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Combustion and Emission Characteristics of a Premixed Charge Compression Ignition Engine

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Abstract The combustion and emission characteristics of a low emission and highly efficient premixed charge compression ignition diesel engine operated with moderate amount of exhaust gas recirculation was investigated in a single cylinder test engine. Tests were carried out under constant speed and fuel quantity with two injection pressures 80 and 140 MPa while the injection timing was varied. Exhaust emissions and in-cylinder pressure were measured for the all the experimental conditions considered. Analysis based on the engine performance and exhaust emissions were carried out. The introduction of exhaust gas recirculation (EGR) under higher injection pressure led to the simultaneous reduction of nitrogen oxides (NOx) and soot emissions due to a lower combustion temperature compared to conventional diesel combustion. However, the emissions of hydrocarbon and carbon monoxide slightly increased due to insufficient oxygen to allow complete combustion. Higher boost pressure led to higher indicated thermal efficiency and indicated mean effective pressure (IMEP) with low emissions of NOx and soot with relatively same CO but slightly higher HC emissions. High engine performance and lower emissions was achieved at the high injection pressure coupled with EGR rate and high boost pressure.

Keywords Exhaust gas recirculation (EGR), premixed charge compression ignition, performance, specific emissions.

1. Introduction

For the past few decades, diesel engines have gained wider usage in transportation and power generation applications due to their durability, reliability and high efficiency [1]. Despite this, diesel engines have been noted to contribute to the environmental pollution because of its higher production of nitrogen oxide (NOx) and soot emissions.

A lot of effort have been put forth to curb this emissions, especially NOx and soot due to their effect on the environment and human health. On the other hand the emission regulation of both emissions is becoming increasingly stringent in the developed and the developing countries.

To meet this stringent regulation advanced diesel combustion strategies and exhaust emission after-treatment systems are required. Emissions after-treatment devices however are costly and this implies that more efforts on advanced diesel combustion strategies will gain more interest than ever in future [2].

PCCI combustion is realized by early and moderately early injection coupled with the use of EGR. Simultaneous reduction of soot and NOx can be achieved by increasing the EGR rate and retarding the injection timing. This also is advantageous in retarding the combustion phasing leading to less negative work hence high IMEP and indicated thermal efficiency.

Many approaches have been proposed to reduce the emissions including both in-cylinder control and after treatment. In-cylinder control methods encompasses but not limited to exhaust gas recirculation EGR, air management and flexible fuel injection schemes which have been seen as potential solutions to advanced combustion strategies.

In PCCI engine the desired ignition delay is achieved through a lower compression ratio, enhanced charge motion, higher injection pressure, and relatively large amounts of cooled external exhaust gas recirculation (EGR) [3].

PCCI is not fully homogeneous like HCCI, but it makes use of injection timing and EGR to greatly increase the controllability of combustion phasing and the rate of combustion. Much research has been conducted to expand the high-load limits of PCCI using fuel properties [4], increase boost pressure [5] and split injection [6].

PCCI combustion strategies employing moderately early injection (approx. 25° BTDC) have been widely investigated because they are advantageous in avoiding lubricant dilution. Under such cases, high levels of EGR coupled with low compression ratios are used to ensure sufficient air–fuel mixing time, leading to suppression of NOx formation and better combustion phasing. Thus, to achieve low soot and NOx simultaneously under moderately early injection timing it is necessary to optimize the injection pressure, intake pressure, EGR rate and injection timing[7], [8]. Like HCCI, PCCI is prone to high HC and CO emission and a high pressure-rise rate, which results in high combustion noise; additionally, it has limitation while considering higher engine loads.

To clearly understand the combustion mechanism inside the combustion chamber in PCCI combustion mode, it is necessary to study the effect of injection pressure, intake pressure and EGR rate in the mixing of the air–fuel mixture, the heat-release process, and the formation of soot, NOx, and other combustion products.

Available data are still insufficient to fully understand the relationship among the intake pressure, mixture formation, and the combustion process in PCCI combustion mode.

The present study sought to understand the combustion characteristics and emissions formation of a low emission and highly efficient PCCI engine. A compression ratio of 13 with two fuel injection pressures coupled with a slightly narrowed cone angle of 140° was chosen to avoid cylinder-wall wetting. The injection pressure, intake pressure and EGR were varied to establish the best formula below:

$$EGRrate = \frac{N_2}{Air + N_2} \qquad (1)$$

operating conditions to achieve simultaneously low NOx and soot emission with a higher indicated thermal efficiency. The fuel injection quantity considered for thus work was kept constant

2. Experimental set-up and procedure

2.1. Test engine

A four-stroke, single-cylinder, direct-injected supercharged diesel engine with a displacement of 781.7 cm^3 was used for the investigation. Table 1 shows the engine specifications and the operating conditions. The low-sulphur, JIS #2 diesel fuels used in the experiment commonly available in Japan was used. A schematic diagram of the test engine is shown in .. 1. A common rail injection system capable of developing an injection pressure of 180 MPa was used. A valve-covered orifice (VCO) injector with four 0.1-mm-diameter holes placed symmetrically in the nozzle tip was used. The top dead centre (TDC) signals and every half-degree crank angle (CA) were detected by photo interrupters and coupled with a controller to control the injection timing and injection duration. In-cylinder pressure was measured with a piezoelectric pressure transducer (6052C, Kistler) coupled with a charge amplifier (5011B, Kistler). The pressure history was analysed to obtain the heat release rate to investigate the combustion characteristics. Exhaust emissions were captured using a NOx–CO analyser (Horiba, PG-240), HC analyser (Horiba, MEXA-1170HFID), and smoke meter (Horiba, MEXA-600s). The data for each engine condition were captured when the engine was in equilibrium, during which there was almost no change in the emission parameters and exhaust temperatures. For a specific engine condition, three data samples were collected and an average done.

2.2. Engine operating conditions

Under this investigation, late and moderately early injections were considered. The test engines was operated at two injection pressures, P_{inj} = 80 MPa and 140 MPa, to isolate the effect of injection pressure on the combustion and exhaust emissions, with a slightly narrow included cone angle of 140^o to avoid cylinder wall wetting. Engine speed was maintained at 1000 rpm. The intake pressure was also varied from 101 kPa to 200 kPa with a constant fuel injection quantity of 15 mg/cycle equivalent to 3.7 excess air ratio (λ) in case without EGR and 101 kPa. The operating conditions are tabulated in Table 1. An injection-timing sweep from 5 to 35^o BTDC was investigated.

Our study endeavoured to use simulated EGR based on N_2 gas dilution previously used, as indicated in the 0% and 45% EGR conditions were considered in this work.

Table 1: Engine Specifications and the operating conditions

Engine type	4-stroke, single cylinder, water cooled
Bore x Stroke	96x108 mm
Swept volume	781.7cm^3
Compression ratio	13
Combustion system	PCCI, direct injection
Combustion chamber	Derby hat
Engine speed	1000 rpm
Intake pressure	101, 150 200 kPa
Injection system	Common rail system
Fuel injection pressure,	80 MPa, 140 MPa
Fuel injection quantity,	15mg/cycle
Straight-hole injector nozzle(VCO type)	ϕ 0.1 mm x 4 holes
Included cone angle	140°
Injection timing	5 - 35°BTDC
Intake temperature	40°C
Coolant temperature	80°C
Lubricant temperature	80°C

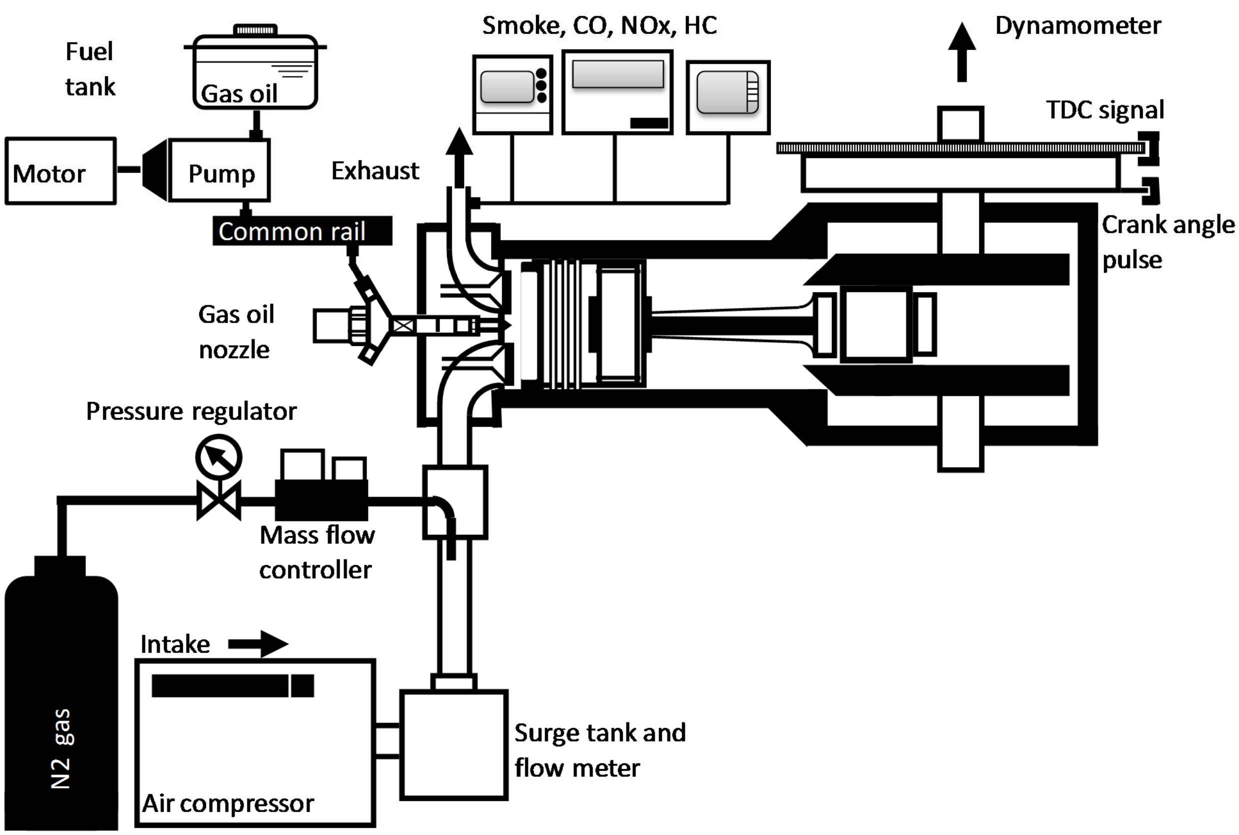
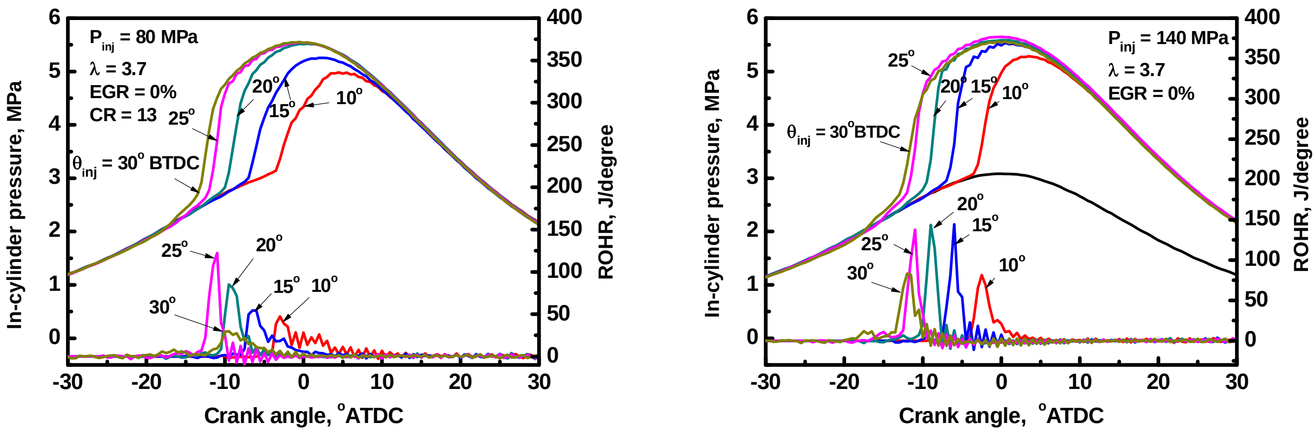


Fig.1. Experimental set up



a) P_{inj} = 80 MPa b) P_{inj} = 140 MPa

Fig.2. Pressure history and rate of heat release (ROHR) P_{inj} = 80 and 140 MPa, $\lambda = 3.7$, 0% EGR

3. Results and Discussion

3.1. Effect of injection pressure

In order to understand the effect of injection pressure on combustion and emission characteristics of PCCI engine two injection pressures were considered. Fig. 2 shows pressure history and rate of heat release for P_{inj} = 80 and 140 MPa, $\lambda = 3.7$ with 0% EGR.

To evaluate the performance and emission characteristics for the two injection pressure the emissions were measured and performance calculated from the in-cylinder pressure.

Fig. 3 shows the indicated thermal efficiency, indicated mean effective pressure (IMEP) and coefficient of variance of IMEP (COV_{IMEP}) as a function of fuel injection timing. Higher injection pressure had superior performance as compared to the lower injection pressure with advanced injection timing giving lower values for both cases. This was associated with negative power as mentioned earlier.

The COV_{IMEP} for both pressures was noted to be less than 3.5% with retarded cases having lower values.

It can be noted that in-cylinder and ROHR increases as the injection timing is advanced till an optimum point is reached then decreases thereafter. For the high injection pressure of P_{inj} = 140 MPa the optimum point is achieved at 20°BTDC. The subsequent decrease in the in-cylinder and ROHR was noted to be related to the high negative power as the injection timing was advanced.

Fig. 4 shows the effect of injection pressure on specific emissions. It was noted that the higher injection pressure led to lower emission of smoke, relatively higher NOx emission in late injection timing but lower in moderately early and early injection timing, relatively same emissions of CO and HC but with a slight increase in the early injection timing after θ_{inj} = 30°BTDC attributed to the low in-cylinder temperature, density and pressure leading to fuel impinging on the cylinder wall and hence incomplete combustion. The same phenomenon was noted in our previous paper under lower fuel quantity [9].

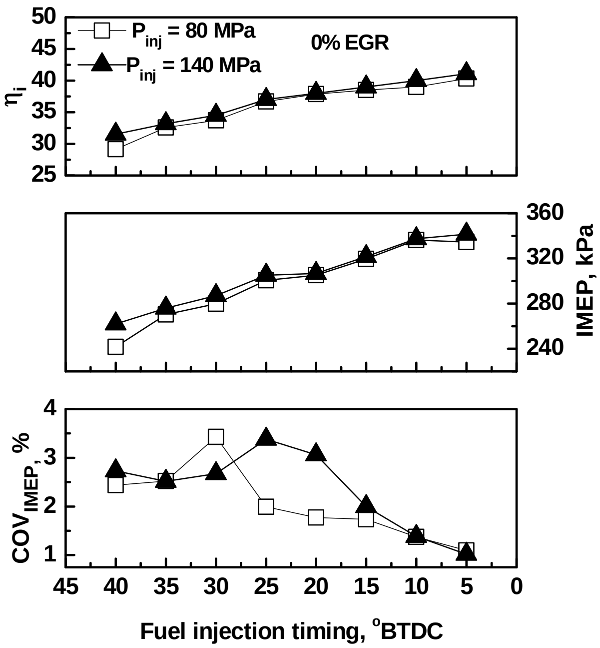


Fig.3. Effect of injection pressure on indicated thermal efficiency, IMEP and COV_{IMEP}

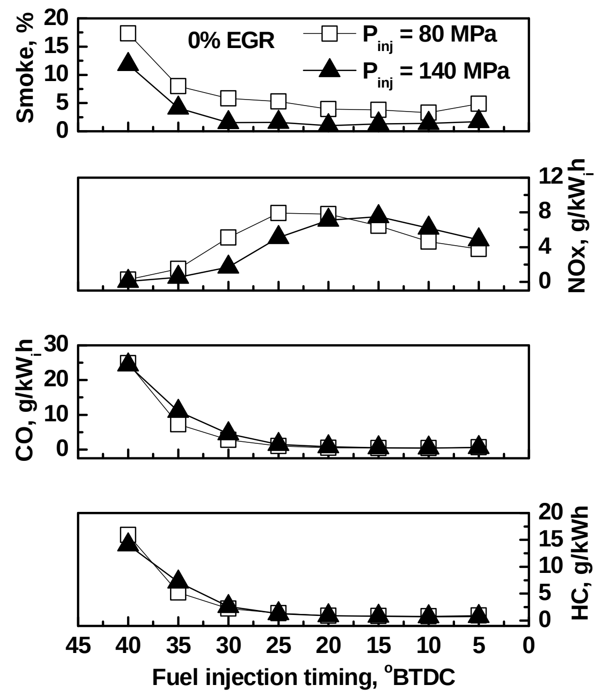
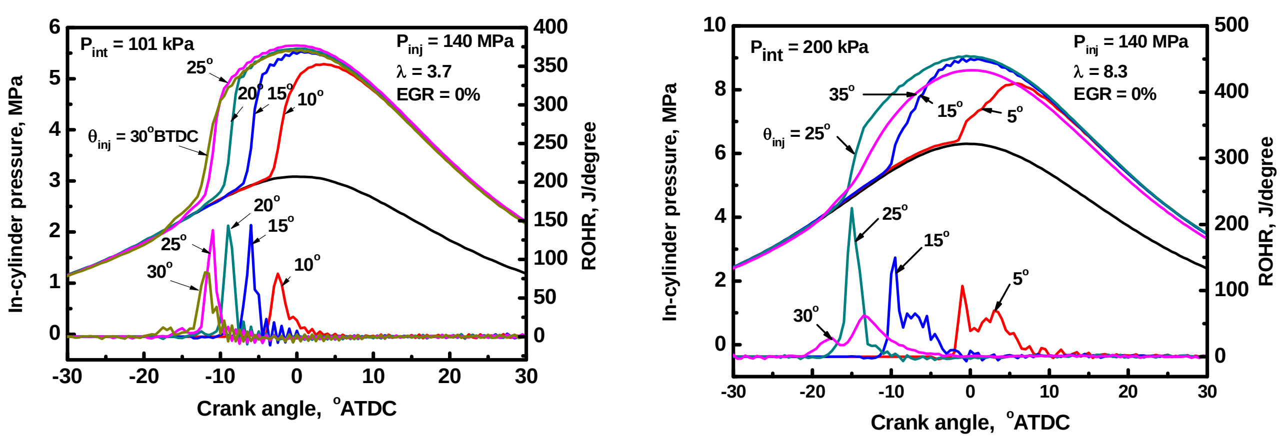


Fig.4. Effect of injection pressure on specific emissions

3.2. Effect of intake pressure

In order to understand the effect of intake pressure on performance and emission characteristics of low emission and highly efficient PCCI engine, the intake pressures of Pint = 101, 150 and 200 kPa were considered for the constant fuel quantity of 15 mg/cycle. Due to the increase in the air density with the cylinder the value of lambda for the supercharged condition was higher than that of the naturally aspirated. Fig. 5 shows the pressure history and rate of heat release for the varied intake pressure. Higher intake pressure led to higher rate of heat release and hence higher in-cylinder pressure.

Fig. 6 shows the effect of intake pressure on indicated thermal efficiency, IMEP and COVIMEP. Higher intake pressure led to higher indicated thermal efficiency and IMEP. This was thought to relate to the amount of oxygen available for combustion under higher intake pressure promoting combustion process. The optimum condition for PCCI combustion was noted at injection timing between θ_{inj} = 15 and 25°BTDC.



P_{int} = 101 kPa b) P_{int} = 200 kPa

Fig.5. Pressure history and rate of heat release (ROHR) P_{inj} = 140 MPa, λ = 3.7 and 8.3, P_{int} = 101 and 200 kPa, 0%EGR

The in-cylinder pressure and rate of heat release was higher for the case with P_{int} = 200 kPa. As the injection timing is advanced the trend is similar that is both in-cylinder pressure and ROHR increase to a maximum at θ_{inj} = 25°BTDC then decreased thereafter. PCCI combustion noted to commence from θ_{inj} = 25°BTDC in case of high intake pressure but for the lower intake pressure case the transition is θ_{inj} = 20°BTDC. The late injection timing of θ_{inj} = 5 to 15°BTDC were typical of conventional diesel combustion with both the premixed and diffusion controlled combustion phase being observed.

The COV_{IMEP} for the intake pressure considered was less than 4% indicating that this combustion process was very stable.

Fig. 7 shows the effect of intake pressure on specific emissions. Higher intake pressure was noted to be effective in the reduction of smoke and NOx emissions. This could be attributed to the abundant amount of air hence allowing better mixing and atomization of the fuel while diluting the fuel in the air achieving very lean mixture hence suppressing the in-cylinder temperature. The emissions of CO and HC was relative higher for the high intake pressure case but a minimum was noted for the injection timing between θ_{inj} = 15 and 25°BTDC for the CO but slightly lower HC for the 101 kPa. This phenomenon was thought to be related to the spray targeting spot where the spray would strike the wall of the Derby hat piston bowl and since the air quantity inside the combustion chamber was high would lead to better mixing of the fuel and air. This is in agreement with our previous work [9]. For the early injection timing of θ_{inj} = 35°BTDC the fuel would impinge on the cylinder wall leading to incomplete combustion and hence the presence of HC emission for the Pint = 200 kPa case.

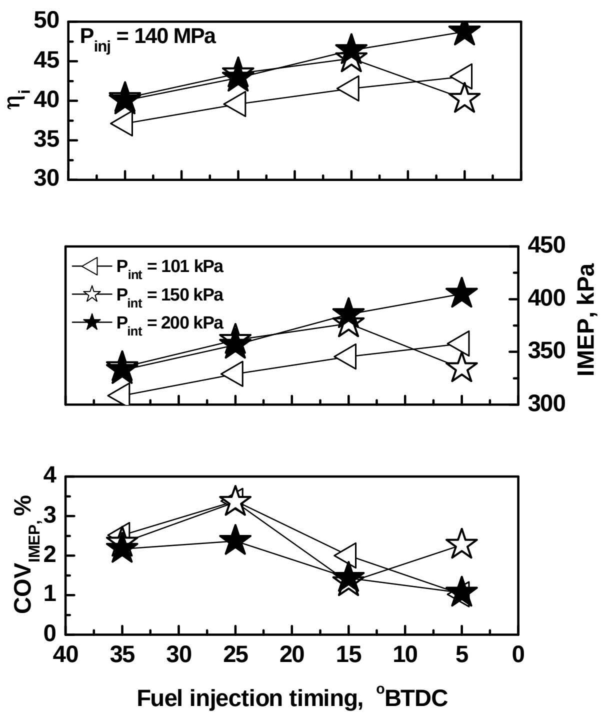


Fig.6. Effect of intake pressure on indicated thermal efficiency, IMEP and COV_{IMEP}

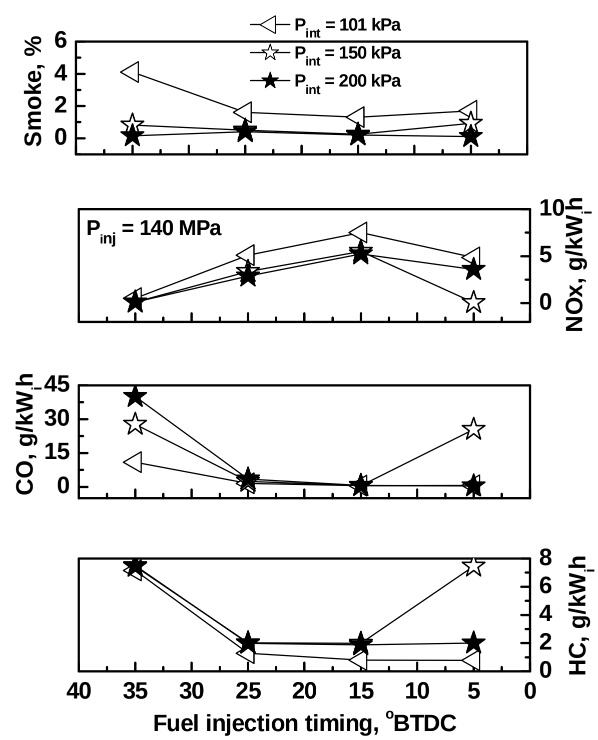


Fig.7. Effect of intake pressure on the specific emissions

3.3. Effect of EGR

In order to understand the effect of exhaust gas recirculation (EGR) on combustion and emission characteristics of a low emission and highly efficient PCCI engine the EGR was varied at higher injection pressure of P_{inj} = 140 MPa. Fig. 8 shows the pressure history and the rate of heat release for 0 and 45% EGR. The introduction of EGR led to lower in-cylinder pressure and ROHR with retarded combustion phasing. EGR has the dilution effect which leads to lean mixtures and lower in-cylinder temperature. This in effect would lead to extended ignition delay and hence longer premixing time for the air and fuel.

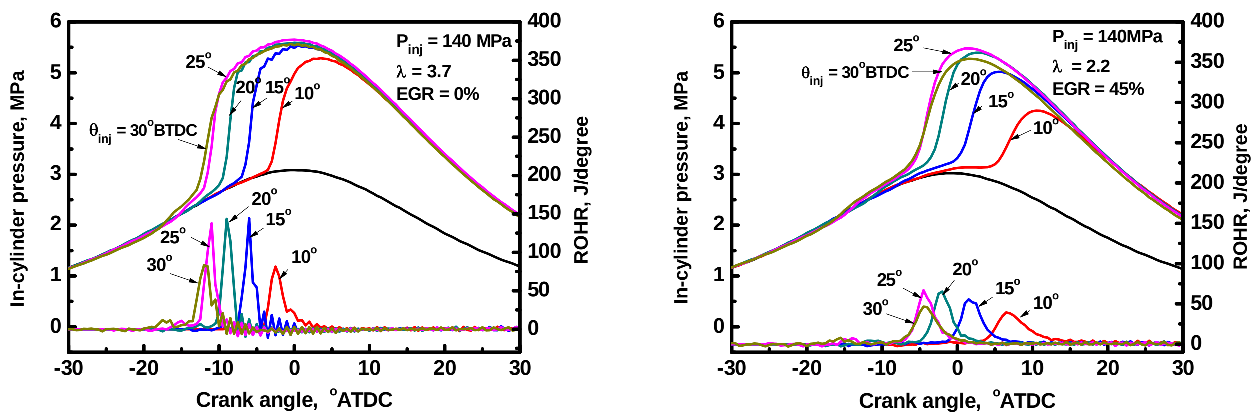
Under PCCI combustion two-stage heat release was observed typically representing the low temperature oxidation (LTO) and high temperature oxidation (HTO) with a negative temperature coefficient (NTC) in between. For all the injection timings with EGR this phenomena was observed but in case without EGR it occurred for θ_{inj} = 20°BTDC and earlier only. Milder combustion with less negative work was observed for all the cases with EGR evidenced by the combustion phasing being close to or after TDC.

In order to understand the effect of EGR on performance on highly efficient PCCI diesel engine an analysis was carried out. Fig. 9 shows the effect of EGR on indicated thermal efficiency, IMEP and COV_{IMEP} for P_{inj} = 140 MPa with 0 and 45% EGR.

EGR was found to lead to higher indicated thermal efficiency and IMEP. This was attributed to longer ignition delay hence a longer time available for the fuel and air to premix before combustion. Injection timing of θinj = 20°BTDC was noted with the highest indicated thermal efficiency of η_i = 45% and IMEP = 375 kPa. The COV_{IMEP} was found to be less than 3.5% for the two conditions considered with the case of EGR having lower values.

Fig. 10 shows the effect of EGR on specific emissions for P_{inj} = 140 MPa with 0 and 45% EGR. Smoke emissions was noted to slightly increase with a minimum being observed between θ_{inj} = 15 and 20° BTDC. EGR was found to be very effective in reduction of NOx but led to slight increase in CO and HC emissions. This was due to the nitrogen gas displacing the oxygen (dilution effect) leading to its deficiency and hence incomplete combustion of some of the fuel. This phenomenon was also noted in our previous work with lower fuel quantity where the introduction of EGR led to better engine performance and lower emissions [9],[10].

It can be clearly seen that a low emission and highly efficient PCCI combustion can be achieved by combining higher injection pressure, higher intake boost pressure and EGR with moderately early injection timing between θ_{inj} = 15 and 25°BTDC



a) EGR = 0% b) EGR = 45%

Fig.8. Pressure history and rate of heat release (ROHR) P_{inj} = 140 MPa, λ = 3.7 and 2.2, 0 and 45% EGR

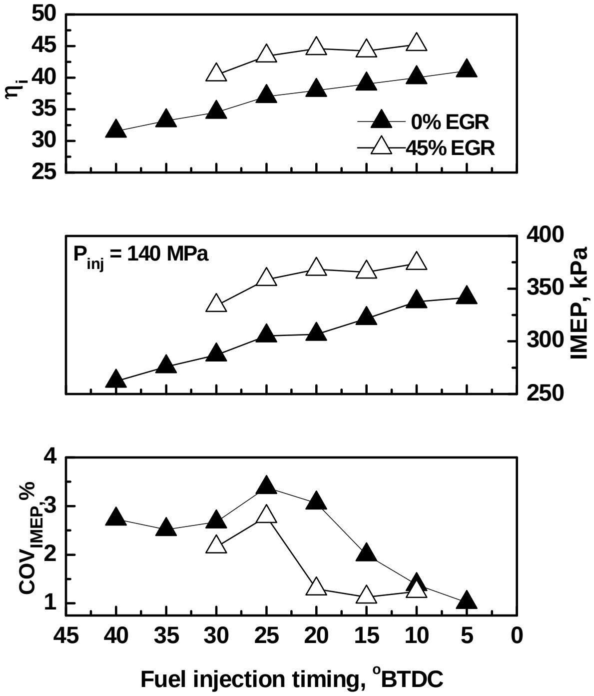


Fig.9. Effect of EGR on indicated thermal efficiency, IMEP and COV_{IMEP}

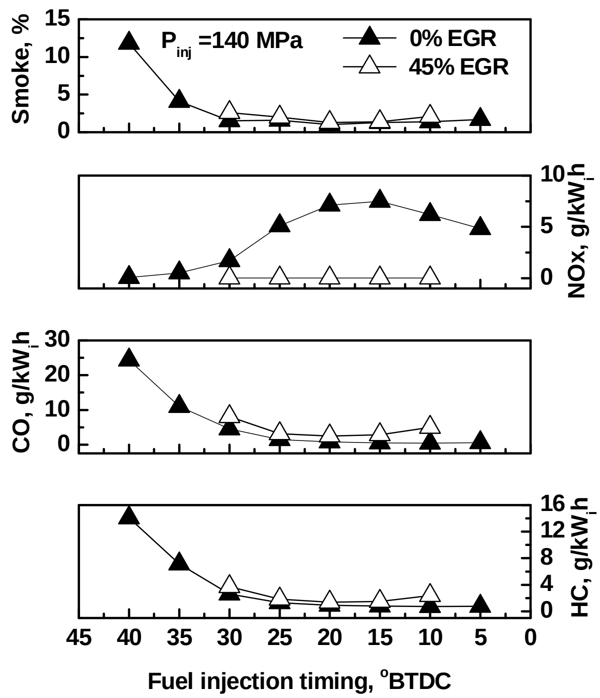


Fig.10. Effect of EGR on specific emissions

4. Conclusions

The main objective of this paper was to evaluate a low emission and highly efficient PCCI diesel engine. The key parameters that we varied to clearly understand the strategy were the injection pressure, injection timing, intake pressure and the EGR.

The following conclusions were arrived at in this work:

1. High injection pressure led to high indicated thermal efficiency, IMEP and less than 3.5% COV_{IMEP}. Simultaneously low emission of NOx and smoke for the high injection pressure were achieved with relatively same emissions of CO and HC.

2. Higher boost pressure was found to lead to higher indicated thermal efficiency, IMEP and COV_{IMEP} of less than 4%. The high boost also led to low smoke, and NOx, same CO but slightly higher HC emissions.

3. The introduction of EGR led to very low emission of NOx and smoke but led to a slight increase in HC and CO. Higher indicated thermal efficiency and IMEP were achieved with EGR with COV_{IMEP} being less than 4%.

It can be clearly deduced from the above conclusions that combining the three parameters: high injection pressure, intake pressure and EGR would ultimately result in low emission and high efficient PCCI engine under optimized conditions.

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