Fuel Injection Pressure and Combustion Chamber Geometry Effects on the Performance and Emission Characteristics of Diesel Fueled CI Engine with EGR

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1 Introduction

Diesel locomotives are the main source of power for the most of the industrial and domestic applications owing to their greater thermal efficiency, higher brake power output, reliability, low fuel use and durability. An innovative variety of injection systems have been designed to get higher injection pressure. In the present scenario, the source of conventional hydrocarbon fuel has been depleting with an increase in its usage as there is an accelerating demand for energy.

The mixing of air and fuel is a vital factor in combustion. The decrease in fuel penetration for the available air in the chamber causes incomplete fuel burning. Therefore, increase in fuel injection pressure and modification of combustion chamber geometry would improve the fuel penetration into air present in the combustion chamber which results in complete atomization of air and fuel [1, 2]. The performance and emission are indicatively influenced by injector type, injection pressure and chamber design in a diesel engine. The performance mostly depends on characteristics of the fuel spray which would be achieved by proportionately high pressure. In olden day's diesel engine, the fuel is injected at pressure of 250 bar but, nowadays the common rail direct injection system employs very high injection pressure at 2000 bar. In this common rail direct injection system, the fuel rail pressure keeps up the injection pressure of the fuel [3, 4].

Very small sized drops are produced by decreasing the nozzle hole diameter of injector, which leads to reduction of time taken for atomization. However, this would thin out the spray penetration due to low spray momentum of fuel. [5]. Wang et al.

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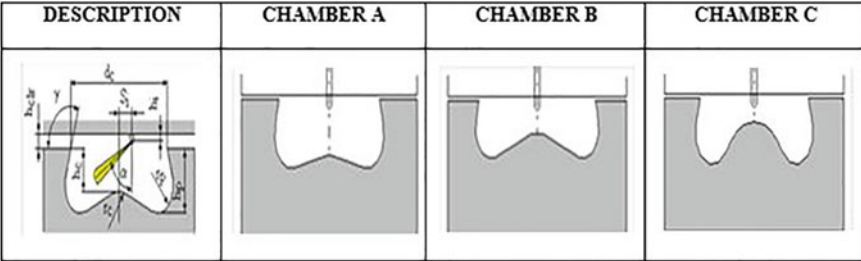e-mail: tamil.maritime@gmail.com

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investigated the effect of injector holes numbers and different injector nozzle diameter on the structure of flame and soot formation of single cylinder compression ignition (CI) engine; the result was reduced atomization time and rapid fuel distribution throughout the combustion chamber [6]. Kuleshov et al. explored the multi-zone thermodynamic model for calculating the performance and combustion parameters of closed cycle direct injection (DI) diesel engine [7]. These studies explain the performance parameters and emission characteristics of single cylinder CI engine using multiple injection method with multi-zone thermodynamic model software. Diesel-RK software is a simulation software of diesel engine which simulates the IC engine combustion by changing the parameters like modifying the piston bowl geometry, injection spray angle, injection timing, injection location, injection pressure, valve timing, turbocharge parameters, EGR, etc. [8–14]. Exhaust gas recirculation (EGR) plays a significant role in reducing the oxides of nitrogen (NOx) of diesel engines by decreasing the flame temperature along with oxygen concentration of the air and fuel mixture in the combustion chamber [15].

From the points of literature review, it is understood that conventional diesel engine receives fuel at the pressure of somewhat up to 250 bar. However, recent advanced fuel injection system injects the fuel at a pressure of 2000 to 3000 bar that would result in reduction of soot formations [6]. The present investigation intends to improve the air efficiency in the combustion chamber. In order to achieve so, three different swirl combustion chambers have been developed and experimentally investigated to study the performance and emission characteristics with respect to injection pressure when the EGR is employed (Fig. 1).

2 Experimental Apparatus and Procedure

2.1 Experimental Setup

The experiment carried out with a single cylinder, four stroke, CRDI, variable compression ratio (VCR) diesel engine which is coupled with eddy current dynamometer. Table 1 indicates the test engine technical specifications and Fig. 2 represents the schematic layout of the experimental setup. Additional setups for measuring injection pressure, in-cylinder pressure and crank angle of the engine were included. Injection pressure and in-cylinder pressure were measured using piezoelectric sensor. The sensor was attached near the injector and head of the engine, respectively. The setup consists of fuel flow measurement devices for measuring the mass flow rate of air and fuel, respectively. Rotameters were implemented to measure the cooling water flow to the engine. Also, online measurement device was used to study the degree of exhaust gas temperature, engine cooling water intake and outlet temperature and engine load. Data acquisition was employed to protocol the signal from the measuring devices to a computer and to display the in-cylinder pressure (ICP), heat release rate (HRR) and cylinder temperature with respect to

DESCRIPTION	CHAMBER A	CHAMBER B	CHAMBER C

Description	Chamber-A	Chamber B	Chamber-C
External diameter, dc,[mm]	50.2	57.2	53.2
In-centre piston bowl depth, hc, [mm]	14	8	6
Radius of sphere in centre of piston bowl, rc, [mm]	15.59	13.59	15.59
Depth of combustion chamber in periphery, hp, [mm]	17.1	14.1	17.1
Radius of hollow chamber in periphery of bowl, rp, [mm]	8.31	7.31	8.31
Inclination angle of a bowl forming a plane of the piston crown, gamma, [deg]	105	105	105
Top-clearance at TDC, h_clr, [mm]	1.001	1.001	1.001
Swirl ratio	3.1431	2.534	2.8555

Fig. 1 Geometric parameters of the chambers A, B and C

crank angle. The sensed signals from the engine are linked to a computer through by a data acquisition system (DAQ) and the engine performance analyzing software "Engine Soft LV" which displays the P-H and P-V diagram. The setup supports the study of performance parameter of the engine that includes brake power (BP), BTE, SFC and combustion analysis.

2.2 Emission Measurements

An exhaust gas measuring instrument was used to measure and analyze the gas composition of the exhaust gas released by a compression ignition engine. The analyzer AVL444 is used to measure the concentration of nitrogen oxides (NO_X, in ppm), hydrocarbons (HC, in ppm), carbon monoxide (CO in % vol), carbon dioxide (CO_2 in % vol) and oxygen (% vol). It was also used for quantifying the opacity of the gas exhausted by the engine. It also quantifies the opacity of the smoke in

Table 1 Technical specification of the test engine

Engine operation parameters	Specifications	Engine operation parameters	Specifications
Engine model	Kirloskar	Fuel injection pressure	600–1400 bar
Cooling system	Water cooled	Fuel injection system	Direct injection (drive solenoid driven)
Number of strokes	4 Strokes	High pressure system	Common rail direct injection
Number of cylinder	Single cylinder	Electronic control unit	Model Nira i7r
Cylinder bore/stoke	87.5 mm/110 mm	Valve per cylinder	2 Valves (1 Inlet. 1 Outlet)
Swept volume	661.45 cc	Liner type/bases	Wet
Compression ratio (CR)	17.5:1	Exhaust gas recirculation (EGR)	SS, water cooled
Rated speed (rpm)	1500	Analysis software	"Engine soft" Engine performance analysis software

1. Engine
2. Eddy current dynamometer
3. Intake air filter
4. Air mass flow meter
5. Fuel tank
6. Fuel meter
7. Fuel filter
8. Fuel pump
9. Common rail
10. Fuel injector
12. Coupling
13. (EGR) flow control valve
14. Engine smoke meter
15. AVL444 five gas analyser
16. Data acquisition system
17. Cooled water inlet
18. Hot water outlet
19. Exhaust line (silencer)
20. Piezoelectric pressure sensor
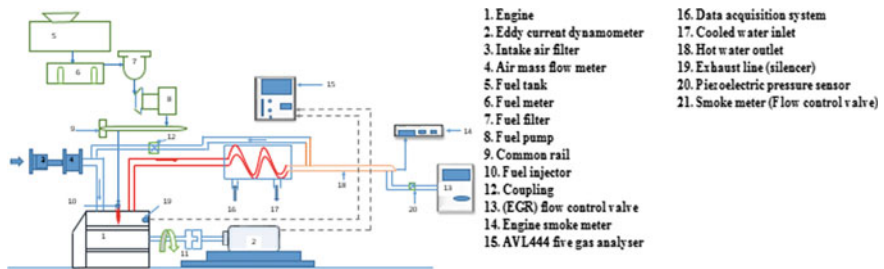21. Smoke meter (Flow control valve)

Fig. 2 Schematic diagram of the experimental test setup

Hartridge smoke unit (HSU). Figure 3 shows AVL 444 five gas analyzer and smoke meter.

2.3 Exhaust Gas Recirculation System

Exhaust gas recirculation is a best way to minimize the emission in diesel engines. The EGR recirculates the exhaust combusted gas about 10% to 30% into inlet manifold of the engine, during the suction stroke it gets mixed with the inlet ambient air, which in turn diminishes the combustion temperature and hence reduce the oxides of nitrogen. The recirculation percentage of EGR intends to improve the performance and reduce the emission characteristics of single cylinder CI diesel engine. It was found from Fig. 4 that connecting EGR to the inlet manifold is a better method of

Fig. 3 Assembly of AVL 444 and smoke meter emission measurement systems

Fig. 4 Experimental exhaust gas recirculation (EGR) setup

employing EGR in a single cylinder diesel engine. About 80% of oxides of nitrogen were reduced when recycled at 15% of EGR. Based on literature review, the exhaust gas recirculation increases the reduction of combustion temperature and also reduces the NO_X. But it increases the brake specific fuel consumption.

2.4 Experimental Procedure

The performance and emission test for the experimental engine were carried out at various Cycle Fuel Mass (g/cycle) CFM, which was computed from the experimental TFC and it varies from 0.0111 g/cycle to 0.0289 g/cycle, from low to high load. It was used as an input fuel quantity for thermodynamic simulation software Diesel-RK. First the simulation was carried out at various CFM, by simultaneously varying the IP and EGR using diesel as fuel for CI engine. Based on the simulation results, the suitable combustion chamber was selected and the experiment was carried out in single cylinder CI diesel engine. The injection pressure was varied by using CRDI fuel injection system as shown in Fig. 5. The experiment was carried out in the base engine and suitable combustion chamber was selected from the simulation results from the
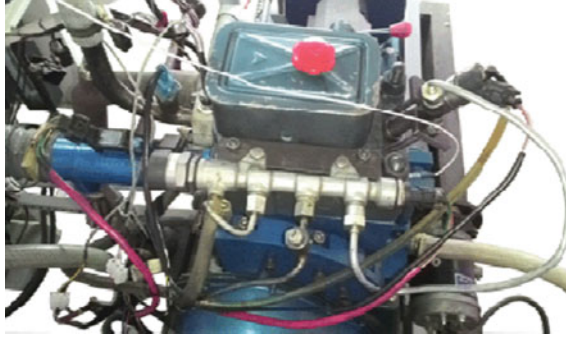
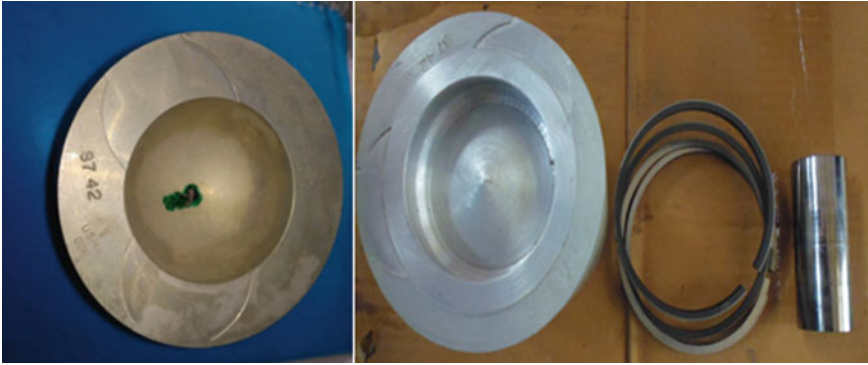**Fig. 5** Common rail fuel injection system

Fig. 6 Photographic view of standard and modified combustion chamber

Diesel-RK software as shown in Fig. 6. The performance and emission characteristics of the single cylinder CI diesel engine were conducted at a compression ratio of 17.5:1, at five injection pressures (600, 800, 1000, 1200 and 1400 bar) maintained at a constant speed of 1500 rpm with and without 15% EGR. The performance and emission tests were recorded from experimental setup and were assessed on the basis of SFC, BTE and emissions of particulate matter (PM) and NO_X.

3   Results and Discussion

The three modified combustion chambers were analyzed based on their performance, combustion characteristics and their emission parameters. They were also compared for SFC, NO_X, PM and ignition delay (ID). The best among the three modified

combustion chambers were compared with standard diesel engine chamber for in-cylinder pressure (ICP), heat release rate (HRR), specific fuel consumption, brake thermal efficiency and oxides of nitrogen.

3.1 Performance Analysis

3.1.1 Specific Fuel Consumption (SFC)

Figures 7, 8, 9, 10, 11 and 12 show the variation of SFC against IP and various bowl geometry with and without 15% EGR at 0%, 50% and 100% load conditions. SFC is the ratio between fuel mass consumed by the engine to the applied brake power. Figures 7 and 8 show the comparison of SFC of three modified combustion chambers at low load condition. The SFC of chamber-A decreases with an increase in IP from 600 to 800 bars, but slightly increases with an increase in IP from 800 to 1400 bar due to the power required to increase the injection pressure. From Figs. 7, 8, 9, 10,

Fig. 7 Variation of SFC against IP and various bowl geometry without EGR (0% load)
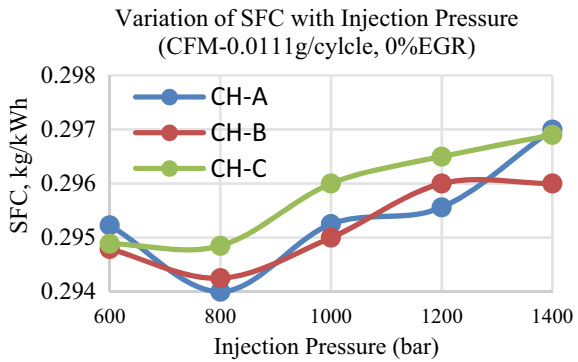
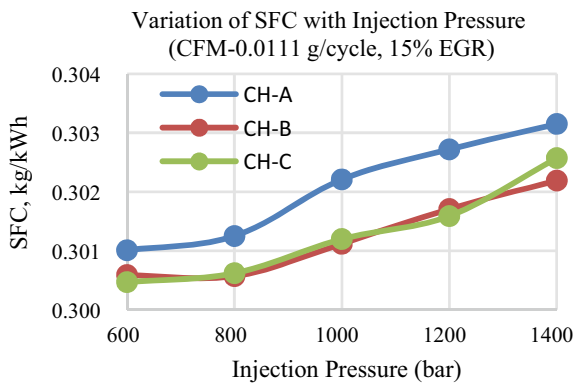**Fig. 8** Variation of SFC against IP and various bowl geometry with 15% EGR (0% load)

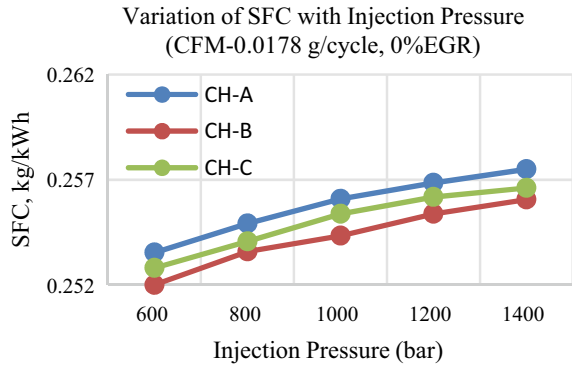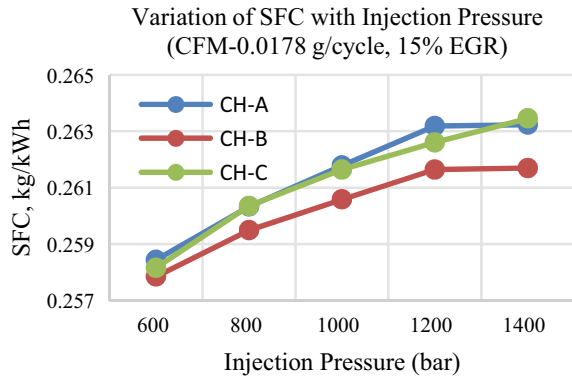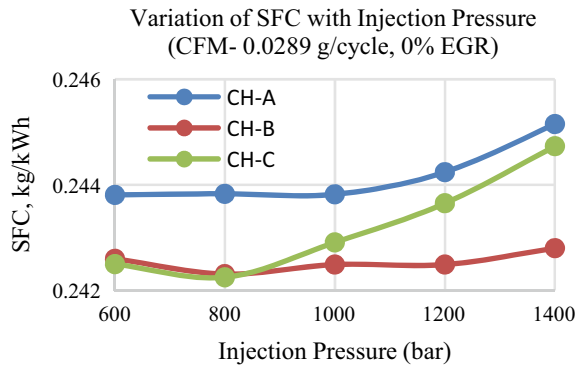**Fig. 9** Variation of SFC against IP and various bowl geometry without EGR (50% load)

Variation of SFC with Injection Pressure (CFM-0.0178 g/cycle, 0%EGR)

Fig. 10 Variation of SFC against IP and various bowl geometry with 15% EGR (50% load)

Variation of SFC with Injection Pressure (CFM-0.0178 g/cycle, 15% EGR)

Fig. 11 Variation of SFC against IP and various bowl geometry without EGR (100% load condition)

Variation of SFC with Injection Pressure (CFM- 0.0289 g/cycle, 0% EGR)

11 and 12, it is noticed that the SFC of chamber-A is increased from 0.295 g/kWh to 0.296 g/kWh, 0.254 g/kWh to 0.258 g/kWh and 0.244 to 0.245 g/kW with increasing in injection pressure from 600 to 1400 bar at 0%, 50% and 100% load conditions, respectively. From Fig. 11, SFC of chamber-B varied from 0.243 to 0.242 g/kWh

Fig. 12 Variation of SFC against IP and various bowl geometry with 15% EGR (100% load conditions)
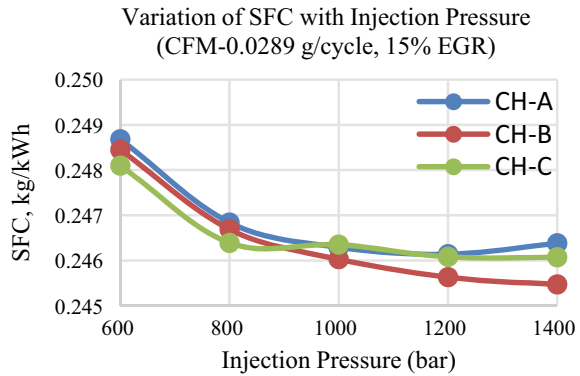
which is slightly lower when compared to chamber-A. SFC of chamber-B is slightly lower than the chamber-C at all the conditions. The SFC of combustion chamber-A with 15% EGR mode simulation results is shown in Fig. 8, 10 and 12 and it varies from 0.301 to 0.303 g/kWh, 0.258 to 0.263 g/kWh and 0.249 to 0.246 g/kWh at different load conditions, respectively. SFC of chamber-A at 600, 800, 1000, 1200 and 1400 bar injection pressure with 15% EGR mode simulation increased at 2.1%, 1.3%, 0.8%, 0.85% and 0.84% when compared to without EGR at high load condition. The variation of SFC with IP in chamber-B is lower (about 0.4% at 0% load, 0.4 to 0.8% at 50% load, 0.4 to 0.56% at 100% load) compared to other chambers C and A. In most cases, SFC of all three chambers without EGR mode is lower when compared to with 15% EGR mode simulation. From the above observations, it can be concluded that piston bowl chamber-B is a better combustion chamber when compared to other two chambers.

3.1.2 Nitrogen Oxide (NO_X)

NO_X is one of the major pollution in CI engine which is formed by chain reaction that takes place between oxygen and nitrogen in air. This reaction is highly dependent on temperature. Figures 13, 14, 15, 16, 17 and 18 show the variation of NO_X with respect to IP and of three modified chambers in with 15% and without EGR mode at 0%, 50% and 100% load conditions. NO_X of chamber-A without EGR mode increased from 5.838 g/kWh to 7.751 g/kWh, 9.629 g/kWh to 13.674 g/kWh and 5.677 g/kWh to 10.030 g/kWh at 0%, 50% and 100% load conditions. Also, at 15% EGR mode, it increased from 0.731 to 1.52 g/kWh, 1.583 to 2.520 g/kWh and 1.684 to 3.800 g/kWh at all three different load conditions. From Fig. 18 the Cycle Fuel Mass (CFM) is as same as the value, 0.0289 g/cycle. NO_X of chamber-A at 600, 800, 1000, 1200 and 1400 bar injection with 15% mode simulation decreased by 71.2%, 54.1%, 59.1%, 57.3% and 63.6% when compared with that of 0% EGR mode. The variation of NO_X with IP of chamber-B is lower (about 2.5% at 0% load, 2–7% at 50% load, 3.9 to 10.5% at 100% load) when compared to other chambers C and A.

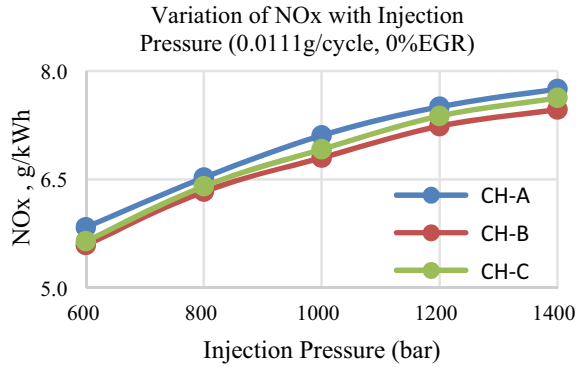**Fig. 13** Variation of NO$_X$ against IP and various bowl geometry without EGR (0% load)

Variation of NOx with Injection Pressure (0.0111g/cycle, 0%EGR)

CH-A
CH-B
CH-C

NOx , g/kWh

Injection Pressure (bar)

Fig. 14 Variation of NO$_X$ against IP and various bowl geometry with 15% EGR (0% load)

Variation of NOx with InjectionPressure (CFM-0.0111 g/cycle, 15% EGR)

CH-A
CH-B
CH-C

NOx, g/kWh

Injection Pressure (bar)

Fig. 15 Variation of NO$_X$ against IP and various bowl geometry without EGR (50% load)

Variation of NOx with Injection Pressure (CFM-0.0178 g/cycle,0%EGR)

CH-A
CH-B
CH-C

NOx , g/kWh

Injection Pressure (bar)
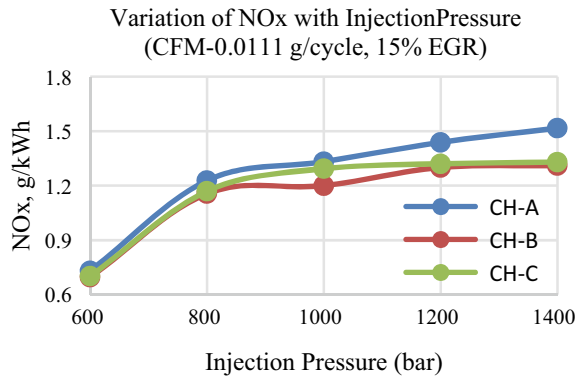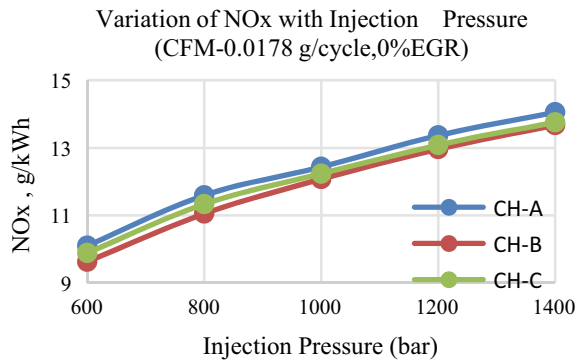
The piston bowl chamber-B is more suitable for combustion temperature than the other two chambers. It is observed that NO$_X$ emission is relatively lesser in 15% EGR mode when compared to without EGR mode, as the oxygen content at the inlet air supply is comparatively low.

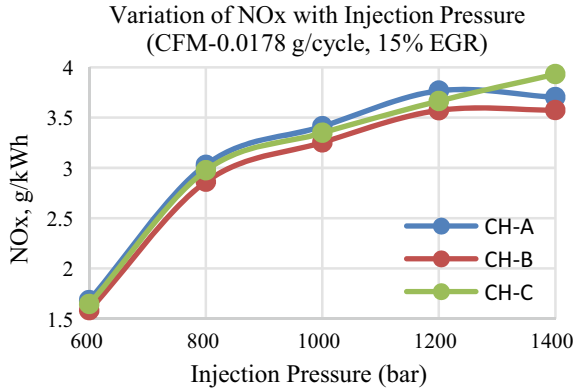**Fig. 16** Variation of NO$_X$ against IP and various bowl geometry with 15% EGR (50% load)

Variation of NOx with Injection Pressure (CFM-0.0178 g/cycle, 15% EGR)

Fig. 17 Variation of NO$_X$ against IP and various bowl geometry without EGR (100% load)

Variation NOx with Injection Pressure (CFM-0.0289 g/cycle,0%EGR)

Fig. 18 Variation of NO$_X$ against IP and various bowl geometry with 15% EGR (100% load)

Variation of NOx with Injection Pressure (CFM-0.0289 g/cycle, 15% EGR)
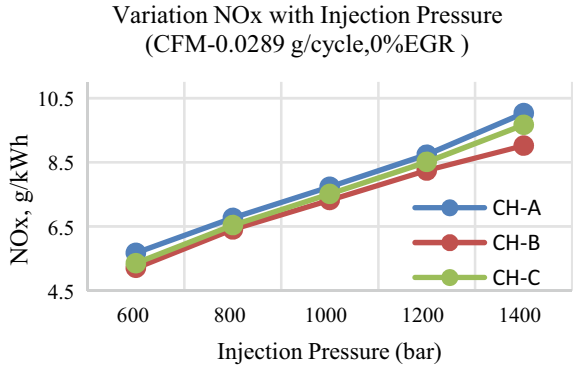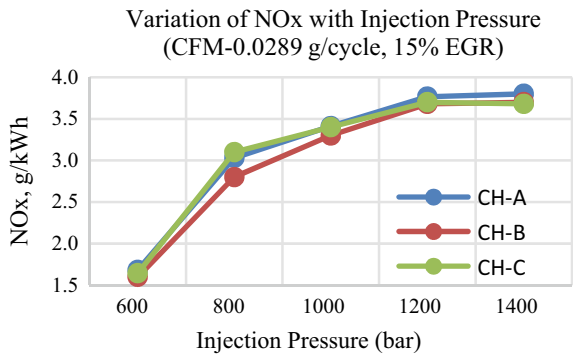
3.1.3 Particulate Matter (PM)

PM emissions from diesel engines are the major contributor to the ambient air contamination problem. Particulate emission draws the attention of several

researchers in recent time, as the primary reemerges in the up-gradation of fuel injection system in diesel engine injects fuel directly into the cylinder. Figures 19, 20, 21, 22, 23 and 24 show the variation of PM against injection pressure and various bowl geometry with and without 15% EGR at 0%, 50%, and 100% load conditions. Variation of PM can be observed from Fig. 19. Chamber-A decreases PM

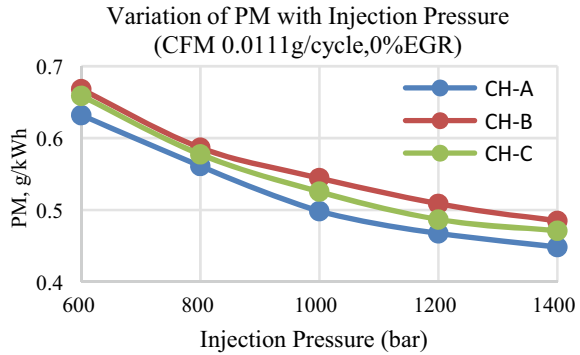**Fig. 19** Variation of PM against IP and various bowl geometry without EGR (0% load)

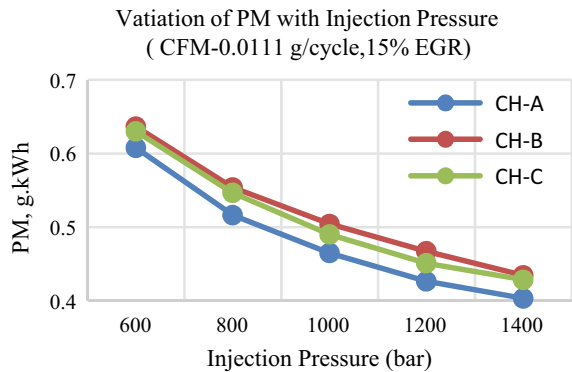**Fig. 20** Variation of PM against IP and various bowl geometry with 15% EGR (0% load)

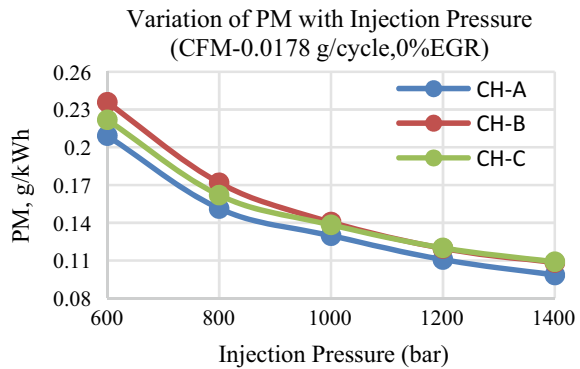**Fig. 21** Variation of PM against IP and various bowl geometry without EGR (50% load)

Fig. 22 Variation of PM against IP and various bowl geometry with 15% EGR (50% load)
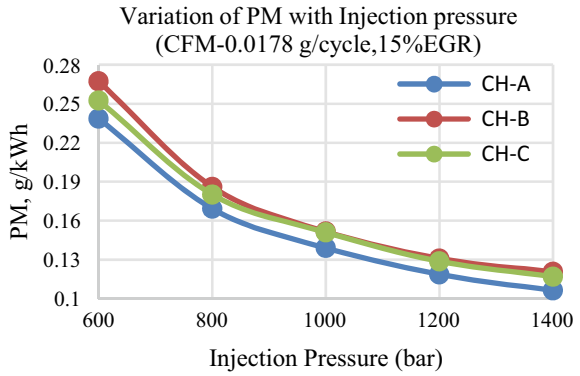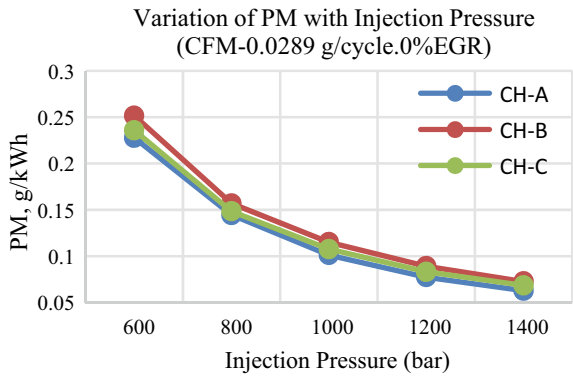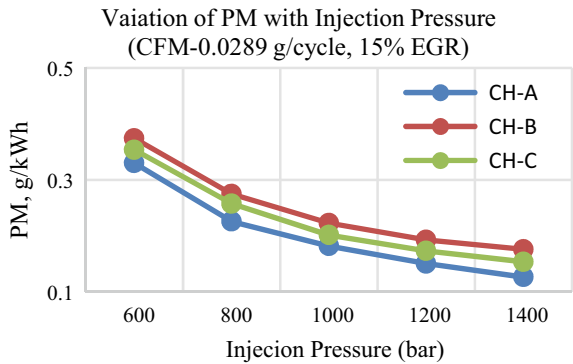
Variation of PM with Injection pressure (CFM-0.0178 g/cycle,15%EGR)

Fig. 23 Variation of PM against IP and various bowl geometry without EGR (100% load)

Variation of PM with Injection Pressure (CFM-0.0289 g/cycle.0%EGR)

Fig. 24 Variation of PM against IP and various bowl geometry with 15% EGR (100% load)

Vaiation of PM with Injection Pressure (CFM-0.0289 g/cycle, 15% EGR)

from 0.632 g/kWh to 0.449 g/kWh with the IP increasing from 600 to 1400 bar. It is also noted that PM decreases with increase in pressure in both with and without EGR modes. Reduction in PM may be associated with higher injection pressure,

that arising as the result of enhanced mixture formation of air and fuel for impoverished combustion. From Fig. 23 it is observed that PM is reduced from 0.236 at 600 bar to 0.108 g/kWh at 1400 bar. At 15% EGR simulation, results of PM vary from 0.239 g/kWh to 0.106 g/kWh as shown in Fig. 24 PM of chamber-A at 600, 800, 1000, 1200 and 1400 bar injection with 15% mode simulation increased to 14%, 11%, 7%, 7.2% and 7.4% when compared to without EGR. But chamber-A is lower (nearly 4.5%, 10 to 12.5%, 10 to 11.5%) compared to other chambers B and C at varying the 0%, 50% and 100% load conditions. The PM of modified three combustion chambers of without EGR is slightly lower than 15% EGR mode. Besides, another reason for increase of PM in EGR mode may be attributed to unavailability of oxygen contained in the combustion chamber.

3.1.4 Ignition Delay (ID)

Ignition delay (ID) is a vital parameter in determining the knocking characteristics of diesel engine. Figures 25, 26, 27, 28, 29 and 30 show the variation of Ignition delay (ID) against injection pressure (IP) and various bowl geometry with and without 15%

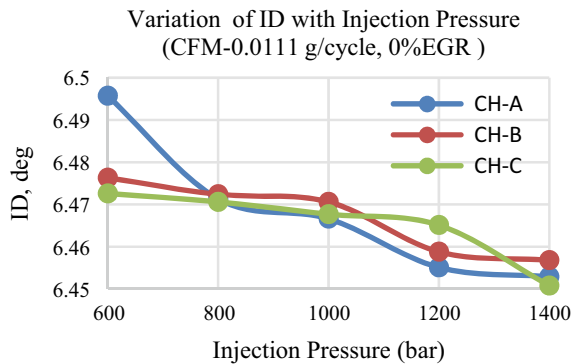**Fig. 25** Variation of ID against IP and various bowl geometry without EGR (0% load)

Fig. 26 Variation of ID against IP and various bowl geometry with 15% EGR (0% load)
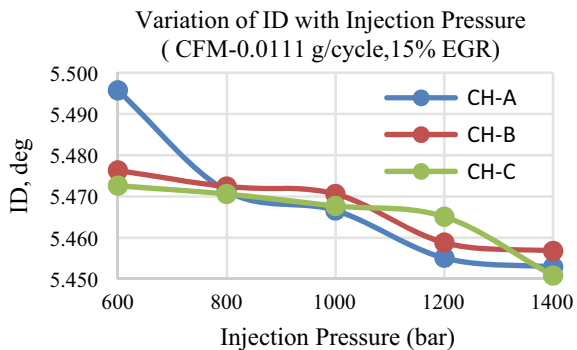
Fig. 27 Variation of ID against IP and various bowl geometry without EGR (50% load)

Variation of ID with Injection Pressure
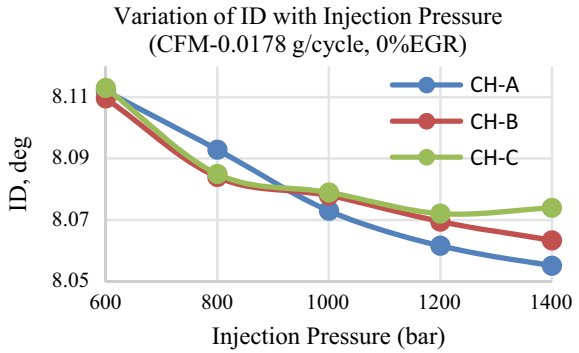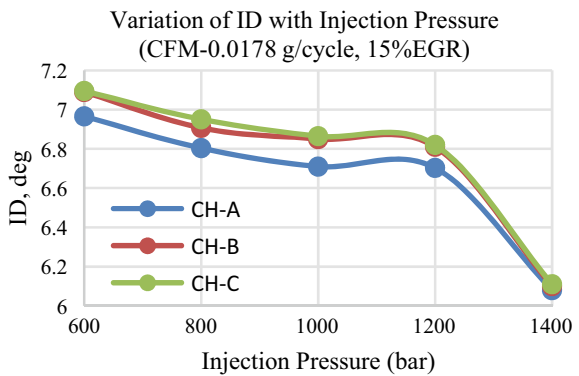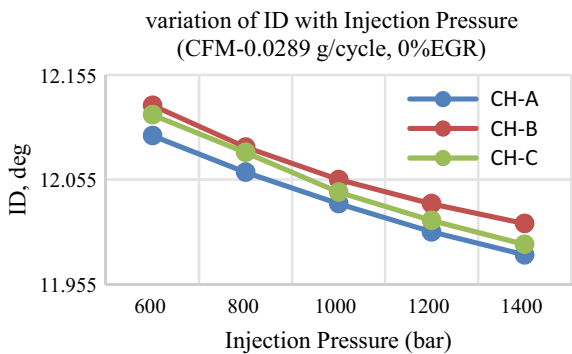(CFM-0.0178 g/cycle, 0%EGR)

ID, deg

Injection Pressure (bar)

Fig. 28 Variation of ID against IP and various bowl geometry with 15% EGR (50% load)

Variation of ID with Injection Pressure
(CFM-0.0178 g/cycle, 15%EGR)

ID, deg

Injection Pressure (bar)

Fig. 29 Variation of ID against IP and various bowl geometry without EGR (100% load)

variation of ID with Injection Pressure
(CFM-0.0289 g/cycle, 0%EGR)

ID, deg

Injection Pressure (bar)

EGR at 0%, 50%, and 100% load conditions. Figure 25 shows the ID decreases from 6.49° to 6.45° with increase in IP from 600 to 1400 bar that means ID decreases with an increase in pressure for both with and without EGR modes. The variation of NO_X with IP, chamber-C is lower (nearly 0.3 to 1.5%) compared to other chambers B and A at low load condition. This may be due to better air fuel mixing at high

Fig. 30 Variation of ID against IP and various bowl geometry with 15% EGR (100% load)
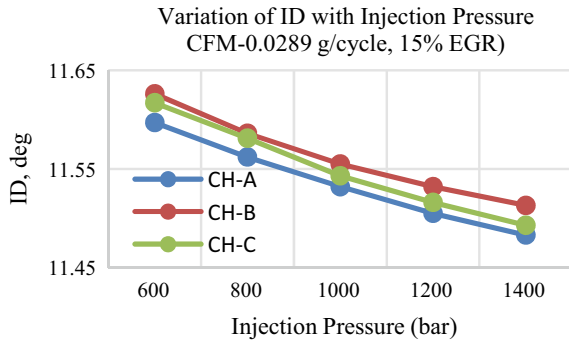
injection pressure and temperature inlet air. It is noticed that combustion chamber geometries reducing the ignition delay decreases with increase the injection pressure and also increases with increase in load. Figures 29 and 30 show the variation of ID with injection pressure of chambers A, B and C followed same trend at all load condition. But without EGR chamber-A is lower (nearly 0.3 to 1.2%) compared to other chambers B and C. Hence, chamber-A is suitable combustion chamber compared to other chambers namely B and C. Influence of geometric characteristics and emission parameters are also been investigated and compared for the chambers A, B and C. It can be concluded from the simulation results that the chamber-A has been better than other two chambers.

4 Experimental Validation of Standard Piston (STDP) and Modified Combustion Chamber-a (CH-a)

Figure 31 shows the geometric parameters of combustion chamber of standard piston bowl. Some parameters of the standard piston bowl geometrics are discussed namely external diameter (dc) −52 mm, in-center piston bowl depth (hc) −23 mm, depth of combustion chamber in periphery (hp) −18.1 mm. From Fig. 1, piston combustion chamber-A swirl ratio = 3.14 is predicted. The inclination angle of a bowl forming a plane of the piston crown is gamma (γ) −105°

4.1 Validation of in-Cylinder Pressure with Crank Angle (CA) for Chamber-a

Variation of in-cylinder pressure with CA diagram is shown in Fig. 32; Cycle Fuel Mass (CFM) is computed from the experiment. Total Fuel Consumption (TFC) varies from 0.0111 g/cycle to 0.0289 g/cycle and used as input fuel mass/cycle in full

Fig. 31 Piston Geometric Parameter of Standard Piston bowl (STDP)

Fig. 32 Variation of In-cylinder pressure with CA comparison (0.5 kW)
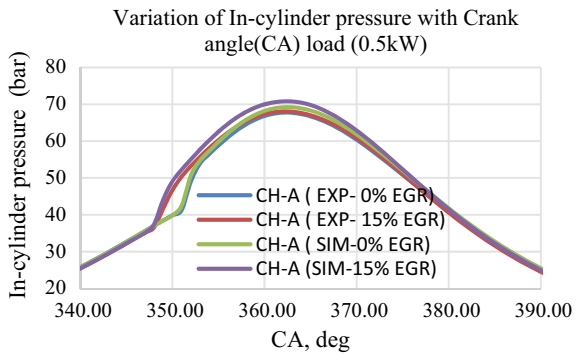
cycle simulation. Pressure variation of modified combustion chamber and standard combustion chamber follows similar form of pressure rise at all load conditions.

It is noted from Fig. 32 that simulated and experimental value of In-cylinder pressure is higher at 15% EGR condition compared to that in 0% EGR mode. This may be due to complete combustion of fuel at high inlet air temperature. Simulated result of in-cylinder pressure varied from 0.8% to 1.2% which is higher compared to experimental value. It can also be seen from Fig. 32 in 15% EGR mode the
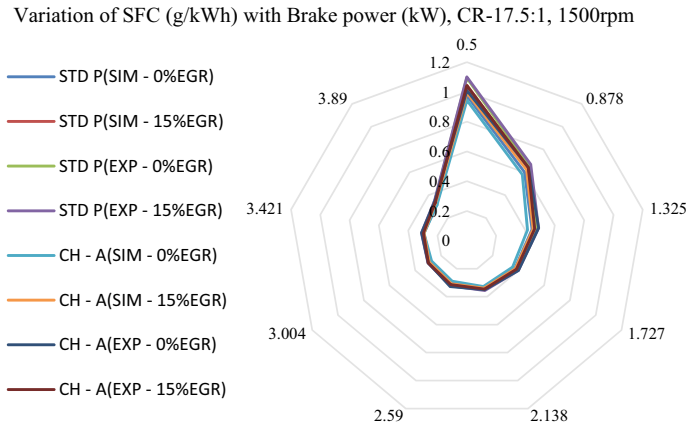
Fig. 33 Variation of SFC with BP at standard piston and modified chamber-A with and without EGR

combustion occurs at a 2 to 5° before the 0% EGR mode and also P_{max} occurs at a different crank angle after top dead center (TDC).

4.2 Validation of SFC with BP Comparison with and Without EGR

Simulation results are validated by experimentally namely SFC with BP as shown in Fig. 33. The SFC of standard and modified combustion chamber with and without EGR modes is discussed. The SFC for with 15% EGR is marginally higher than that of without EGR mode of standard and modified combustion chamber at low load conditions. SFC with EGR is almost similar to that of without EGR for diesel fuel at high load condition. SFC of STDP simulation is 3.6% to 10% lower compared with experimental results with and without EGR operation. SFC of experimental value of chamber-A is nearly 0.5 to 2.8% higher compared to simulation result and with and without EGR. The experimental and simulation result of STDP and CH-A is compared and the higher SFC for SDTP might be result of improper air and fuel blending, which leads to poor combustion.

4.3 Validation of BTE with BP Comparison with and Without EGR

Figure 34 shows the variation of brake thermal efficiency (BTE) for STDP and CH-A with respect to brake power. The BTE of STDP simulation result is higher when
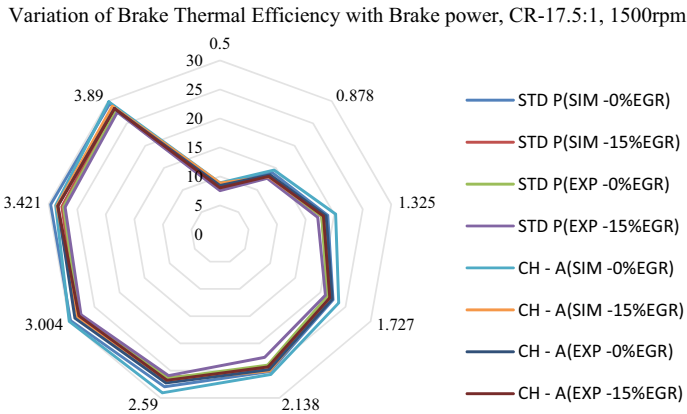
Variation of Brake Thermal Efficiency with Brake power, CR-17.5:1, 1500rpm

Fig. 34 Variation of SFC with BP at STDP and CH-A with and without EGR

compared to that of the experimental result at all load conditions with and without EGR. The BTE of simulation and experimental result increases for increase in BP with EGR, but at lower load it will decrease. This may be attributed to re-burning of hydrocarbon that enters during the recirculation of exhaust gases and at full load condition the BTE is nearby the same. BTE of STDP simulation is 5 to 9% higher compared experimental results with and without EGR. BTE of CH-A (EXP 15% EGR) is nearly 2 to 8% lower compared to CH-A (SIM-15%EGR). BTE of CH-A (EXP 0% EGR) is nearly 0.5% lower compared to CH-A (SIM-0%EGR). The experimental and simulation result of STDP and CH-A, the BTE for CH-A is higher compared to SDTP this may be due to of better air motion in CH-A.

4.4 Validation of NO_X with BP Comparison with and Without EGR

Figure 35 shows the variation of NO_X plotted against BP. The standard piston (STDP) simulation results are well in agreement with experimental results. The result shows that oxides of nitrogen increases with increase in the brake power due to high in-cylinder combustion temperature and also due to the excess amount of oxygen contained in combustion chamber. NO_X of CH-A (EXP 0% EGR) is nearly 1.4 to 1.7% lower compared to STDP (EXP-0%EGR). NO_X of CH-A (EXP 15% EGR) is nearly 15 to 25% lower compared to STDP (EXP-15%EGR). When the experimental and simulation result of STDP and CH-A was compared, the oxides of nitrogen for CH-A is lower to SDTP as a result of better air motion and shorter ignition delay in CH-A.
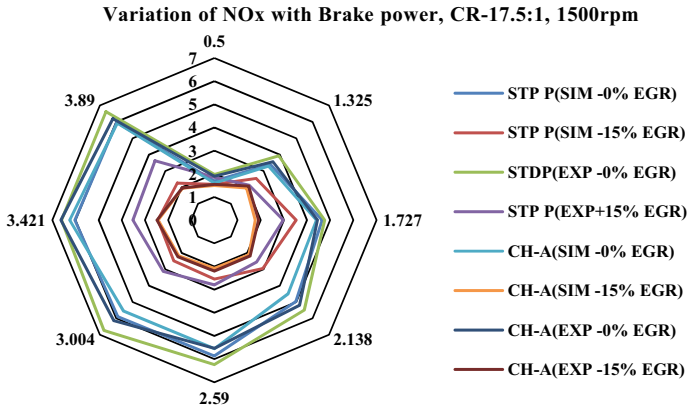
Fig. 35 Variation of NO$_X$ with BP at STDP and CH-A with and without EGR

5 Conclusion

In this experimental investigation, the performance and emission characteristics of three modified combustion chamber are investigated. The following conclusions have been drawn from the simulation and experimental result.

- This study concludes a desirable combustion chamber among three modified chambers with respect to injection pressure 600 bar to 1400 bar, with and without EGR mode. Based on the simulation result, the SFC of chamber-B is better than the other chambers A and C at 600 bar injection pressure without EGR. NO$_X$ of chamber-B is better than other chamber-A and C at 600 bar injection pressure with 15 EGR. PM and ID of chamber-A is better than other chambers B and C at 1400 bar injection pressure with EGR mode. Finally, it is concluded from the simulation results that the chamber-A at 1000 injection pressure with EGR mode has a better performance and emission characteristics than other combustion chambers.
- SFC of STDP simulation is 3.6% to 10% lower compared to experimental results with and without EGR operation. SFC of experimental value of CH-A is nearly 0.5–2.8% higher compared to simulation results at with and without EGR.
- BTE of CH-A (EXP 15% EGR) is nearly 2–8% lower compared to CH-A (SIM-15%EGR). BTE of CH-A (EXP 0% EGR) is nearly 4–7% lower compared to CH-A (SIM-0%EGR).
- NO$_X$ of CH-A (EXP 0% EGR) is nearly 1.4–1.7% lower than STDP (EXP-0%EGR). NO$_X$ of CH-A (EXP 15% EGR) is nearly 15–25% lower to STDP (EXP-15% EGR).
- The NO$_X$ emission decreased by 22–60% in CH-A, when 15% exhaust gas recirculation was used.

- For better performance and emission reduction, the injection pressure, better bowl geometry and exhaust gas recirculation should be optimized for diesel fueled CI engine.

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