INVESTIGATION OF DIESEL ENGINE FOR LOW EXHAUST EMISSIONS WITH DIFFERENT COMBUSTION CHAMBERS

by

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Upcoming stringent Euro-6 emission regulations for passenger vehicle with better fuel economy, low cost are the key challenges for engine development. In this paper, 2.2L, multi cylinder diesel engine have been tested for four different piston bowls designed for compression ratio of CR 15.5 to improve in-cylinder performance and reduce emissions. These combustion chambers were verified in CFD at two full load points. 14 mode points have been derived using vehicle model run in AVL CRUISE software as per NEDC cycle based on time weightage factor. Base engine with compression ratio CR16.5 for full load performance and 14 mode points on engine test bench was taken as reference for comparison. The bowl with flat face on bottom corner has shown NO_x emission reduction by 25% and 12 % NO_x at 1500 and 3750 rpm full load points at same level of Soot emissions. Three piston bowls were tested for full load performance and 14 mode points on engine test bench and combustion chamber C has shown improvement in thermal efficiency by 0.8 %. Combinations of cooled exhaust gas re-circulation and combustion chamber C with geometrical changes in engine have reduced exhaust NO_x , soot and CO emissions by 22 %, 9 % and 64 % as compared to base engine at 14 mode points on engine test bench.

Key words: *piston bowl geometry, spray angle, spray hitting plane, emission*

Introduction

In Europe, Asia, and Latin America auto market, the share of diesel powered cars has increased over the last decade. In view of increasing demands on fuel economy, increase of fuel prices and increase of the global warming, there is need to cut CO_2 emissions from transportation. The main drawback of the traditional diesel engine has higher NO_x and particulates matters (PM). However, the difference in emissions between diesel and gasoline engine decreased with each new European legislation level. By 2014, Euro 6 standard brought an almost complete convergence of diesel and gasoline emissions [6] as shown in fig.1. One notable new technology is the diesel particulate filter (DPF) which is now used in practically all in diesel cars to meet present emission norm. It drastically reduces the particulate emissions, but it needs to be regenerated to remove the trapped soot particles and in low load operation the exhaust gas temperatures are

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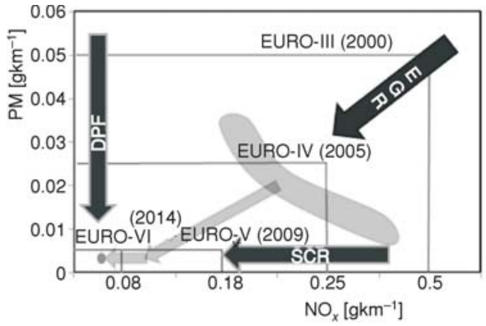


Figure 1. European emission regulations for passenger cars application

often too low for regeneration. This is often done by a post injection in the very late expansion stroke [4]. As this injection provides only heat and no work, it decreases the fuel efficiency of the engine. It is therefore of great importance to maximize the interval between regenerations. The need for in-cylinder control of PM is thus not reduced by using a DPF. Combustion system measures to reduce engine out soot emissions are still of great importance to improve fuel economy. Another new technology is the selective catalyst reduction (SCR) which is now required for diesel vehicle to meet Euro 6 NO_x emission limits. On-road NO_x conversion efficiency of SCR is about 70 to 75%. The combustion system measures to reduce engine out soot and NO_x emissions are still of great importance to improve fuel economy.

This paper describes the development of the combustion system through combustion chamber re-design and calibration strategy suggested for diesel engine to comply Euro 6 emission norms. The focus is on the combustion chamber geometry, the injector nozzle configuration and calibration strategy to reduce in-cylinder engine out emissions. These have a great influence on the combustion process and then the formation of emissions and on the thermodynamic efficiency of the engine. A classic dilemma in diesel combustion development is the trade-off between soot and NO_x. Decreasing one of these emissions usually leads to an increase in the other. This is explained in textbooks by the different temperature dependencies of these emissions. NO_x formation is promoted by high local temperatures, conditions during which soot is oxidized [8]. This is, however, a simplified description that does not account for effects of *e. g.* chemistry and turbulence. Soot and NO_x concentrations apparently develop in quite different ways as function of time. The NO_x curve is dominated by a steep increase in the early parts of the cycle. Here the so-called thermal (or Zeldovich) mechanism is the dominating source of NO_x. Mixture that burns early is later compressed to higher temperatures and has plenty of time to form NO_x via this relatively slow mechanism. Thermal NO_x is formed in stoichiometric or slightly lean mixtures. The soot curve, on the other hand, shows both a steep increase in the beginning of the cycle and a slower decrease during the expansion stroke. This is because soot is first formed in fuel-rich zones under high temperature conditions, and later oxidized in leaner zones. There seems to be at least two possible methods to decrease soot independently of NO_x. First we note that soot and NO_x are formed in different portions of the charge. If the richest portions of the charge were removed, soot production would decrease independently of NO_x production. Secondly note that the soot formed is consumed in the late cycle. If the late cycle oxidation were enhanced, soot emissions would decrease independently of NO_x. The first method is employed in low temperature combustion (LTC) concepts [1, 9, 11]. A long ignition delay allows a longer mixing period, thus decreasing the peak local equivalence ratios. In these systems, long ignition delays are produced using high levels of cooled exhaust gas re-circulation (EGR). This also decreases NO_x, but it often slows down the late cycle burn and can lead to poor efficiency. Another way of reducing the peak equivalence ratios is to use smaller diameter orifices in the nozzle, enhancing the air entrainment rate into the spray. There is also a practical limit to the minimum orifice diameter caused by the increased risk for deposits [10]. Still, both of these technologies are gradually introduced in modern Diesel engines. The second method to decrease soot emissions is to enhance the late cycle oxidation. This involves the fluid mechanic processes in the cylinder and, thus, modifications to the combustion system geometry. These aspects will be discussed in the remaining parts of this section. A typical heat release rate in a light duty diesel engine has a long tail after the main heat release, well after the end of injection. This is the slow, mixing-controlled burn of combustion products formed earlier in the cycle, *e. g*. soot. If this tail were shifted toward top dead centre (TDC), the thermodynamic efficiency would increase. Emissions would also decrease, since enhanced oxidation permits less partially oxidized products to survive into the exhaust port. Enhancing the late cycle mixing rate is therefore a promising method for reducing both soot emissions and fuel consumption. A combustion system providing an enhanced late-cycle mixing rate is therefore likely to be more suited for implementing LTC as well as improving traditional diesel operation. Enhancing the late-cycle mixing rate involves producing turbulence during expansion. It has been shown that swirl-supported direct injection diesel combustion systems are capable of doing this [11]. The mechanism is based on the interaction between swirl and injection, which stratifies high-swirling and low-swirling fluid in the bowl, generating shear forces and turbulence [12]. Although the physics of this process have been described elsewhere, a brief account will be given here as it supports the analysis of our results.

Piston cavity design and spray hitting plane optimization

Design of piston bowl geometry plays a vital role in reducing exhaust emissions and improving fuel consumption at source. Lip shape, spray impingement, bottom corner radius, bowl throat diameter, pip and depth of the bowl have critical role and its influence on the combustion efficiency engine. Figure 2 shows the combustion chamber lip dimensions, spray impingement region, bowl corner region and other critical bowl parameters, respectively [13].

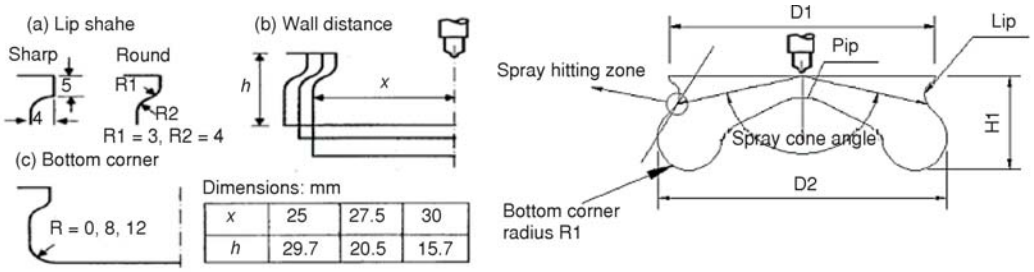


Figure 2. Critical parameters of combustion chamber design

The effect of the lip shape on the exhaust emission is significant on unburned HC emission. At low loads, there is no significant improvement in smoke emission, however at high loads, smoke emissions decreases with a round and sharp lip at the cavity entrance as compared to cavity without lip. In modern high speed diesel engine, the injection timing changes with the engine speed, thereby keeping the position of spray impingement on the cavity wall approximately constant. However the position changes significantly from start to end of impingement and the impinging position on the cavity wall has a significant effect on air fuel interaction which determine the combustion process. The position where the spray impinges on the cavity wall has a significant effect on the fuel distribution especially if the cavity is the re-entrant type. A sharp squish lip shape can reduce the smoke without increasing NO_x emission in a typical combustion chamber, but smoke reduction capability strongly depends on the interaction with

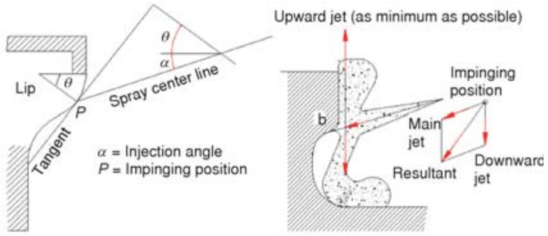


Figure 3. Fuel spray jet hitting direction and movement

other parameters. However, with increasing injection pressure in common rail injection system, reduced throat diameter could cause too much fuel to impinge on a combustion chamber wall, which increases THC emissions due to wall wetting. Various other bowls parameters such as pip height, bowl depth, corner radius, and bowl throat diameter are other critical parameters for effective design of combustion chamber for better emissions and performance [14].

Figure 3 shows the lip zone spray hitting direction and spray travel and the spray path in the bowl. For efficient bowl design, the spray hitting zone needs to be defined in such way that the lip zone to be design should divert the upward spray travel as minimum and downward jet travel should be maximum to get mixing well with fresh air.

Typical thumb rule is spray hitting plane should be in the bowl at minimum 45% of bowl depth from top face of piston to dynamic conditions to avoid portion of fuel spray moving outward in dynamic condition. Design of bowl and hitting plane should minimize the upward spray quantity. In addition to this, bowl swirl is a vital parameter in deciding the mixing rate of air and fuel as well as ignition delay period. Number of holes from the nozzle depends upon the injection quantity required for full load and spray should evaporate before mixing. Increase in injection pressure helps in reducing droplet size and fast evaporation; however increase in spray penetration lead to spray-wall interaction.

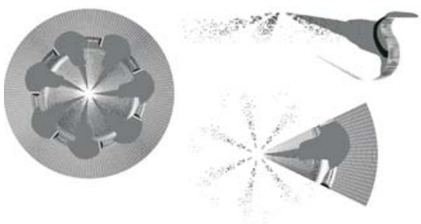


Figure 4. Injector holes and swirl matching through CFD analysis

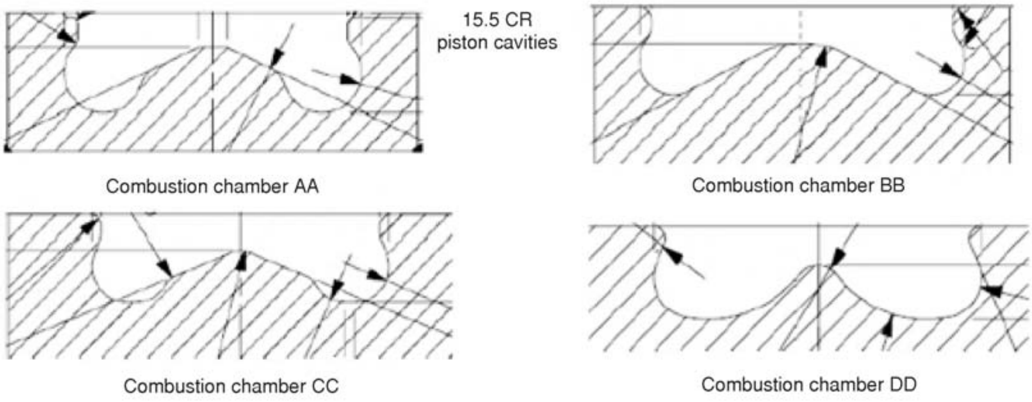
The CFD simulation of spray layout is shown in fig 4. From CFD [3] simulation study, experimental engine cylinder port mean swirl was reduced from 2.05 to 1.9 and nozzle number of holes and spray cone angle changed from six to seven holes injector and 152º to 148º to match spray-bowl with increase in hydraulic through flow by 15% and hole size reduced by 10% as compared to the base engine.

Piston bowls used for experiments

Four different piston cavities with compression ratio CR15.5 were designed and detailed CFD analysis has been carried out for NO_x and PM reduction. Refer tab. 1 for critical parameters of piston cavities designed for experimental engine.

Table 1. Critical parameters of four piston cavities designed for experimental engine testing

	New bowls (CR 15.5)			
Base bowl (CR 16.5)	CC-A	CC-B	CC-C	CC-D
0.815	0.823	0.827	0.827	0.827
140	130	125	136	136
0.61	0.62	0.66	0.64	0.64
0.55	0.56	0.62	0.6	0.59
0.2	0.206	0.178	0.182	0.176
1.11	1.1	1.08	1.06	1.08



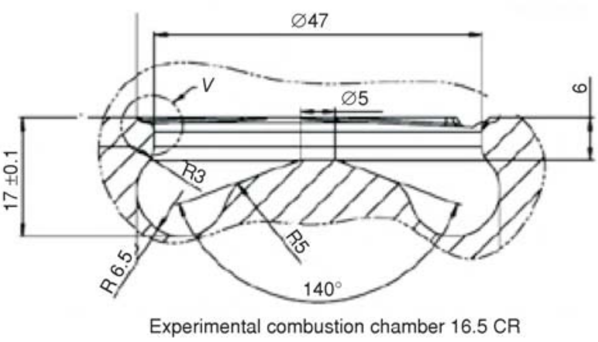


Figure 5. Different piston cavities with CR15.5 for experimental engine testing

Refer fig. 5 for experimental engine piston cavities with different shapes with CR 15.5.

Experimental set-up and testing

The 2.2 litre, four cylinder, 4-stroke diesel engine having 4-valves/cylinder and common rail fuel injection was selected for experiments. Engine was coupled to an eddy current dynamometer for all experiments. Engine experimental set-up is shown in fig. 6 and specifications in tab. 2 which includes various measuring units and connections. Un-cooled pressure sensor with range of 0-250 bar was installed into the combustion chamber to record the in-cylinder pressure history for each test condition at resolution of 0.1 crank angle degrees. Installation of four pressure sensors gives understanding of all four cylinders. A crank angle encoder was fixed on crank shaft to sense the top dead centre (TDC) position. Adequate care has been taken to position the amplifier to minimize the fluctuations in signal noise during high speed data acquisition. The information of in-cylinder pressure *vs*.

Table 2. Experimental engine specification

Engine type Mahindra make	Base engine 2.2 L, inline, 4 cylinder, DOHC, HSDI Diesel specifications
Compression ratio	Base 16.5 : 1 and modified 15.5 :1
Rated power	103 kW
Max torque	330 Nm
Rated speed	3750 rpm
Injection system	Common Rail, Bosch Gen 2
After-treatment system	DOC+cDPF
Emission level	Euro 5
Vehicle reference mass	1860 kg
Dynamic rolling radius	0.331 m

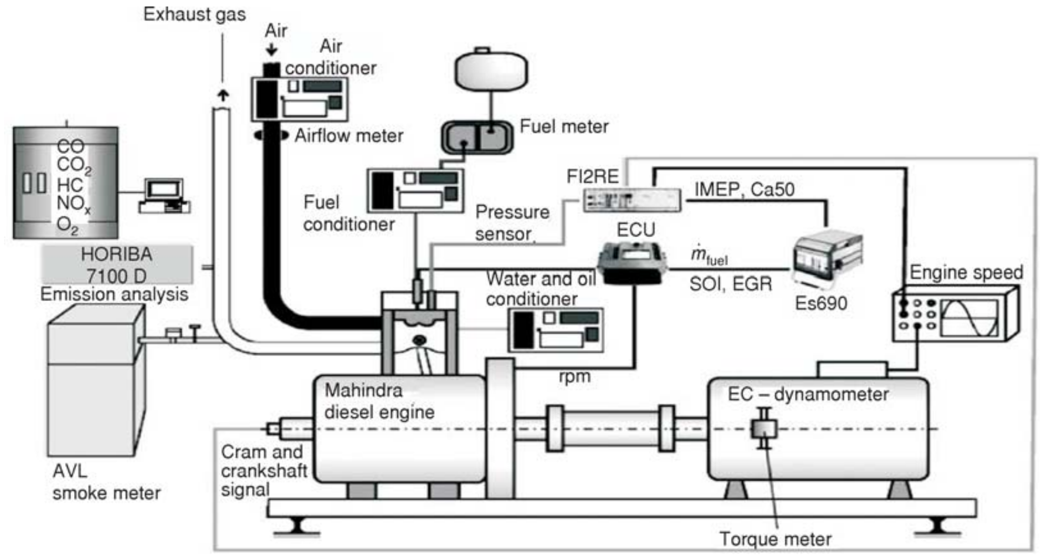


Figure 6. Schematic of experiment set-up with measurement system

crank angle was recorded in high speed data acquisition system. ABB sensy flow meter (hot wire anemometer) with a measuring accuracy 0.5% was used to measure the air flow rate. Other test conditions maintained as per ISO: 8178.

Experimental engine with CR 16.5 was taken as reference for comparing performance of newly designed piston cavities with compression ratio 15.5 based on 3-D, CFD analysis. Three piston cavities have been selected for full load performance viz. combustion chamber A, C, and D. Combustion chamber A is replica of experimental engine combustion chamber except cavity volume have been adjusted such that compression ratio changes from CR 16.5:1 to CR 15.5:1. The experimental engine with combustion chamber A was tested for full load performance and main injection timing was adjusted such that power was same as experimental engine with CR 16.5:1. All other combustion parameters and other engine hardware were kept same for testing of experimental engine with combustion chamber C and combustion chamber D during full load performance test.

Table 3. 14 Modes steady-state points derived from NEDC cycle

Points	Speed [rpm]	Load [Nm]	Weightage factor [%]	Power [kW]
1	1000	4.9	29.1	0.4
2	1704	56.7	10.1	10.1
3	2038	14.8	8.3	3.2
4	1811	16.7	7.2	3.2
5	1375	34.8	6.1	5.0
6	1261	23.9	5.1	3.2
7	1711	110.9	4.5	19.9
8	2005	112	3.8	23.5
9	1430	91.9	3.7	13.8
10	1004	50.3	3.3	5.3
11	1152	77.2	2.5	9.3
12	2295	137	2.2	32.9
13	2210	199.1	1.7	46.1
14	2048	72.1	1.2	15.5

Influence of combustion chamber geometry has been studied for full load performance (1000 rpm to 3750 rpm and 14 mode steady state points which were derived from NEDC cycle as shown in tab. 3. All three combustion chambers were selected based on CFD study (CC-A, CC-C and CC-D) and also extensively tested to understand sensitivity of combustion to injector tip protrusion with ±0.3 mm at same operating condition. The 14 modes points resemble the complete NEDC cycle with time weightage factor has been derived from vehicle CRUISE model [2] run with NEDC test cycle is shown in table.

Results and discussion

CFD results of four combustion chambers were compared at two full load point viz 1500 rpm and 3750 rpm, respectively, for cumulative NO and soot formation.The comparison of NO and soot trends at full load of all the four combustion chambers is shown in figs. 7 and 8 at 1500 rpm and 3750 rpm, respectively. At 1500 rpm full load point, the higher soot (600% higher than combustion chamber A) from combustion chamber D was mainly due to poor combustion leading to lower combustion temperature which generated 43% lower NO. In other three combustion chambers soot emissions are almost same, however combustion chamber A soot was lower as compared to combustion chamber B and C. Combustion chamber C has shown 25% lower NO as compared to combustion chamber A. The combustion chamber C has shown the benefits of same soot and 25% low NO as compared to base combustion chamber A.

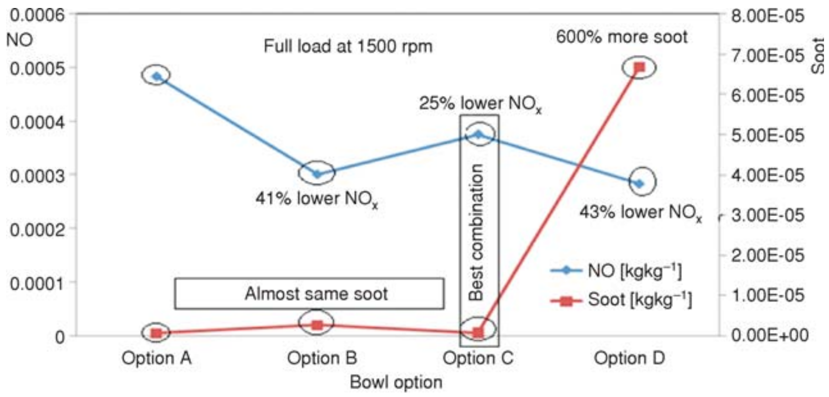


Figure 7. Cumulative NO comparisons and soot of different combustion chambers at 1500 rpm full load point

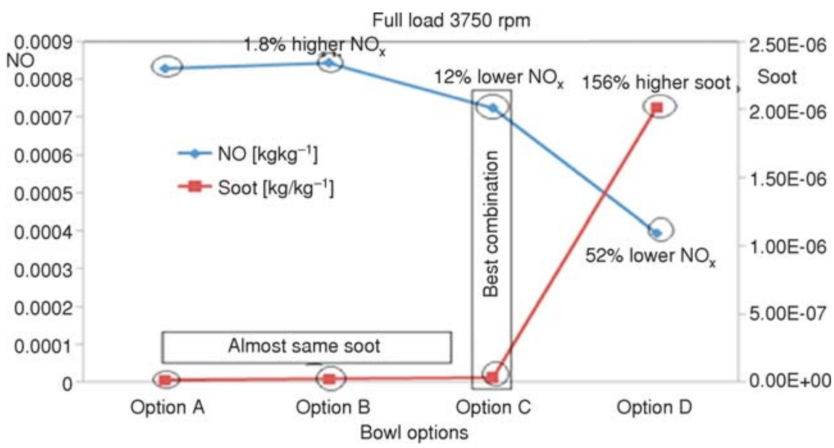


Figure 8. Cumulative NO comparisons and soot of different combustion chambers at 3750 rpm full load point

At 3750 rpm full load point, higher soot (156% higher than combustion chamber A) from combustion chamber D was mainly due to poor combustion and low combustion temperature reflected in 52% lower NO. In other three combustion chambers soot emissions are almost same, however in combustion chamber A, soot was lower as compared to combustion chamber B and C. At 3750 rpm, combustion chamber C has shown 12% lower NO as compared to combustion chamber A. The combustion chamber C has shown the benefits of same soot and 12% lower NO as compared to base combustion chamber A.

In part load condition , NO_x exhaust emissions at engine operating region from 1000 to 2000 engine rpm and 25% load to 60% load without EGR were compared between combustion chamber A and combustion chamber C as shown in fig. 9. Experimental engine results show that combustion chamber C is generating lower NO_x exhaust emission as compared to bowl A. Main region for lower NO_x is due to improved mixing reduces high temperature luminous flame zones. The results indicating that the long travel of spray helped to improve the mixing characteristics so that to reduce diffusion combustion.

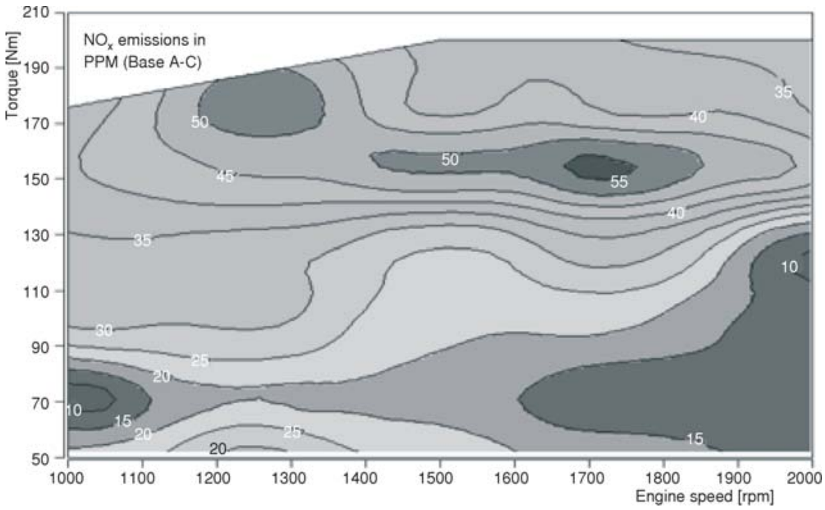


Figure 9. NO_x emission (PPM) difference with combustion chamber A and C

Figure 10 shows full load power, torque, brake specific fuel consumption (BSFC) and smoke comparison of engine with different combustion chambers. Combustion chamber C has shown better torque curve as compared to all other combustion chambers between 3000 to 4000 rpm by 4.5 Nm (1.75%) and power by 1.7 kW (1.7%) as compare to CR 16.5:1 base engine. The observations are in very well agreement with 3-D, CFD simulation output. The effect of combustion chamber shape have slowed down the premixed phase of combustion thereby reduces the work done during compression and reduces diffusion combustion which in turn improves the efficiency and power. Fuel consumption of experimental engine with combustion chamber C and CR 16.5:1 matches throughout the engine speed range except between 1000 to 1200 rpm. BSFC of combustion chamber C is inferior by 3 g/kWh due to lower compression ratio leading to poor combustion at lower speeds. In bsfc comparison of piston cavity A and C with same compression ratio (CR 15.5:1), Piston cavity C is superior at all speeds and better by 4.5 g/kWh at lower speed between 1000 to 1500 rpm and 2-3 g/kWh better between 2000 to 3750 rpm as

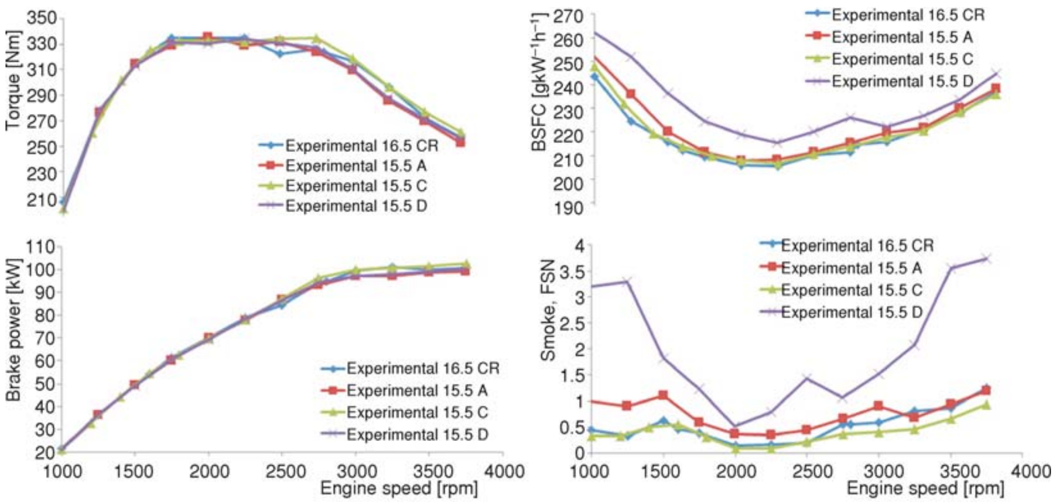


Figure 10. Full load performance comparison of engine with different combustion chambers

compared to combustion chamber A. The BSFC of piston cavity D is much inferior as compare to combustion chamber C throughout engine speed range by 8 to 12 g/kWh and these observations were closely in agreement of 3-D, CFD observations. Experimental engine with combustion chamber C is showing lower most smoke among all combustion chambers indicating better combustion happening due to its shape and correct matching of swirl and injector spray hitting. With same compression ratio (CR 15.5:1) smoke level of combustion chamber C is less by 0.4 to 0.7 FSN throughout the speed range as compared to combustion chamber A indicating better combustion happening as estimated in 3-D simulation study. Smoke generated with combustion chamber C is even lower than compression ratio 16.5:1. The design features of combustion chamber have improved the NO_x-PM trade-off.

Figure 11 shows thermal efficiency of experimental engine with combustion chamber of CR 16.5:1, combustion chamber of CR 15.5:1 A, C, and D. Thermal efficiency of combustion chamber with CR 16.5:1 is best amongst all combustion chamber. With same compression ratio (CR 15.5:1), thermal efficiency of combustion chamber C is better than combustion chamber A by 2.5% at lower speed and 0.5% at higher speed. Thermal efficiency of combustion chamber D is lowest. In bowl C, squish influence is higher, which help and improve the fuel-air interaction and subsequently reduces the diffusion combustion which is major lead for smoke formation. As the engine speed goes higher, the time available for squish interaction with fuel and fresh air in the bowl bottom zone reduces.

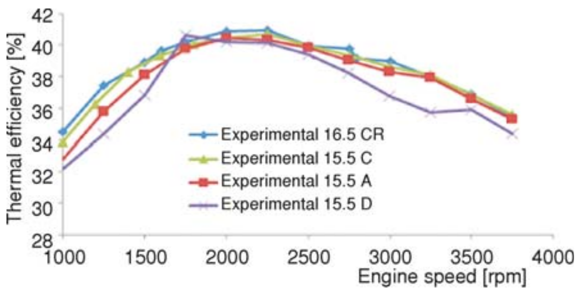


Figure 11. Comparison of brake thermal efficiency of engine with different combustion chambers at full load performance

The summary of engine out exhaust emission results of 14-modes of NEDC cycle converted in to g/hr and g/km are shown in figs. 12 and 13 with different combustion chambers. Experimental engine with CR 15.5:1 and combustion chamber C at 14 modes points, overall

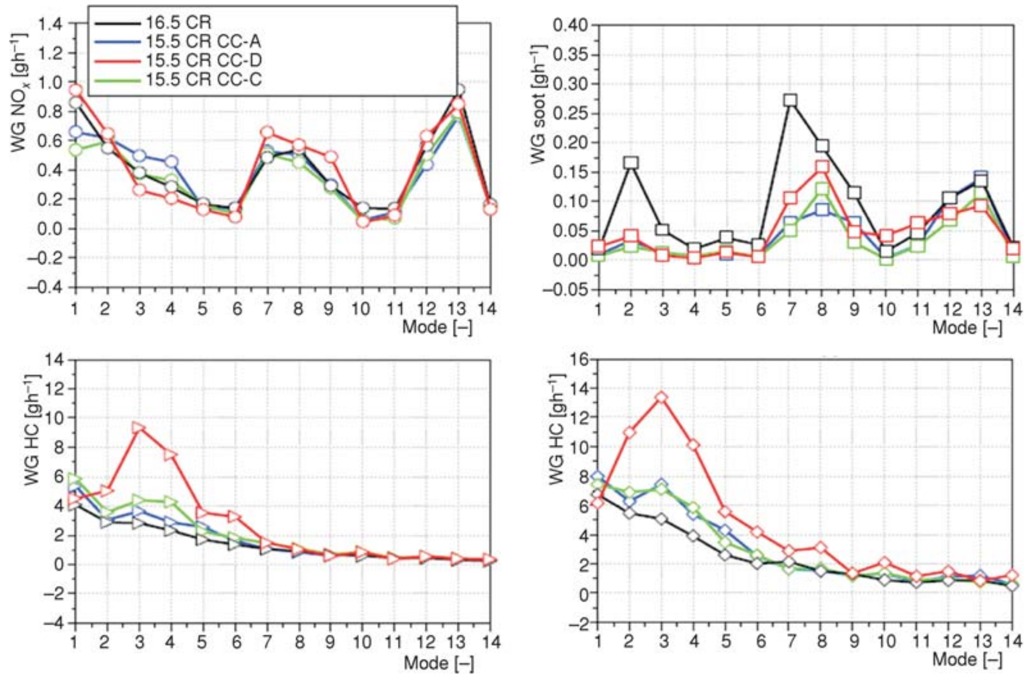


Figure 12. 14 mode emission comparison of engine with different combustion chambers
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engine out soot emissions has been reduced from 0.037 g/km to 0.013 g/km (64%), CO from 1.43 g/km to 1.310 g/km (9%), and NO$_x$ from 0.207 g/km to 0.161 g/km (22%). If we assume

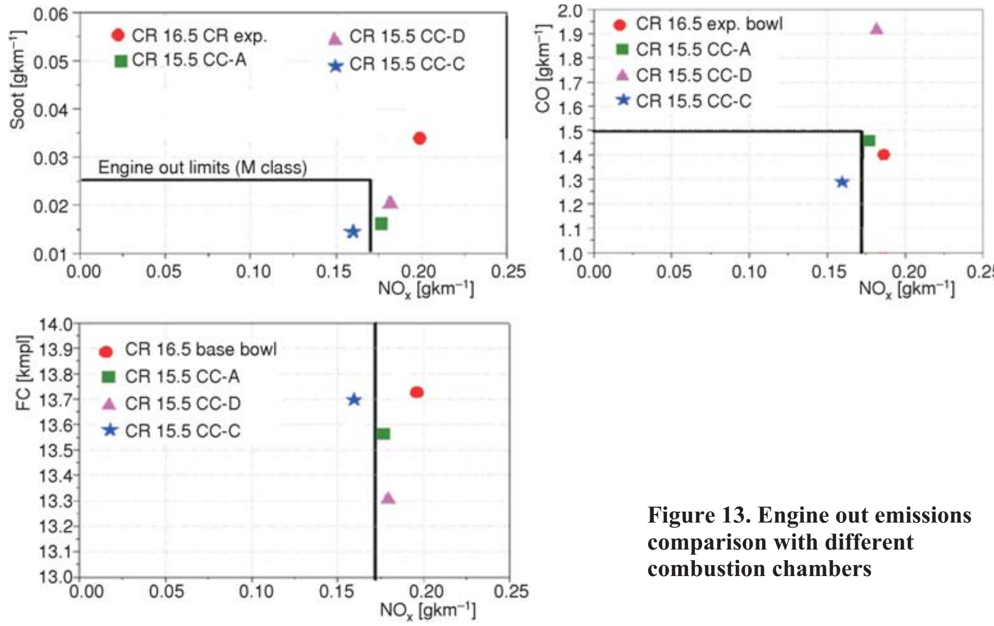

Figure 13. Engine out emissions comparison with different combustion chambers

DPF (90% soot conversion efficiency) and SCR (70% NO_x conversion efficiency) is fitted with this engine, exhaust soot and NO_x emissions would be clearly comply Euro-6 emission limits.

Conclusions

Three different bowl geometries were extensively studied for performance, emissions and fuel economy benefits. After detailed experimental investigation of bowl A, B and C, the following conclusions have been drawn.

With correct piston bowl geometry, injector spray hitting plane matching shows that engine with combustion chamber C, NO emissions were less by 25% and 12%, respectively, with the same level of soot emissions at 1500 rpm and 3750 rpm full load points, respectively. The flat face at the bottom of the combustion chamber makes difference in mixing phenomenon, which has shown significant benefits in reduction of full load smoke and exhausts emissions. In CR 15.5:1 engine, at full load performance points, thermal efficiency matches at lower speed between 1000 rpm to 2000 rpm with base engine and seen more by 0.8% between 2000 to 3750 rpm with combustion chamber C. Experimental engine with CR 15.5:1 and combustion chamber C at 14 modes points , overall engine out soot emissions has been reduced from 0.37 g/km to 0.013 (64%) , CO from 1.43 g/km to, 1.310 g/km (9%) and NO_x from 0.207 g/km 0.161 g/km (22 %), respectively. If we assume DPF (90% soot conversion efficiency) and SCR (70% NO_x conversion efficiency) is fitted with this engine, exhaust soot and NO_x emissions would be clearly comply Euro-6 emission limits.

Combustion chamber play vital role in achieving better combustion, in conjunction with correct spray hitting plane, number of hole in nozzle, swirl, boost, injection timing, injection pressure, pilot injection strategy and boost pressure also plays important role in achieving better combustion and reducing exhaust emissions . Hence combustion development requires overall parameters optimisation to gain benefit in fuel consumption and exhaust emissions.

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