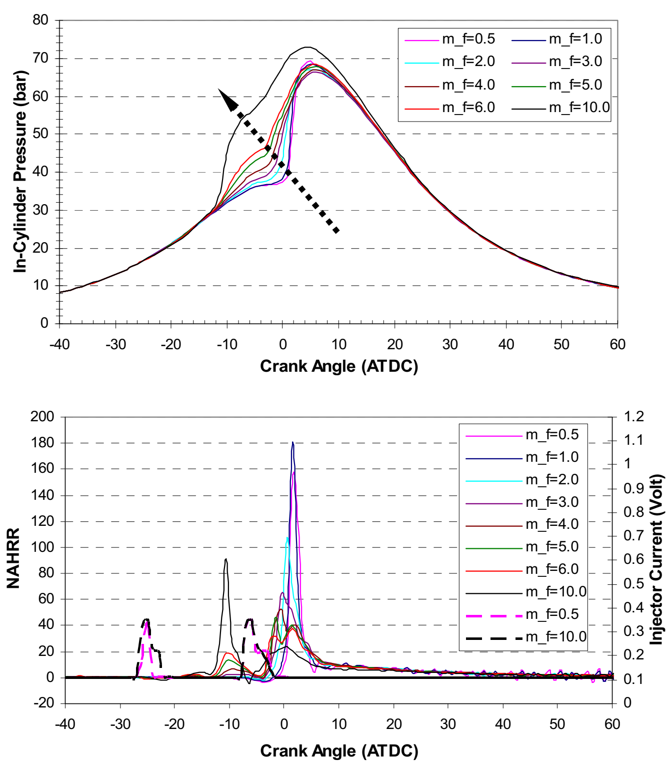


Fig.21: The effect of the amount of the first injection on combustion characteristics

As can be seen in Fig.20, there is an optimum value to give the lowest NOx emissions. But soot emissions increased with increasing the amount of the first injection. As for 10 mg/inj, rich premixed and high temperature combustion is initiated even for the first injection, and additional combustion occurs after second injection is employed. Because of limited mixing time between two pulses, high NOx and high soot emissions were obtained (see Fig.21). Large amount of first injected fuel may produce local rich regions and prevent uniform air-fuel distribution.

To enhance mixing characteristics, injection pressure was increased from 700 bar to 1500 bar. Fig.22 shows the results obtained from the amount of the first injection swing test with four different injection pressures. NOx and soot emissions show similar overall pattern regardless of injection pressures. As injection pressure increases, soot emissions decrease but NOx emissions increase. The amount of the first injection to give lowest NOx and soot emissions is varied with different injection pressures. It was determined that there is an optimum combination of dwell and the amount of the first injection to achieve lowest NOx and soot emissions, moreover these combination of optimum will vary with other operating parameters such as EGR rate, injection pressure, swirl ratio, boost pressure. The reason is that these operating parameters have a direct impact on mixing characteristics inside the cylinder.

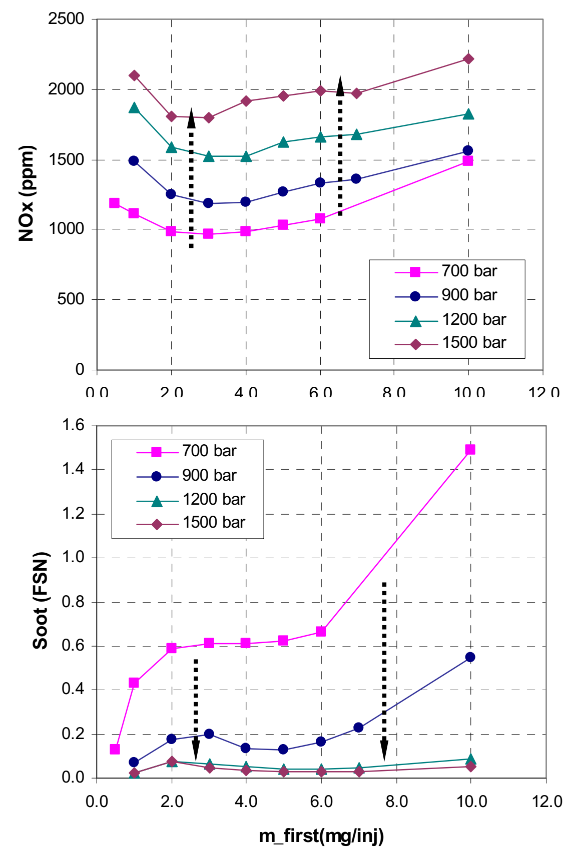


Fig.22: The effect of the amount of the first injection on emissions with four different injection pressures

To find out optimum dwell and the amount of first injection at high EGR condition, cooled EGR was employed, and to expand the limit of EGR level, boost pressure was increased. The effect of the amount of first injection was investigated again at different settings of EGR and dwell. The results are shown in Fig. 23.

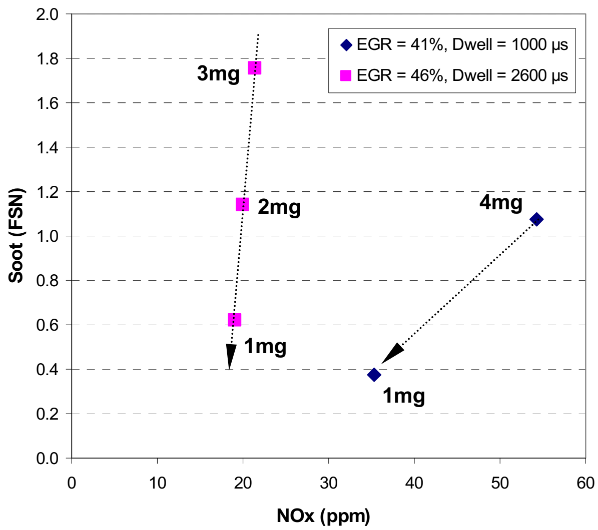


Fig.23: The effect of the amount of the first injection on emissions at two different operating conditions

The amount of first injection produced different results at these test conditions. According to Fig. 22, for the condition of no EGR, between 2 mg/inj and 4 mg/inj was an optimum value to achieve the best NOx and soot trade-off characteristics. However, as shown in Fig.23, with high EGR condition, the lowest amount of the first injection gives the best NOx and soot emissions. As the amount of the first injection decreases, NOx and soot emissions decrease simultaneously regardless of EGR level and dwell between injections. So 1 mg/inj of the amount of first injection was selected as an optimum. Then dwell was changed from 500 µs to 3000 µs at 47% EGR and 1.5 bar boost pressure. The results are presented in Fig.24. Different NOx and soot emission pattern was obtained compared to Fig.18. Dwell has a greater effect on soot emissions than on NOx emissions. Dwell of 1500 µs was chosen as an optimum.

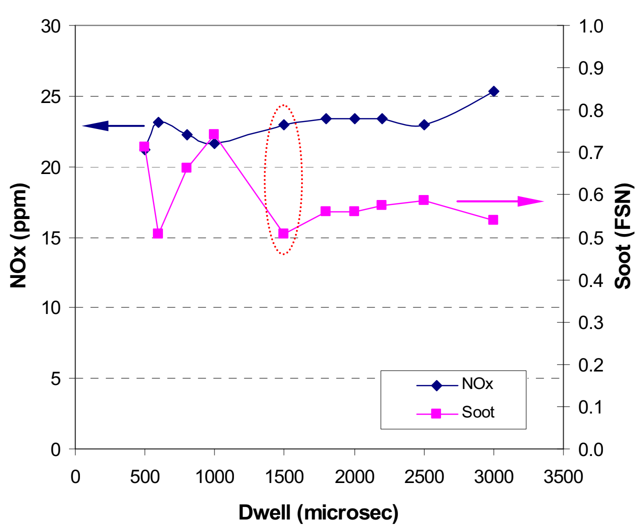


Fig.24: The effect of dwell on emissions at 1 mg/inj of first injection, 1500 bar injection pressure, 1.5 bar boost pressure, 4.0 swirl ratio, and 47% EGR condition

It is well known that the post injection has an advantage in reducing soot emissions [19, 20, 21]. To reduce soot emissions further, the effect of post injection strategies were explored. The amount of post injection was fixed with 1.0 mg/inj and the dwell was varied from 500 µs to 3000 µs. Post injection dwell is defined as the time duration between EOI of the main and SOI of the post (see right side of the plot in Fig. 25).

Fig.25 and Fig.26 show comparison results with and without post injection. Dotted line indicates the results obtained without post injection. Note that there is an optimum dwell to generate minimum soot emissions. Small amount of fuel injection during post injection enhances the mixing rate by adding secondary air movement. In addition, optimum dwell timing should be determined to prevent local rich combustion region inside cylinder caused by colliding with previously injected fuel. Also this effect improves the efficiency while maintaining NOx emissions level as shown in Fig.26.

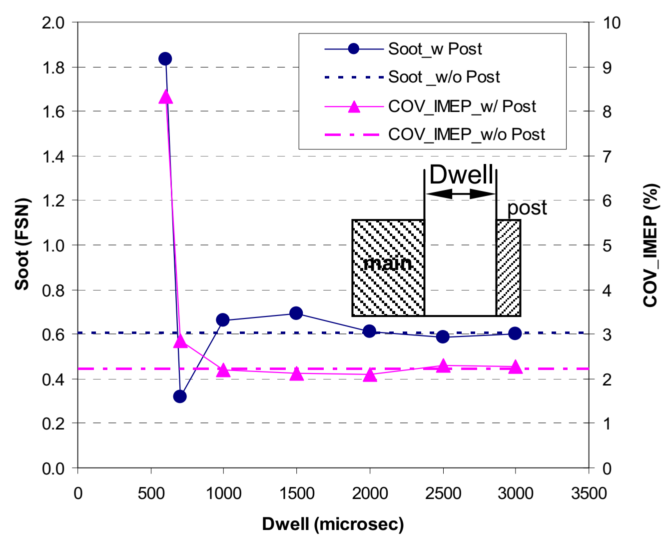


Fig.25: The effect of dwell between main and post injection on soot emissions and combustion stability

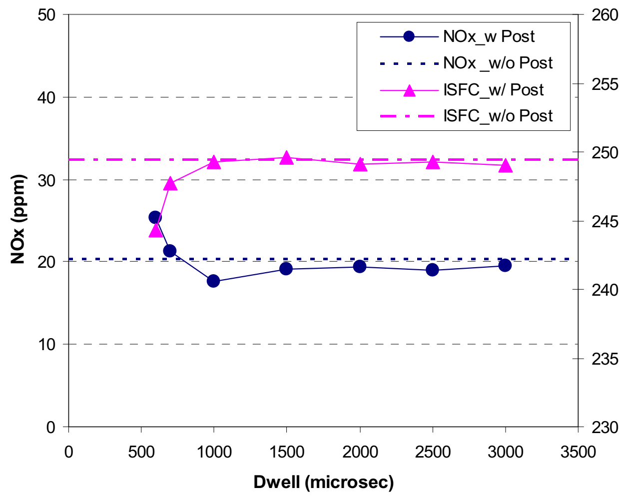


Fig.26: The effect of dwell between main and post injection on NOx emissions and fuel consumption

Combustion characteristics of PLTDC at 6 bar IMEP are shown in Fig. 27. Detailed comparison results between baseline for conventional combustion and PLTDC are shown in Table 6. Compared to baseline, huge reduction in NOx emissions and marginal decrease in soot emissions were realized with fuel economy improvement. Based on reduction of MPRR, lower combustion noise is also expected. Uniform air-fuel mixture distribution inside cylinder through splitting injection pulses and mixing enhancement and low combustion temperature via high EGR rate are the main keys to obtain premixed low-temperature diesel combustion.

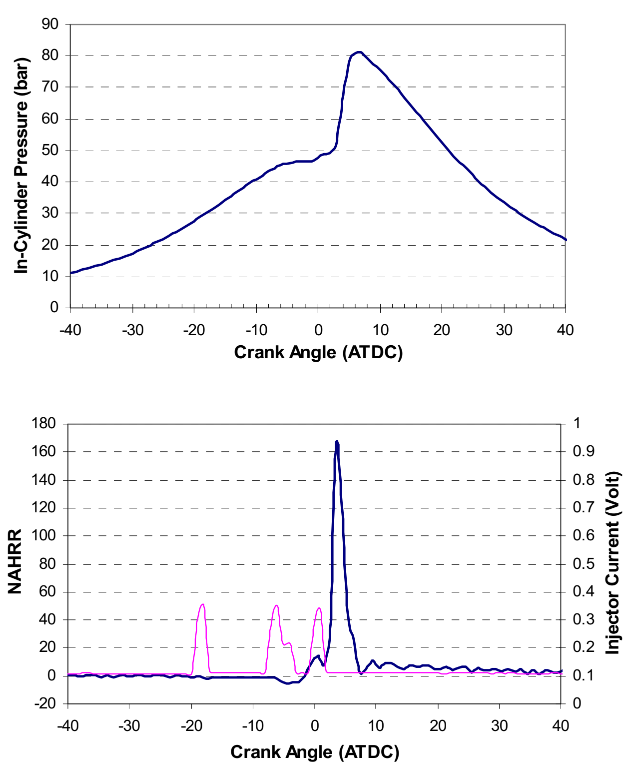


Fig.27: Combustion characteristics of premixed low-temperature diesel combustion at 6 bar IMEP.

	Conv.	PLTDC	Gain / Loss
NOx [ppm]	1208	25	- 98 %
Soot [FSN]	0.275	0.224	- 19 %
HC [ppm-C3]	37.1	67.7	+ 83 %
CO [%]	0.17	0.27	+ 57 %
MPRR [bar/deg]	19.4	15.0	- 23 %
A/F Ratio	26.5	19.9	
ISFC [g/kW-hr]	256.7	246.3	- 4 %

Table 6: Comparison results between baseline (conventional combustion) and PLTDC

Fig.28 shows the emissions reduction path. In order to achieve PLTDC, starting baseline, which shows the conventional combustion feature, seven steps were adopted. Swirl ratio was increased, double injection strategy was applied, injection pressure was increased, EGR rate was employed, boost pressure was increased, double injection strategy was re-optimized, and finally post injection strategy was applied.

But, this emissions reduction path may vary depending on the target such as raw emissions, efficiency, and after-treatment requirements. For example, for a vehicle equipped with DPF system, engine-out soot emissions can be sacrificed to obtain minimum NOx emissions or to improve efficiency.

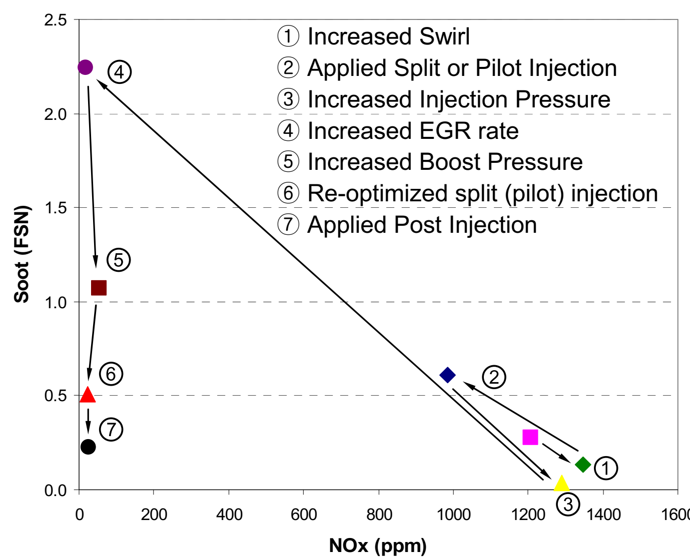


Fig.28: Emissions reduction path at 1500 rpm, 6 abr IMEP condition

CONCLUSION

The pursuit of new combustion concepts is ongoing to meet future emissions regulations and to reduce the burden on aftertreatment systems. Premixed Low Temperature Diesel Combustion (PLTDC) was developed using single-cylinder engine to achieve low NOx and soot emissions while maintaining good fuel efficiency. Operating conditions considered were 1500 rpm, 3 bar and 6 bar IMEP. The effects of injection timing, injection pressure, swirl ratio, EGR rate, and multiple injection strategies on the combustion process have been investigated.

The results are as follows.

1. Low NOx and soot emissions can be realized up to 6 bar IMEP while maintaining good fuel efficiency.

2. Single injection strategy was developed to 3 bar IMEP condition; triple injection strategy was developed at 6 bar IMEP condition.

- Small preinjection of 1 mg/cycle at 6 bar IMEP and high EGR produced very low NOx and soot emissions, and reduced combustion noise.

- Additional small post injection of 1 mg/cycle at high EGR produced additional soot reductions. Injection dwell of about 700 μs was needed at this condition.

3. Uniform A/F ratio distribution inside cylinder (homogeneous condition) by enhancing mixing rate and lengthening ignition delay and adopting multiple injection strategies is one of main keys to achieve premixed diesel combustion.

4. Low combustion temperature through high EGR is another important factor to obtain low NOx and soot combustion.

5. Emissions reduction path may vary depending on the target such as raw emissions, efficiency, and after-treatment requirements. For example, for a vehicle equipped with DPF system, engine-out soot emissions can be sacrificed to obtain minimum NOx emissions or to improve efficiency.

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ACRONYMS, ABBREVIATIONS

BSFC: Brake Specific Fuel Consumption

CAD: Crank Angle Degree

HCCI: Homogeneous Charge Compression Ignition

LTC: Low Temperature Combustion

NADI: Narrow Angle Direct Injection

PCI: Premixed Compression Ignition

PCCI: Premixed Charge Compression Ignition

PLTDC: Premixed Low-Temperature Diesel Combustion

TDC: Top Dead Center

SOF: Soluble Organic Fraction

SOI: Start Of Injection

VVA: Variable Valve Actuation

LPP: Location of Peak Pressure