COMBUSTION CHAMBER GEOMETRY AND FUEL SUPPLY SYSTEM VARIATIONS ON FUEL ECONOMY AND EXHAUST EMISSIONS OF GDI ENGINE WITH EGR

by

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In this study, the combustion chamber geometry for spray-guided, wall-guided, and air-guided combustion strategies were fabricated. The piston crown shape and the cylinder head in each combustion chamber geometry was machined by fixing the fuel injector and spark plug at proper positions to obtain swirl, turbulence, and squish effects for better mixing of fuel with air and superior combustion of the mixture. Conducted tests on all the three modified gasoline direct injection engines with optimized exhaust gas recirculation and electronic control towards fuel injection timing, the fuel injection pressure, and the ignition timing for better the performance and emissions control. It is clear from the results that NO_x *emissions from all three combustion modes were reduced by 4.9% up to 50% of loads and it increase for higher loads due to increase of in-cylinder pressure. The fuel consumption and emissions showed better at 150 bar fuel injection pressure for wall-guided combustion chamber geometry. Reduced HC emissions by 3.7% and 4.7%, reduced CO emissions by 2% and 3.3%, reduced soot emissions by 6.12% and 10.6%. Reduces specific fuel consumption by about 10.3% and 13.3% in wall-guided combustion strategy compare with spray-guided and air-guided combustion modes respectively.*

Key words: *gasoline direct injection combustion, piston crown shape, combustion chamber modification, spark plug position, fuel injector position*

Introduction

To achieve the current regulations of emissions norms globally, it is more challenging task for the automotive manufacturers. Other than using a three-way catalytic converter, particulate matter filter, and selective catalytic reactor, it is required to minimize the concentrations of harmful emissions coming out from the exhaust of the gasoline direct injection (GDI) engine through precise control of various engine control variables. The engine control variables include ignition timing, fuel injection (FI) timing, FI duration, fuel injection pump (FIP), split injection strategy, compression ratio, and majorly combustion chamber modification includes piston crown profile, spark plug, and fuel injector positions.

The CFD analysis shows that the lower FIP are better towards a combustible mixture near the spark plug tip. The FIP above 200 bar results in misfiring. Increase in the combustion ratio (CR) from 10:1 to 12:1 results in decrease of turbulence kinetic energy (TKE) by

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70%, decrease of throttle regulation by 49%, increase of indicated mean effective pressure (IMEP) by 4%, increase of NO_x emissions by 50%, and the HC emissions are deeply reduced [1]. Deep contraction in particulate mater (PM) and particle number (PN) concentrations with FIP up to 172 bar, further increase in FIP, the PN concentration was boosted up due to a nominal increment of nucleated fuel particles [2-4]. With the increase of combustion rate due to better spray atomization, the thermal efficiency of an engine increases [5]. Extreme brake power and brake torque are achieved at end of injection (EOI) 120° bTDC but specific fuel consumption (SFC) was decreased. The CO_2 emission is decreased at low speeds but higher at high speed [6]. The high peak pressure, temperature and NO_x emission were reduced by using external dilution and two pulse injections [7].

Optimized FI and ignition timings, FIP are required to diminish the concentrations of PM emissions from an engine exhaust. Early FI, higher FIP and warm intake were results in reduction of particulate emissions [8]. A six-hole fuel injector is utilized with fuel spray cone angle of 10°, located below the intake ports such that one of the spray plumes is pointing the spark plug tip. Results showed that heat release rate and the average in-cylinder temperature were reduced with an increase in exhaust gas recirculation (EGR) percentage, thereby reduced NO_x emissions [9].

Four modified piston crown shapes are examined namely offset piston-offset bowl and offset scoop, flat piston-offset bowl, and tapered piston-offset bowl. Better in-cylinder flows and higher heat release rates (HRR) are obtained for the offset pentroof with offset bowl (OPOB) piston. Lower HC and CO emissions, and higher NO_x emissions compared to that of other piston modifications [10]. Engine control unit software is updated without any tune up/modification in the hardware to improve an engine efficiency and reduce emissions [11, 12]. The CFD analysis on the effect of FIP with injection control strategy towards the mixing characteristics was studied with different FIP of 90, 120, 150, and 180, and observed that the in-cylinder pressure, HRR and the ITE were higher at 180 bar FIP. The NO_x emissions are higher with split injection mode at higher FIP due to homogeneous mixture [13]. Wall wetting of fuel over the CC surface was found to be tiniest by maintaining the FI timing very close to the middle of suction stroke. The air-flow vortex formed inside an engine cylinder was sufficient to disperse the fuel droplets, therefore it reduces the percentage of fuel impingement over the CC wall surface [14].

Developed a driver circuit for ignition dwell timing control using Arduino controller with trigger IC and CAM position sensor [15]. The DI engine was analyzed using CFD with flat and bowl shape piston crown profiles at wide-open throttle under non-firing conditions. Bowl shaped piston crown profile shows better results towards mixture characteristics because of uphold of tumble effect for wider period than flat crown shape [16].

The *K*-type thermocouples were used to measure the temperature distribution and the heat transfer co-relations over the surface of combustion chamber. Based the temperature distribution over combustion chamber surface and to impart squish and turbulence effects, and to reduce wall wetting, the piston crown part was modified for each combustion mode strategy in GDI engine [17-19]. The fuel control parameters such as multiple injection strategy and the fuel impingement with optimized FIP were results in high turbulence and fast mixture formation over impinging surface at high speeds. Soot emissions tend to reduce without a measurable change in NO_x formation [20-22]. Fuel impingement over the CC surface was found to be minimized for FI timings about at the middle of the suction stroke. Also the air flow vortices formed inside the CC was strong enough to divert the fuel droplets and hence it minimizes the amount of fuel impingement over the CC surface [23].

The main aim of this work is to modify both piston crown surface and cylinder head part, which includes the provisions for spark plug and GDI high-pressure fuel injector (combustion chamber geometry) for achieving reduced emissions and better fuel economy. The modified piston crown shapes such as trapezoidal, comparatively enlarged trapezoidal and a pent roof piston crown with a small bowl on one side of pent roof for spray-guided, air-guided and wall-guided combustion geometry provide a better swirl, turbulence and squish effects. Also, the modified piston crown profiles minimized the wall impingement of fuel over the combustion chamber surface which results in avoidance of rich mixture regions on the combustion chamber surface, thereby reduces CO and soot emissions. Three split injections were maintained to control an engine knock and NO_x emissions.

The GDI combustion profile modifications

Piston crown with spark plug and injector positions

Figure 1(a) shows the modified piston crown shape for the wall-guided combustion mode with the positions of the spark plug and the fuel injector. The piston crown was modified like a pent roof structure and a small bowl on one side of the pent roof of the piston crown. The Fuel injector was fixing through the cylinder head provision such that the injected fuel mixture must slide over the bowl surface and it divert towards the spark plug to initiate the combustion of a mixture. Also, the pent roof structure provides greater extent of turbulence effect of air during compression, results in better mixing of air with injected fuel.

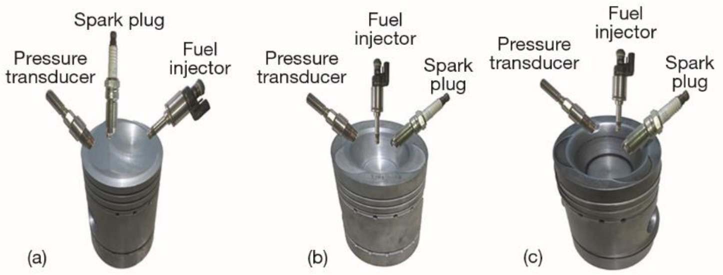


Figure 1. Modified piston crown profiles with positions of spark plug and fuel injector for various combustion mode; (a) wall guided, (b) spray guided, and (c) air guided

Figure 1(b) shows the modified piston crown profile for spray-guided combustion mode with the positions of the spark plug and the fuel injector. The trapezoidal shape bowl on the piston crown provides better swirl and squish effects. The position of the fuel injector is such that the nozzle tip is at the center of the piston bowl, results in the distribution of fuel throughout the volume of a combustion chamber and better mixing of air with fuel. Figure 1(c) shows the modified piston crown profile for air-guided combustion mode with the positions of the spark plug and the fuel injector. A comparatively enlarged trapezoidal bowl was fabricated on the piston crown than spray-guided combustion mode to avoid wall impingement, and to develop better swirl, squish effects. The fuel injector is installed at an angle of 15° to the horizontal through the cylinder head. The injection spray spread throughout the compressed air so that it creates an environment to mix air and fuel properly.

Cylinder head modifications

Cylinder head CADD model

The cylinder head is the most complicated structure in water-cooled IC engines; the modification of this cylinder head requires a proper 3-D cut sectional image. The drawn CAD sketch of cylinder head using CREO modeling software to fix the spark plug with coil on plug, a GDI fuel injector, and the pressure transducer on the cylinder head without any flow restrictions through the water jacket of the cylinder head. Figure 2 shows the cut sectional image of an engine cylinder head for spray-guided combustion chamber geometry includes an internal water jacket cavity and provisions for a spark plug, fuel injector, and pressure transducer.

Cylinder head fabrication

Figure 3 shows the fabricated cylinder heads for wall-guided, spray-guided, and air-guided combustion mode strategies. Each mode includes different locations of the spark plug and the fuel injector. With the help of the cylinder head CADD model, took the measurements for fabricating the sleeves to install the spark plug, fuel injector, and pressure transducer in each case of combustion mode.

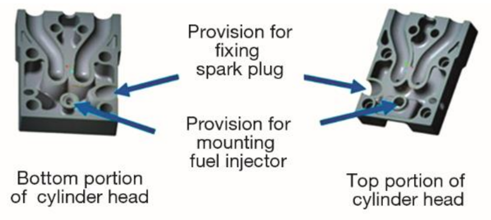


Figure 2. Cut sectional image of an engine cylinder head CAD model

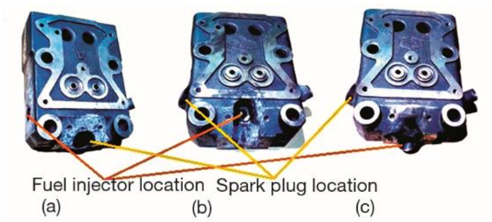


Figure 3. Modified cylinder heads for various combustion modes; (a) wall guided, (b) spray guided, and (c) air guided

Experimental work

The Kirloskar 5HP, water-cooled, 1500 rpm rated speed, stationary four-stroke single-cylinder Diesel engine was chosen for the modification into GDI engine with the similar features 80 mm bore, 55 mm crank radius, 235 mm connecting rod length, and displacement volume of 533 cc. The compression ratio of the modified GDI engine was adjusted to 10:1 while machining each piston crown profile for the type of combustion mode. Table 1 shows the variation of the parameters of Diesel and modified GDI engines.

Table 2 shows the optimized EGR, ignition timing, and fuel injection timings with injection durations for all three combustion modes.

The National Instrument's DIDS fuel injection driver with SCM software was used to control rail pressure, FI timing, FI duration, and split injections. An ignition timing is controlled with a self-developed ignition driver circuit using Arduino Uno board and motor driver circuits. Kistler pressure transducer was used for recording an in-cylinder pressure with a NI-DAQ combustion analyzer system. Using AVL's 444 Digas analyzer and 437C smoke meter, measured the engine-out emissions.

Table 1. Major different parameters of Diesel engine and modified GDI engine

Engine parameters	Values	
	Diesel engine	GDI (modified) engine
Type	Diesel direct injection – compression direct ignition engine	GDI – spark ignition engine.
Compression ratio	16.5 : 1	10 : 1
Injection timing	24˚ bTDC	Three split injections during suction and compression bTDC
Injection pressure	200 bar	100, 125, 150, and 175 bar (for each GDI combustion mode)
Combustion chamber	Hemispherical open type	Trapezoidal type bowl (or) pent roof with small bowl on one side of pent roof

Table 2. Optimized EGR, ignition and fuel injection timings with injection durations for all three combustion modes

Parameters/combustion chamber geometry	Wall guided	Spray guided	Air guided
EGR [%]	10%	9%	9%
Ignition timing, bTDC	10°	9°	11°
Fuel injection timing, bTDC (3 split injections)	320°, 220°, 90°	270°, 110°, 60°	290°, 110°, 70°
Total injection durations [ms]	3.2	3.6	3.7

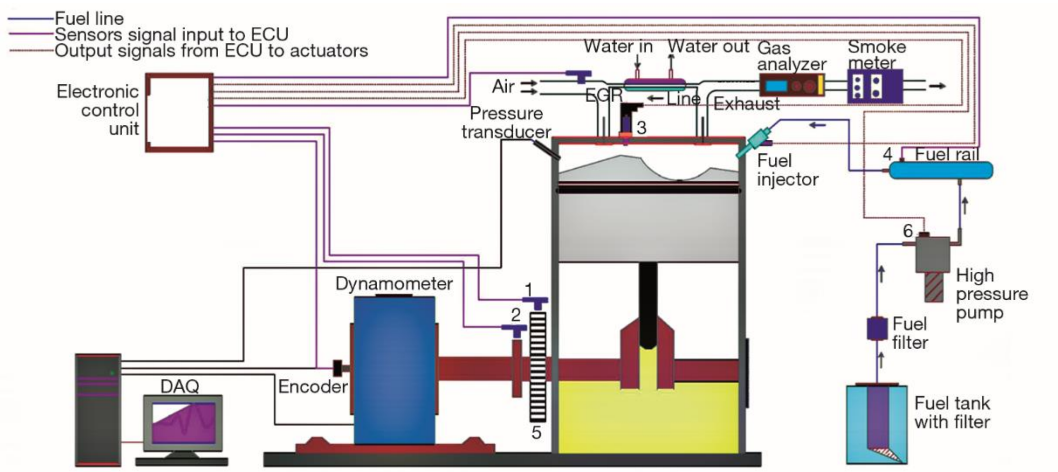


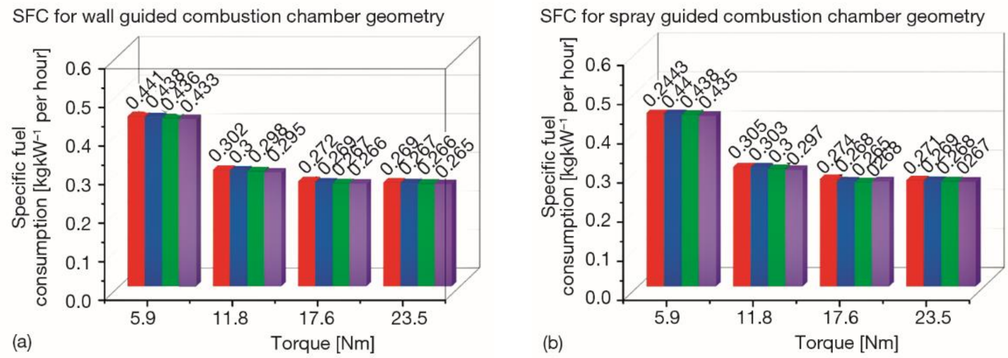
Figure 4. Block diagram of modified GDI engine; *1 – carnk position sensor, 2 – cam position sensor, 3 – spark plug with ignition coil, 4 – rail pressure sensor, 5 – triger wheel, and 6 – inlet meetering valve*

Results and discussion

Conducted fuel economy and emission tests on spray-guided, wall-guided, and air-guided combustion modes of the GDI engine at no load, 25%, 50%, 75%, and 100% loads under optimized operating conditions shown in tab. 2.

Specific fuel consumption variation

Figure 5 show the variations of specific fuel consumption *vs.* torque at different FIP for wall-guided, spray-guided and air-guided combustion chamber geometry of GDI engines. The SFC values were decreased by increasing FIP from 100-175 bar and also by increasing load/torque. Better fuel atomization results in homogeneous mixture formation, which leads to superior combustion of air-fuel mixture. Fuel droplet velocity and the spray cone angle were increased by increasing FIP but mean droplet diameter decreased, which resulted in superior mixture formation so that BSFC reduced at higher FIP. The SFC was reduced in wall-guided combustion chamber geometry when compare with spray-guided and air-guided combustion chamber geometry GDI engines.



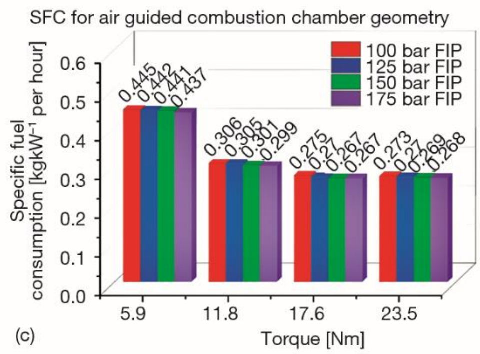


Figure 5. Variation of SFC *vs.* torque at different FIP at specified operating conditions (*for color image see journal web site*)

Emissions measurement

The NO_x emission variation

Figure 6 show the variations of NO_x emission at various loads and FIP. The NO_x emissions were increased with the increase of engine torque, because of rise in in-cylinder temperature. The NO_x emissions are higher at higher FIP, due to boost up of fuel spray droplet velocities with fine atomization which leads to complete combustion and higher peak in-cylinder temperature. Early FI timing leads to longer delay period, results in more time to mix the fuel with air before combustion, however, the burned fuel particles prevail in the CC and the heat discharged from cylinder wall is answerable for in-cylinder temperature; thereby formation of NO_x emission. At lower torque, NO_x concentrations were lower in all three combustion modes, whereas at higher torque, it was higher due to better mixing, complete combustion and rise in in-cylinder temperature. At higher FIP of 150 bar and 175 bar, the NO_x concentrations were higher for wall guided combustion mode.

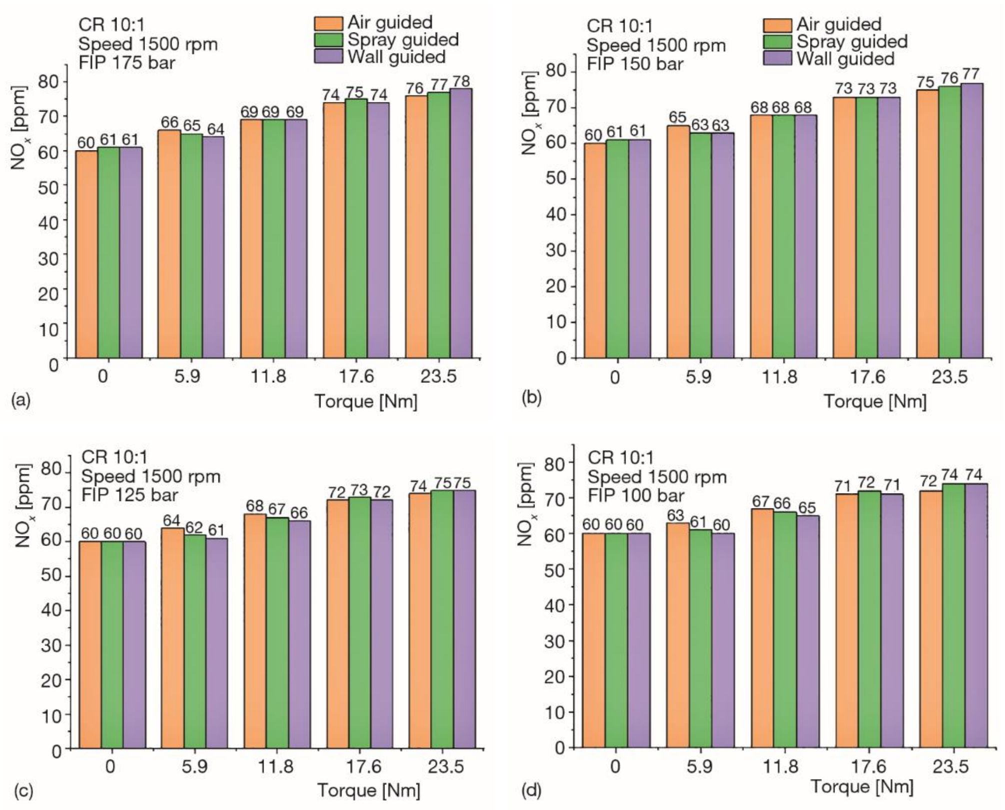


Figure 6. The NO_x emission at various loads and FIP at specified operating conditions
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The UBHC emission variation

Incomplete combustion is the reason for UBHC emission formation. Figure 7 shows the variations of UBHC emission at various loads and FIP. The HC emissions decreased considerably with increase of engine torque, because proportionately quick rise of suction charge flow increases the fuel spray droplet velocities inside the CC which leads to better mixture development, greater extent of complete combustion and combustion efficiency, thereby lower THC emissions. At lower engine loads, THC emissions are higher because of lower peak in-cylinder temperature. At higher FIP, better atomization of fuel results in comparatively fine spray droplets, better mixing and complete combustion which lead to lower THC emissions. The HC concentrations is less in case of wall guided combustion mode than spray guided and air guided combustion modes, because of greater extent of turbulence effect in wall guided mode with pent roof piston crown results in better mixing of air and fuel; thereby complete combustion of the mixture.

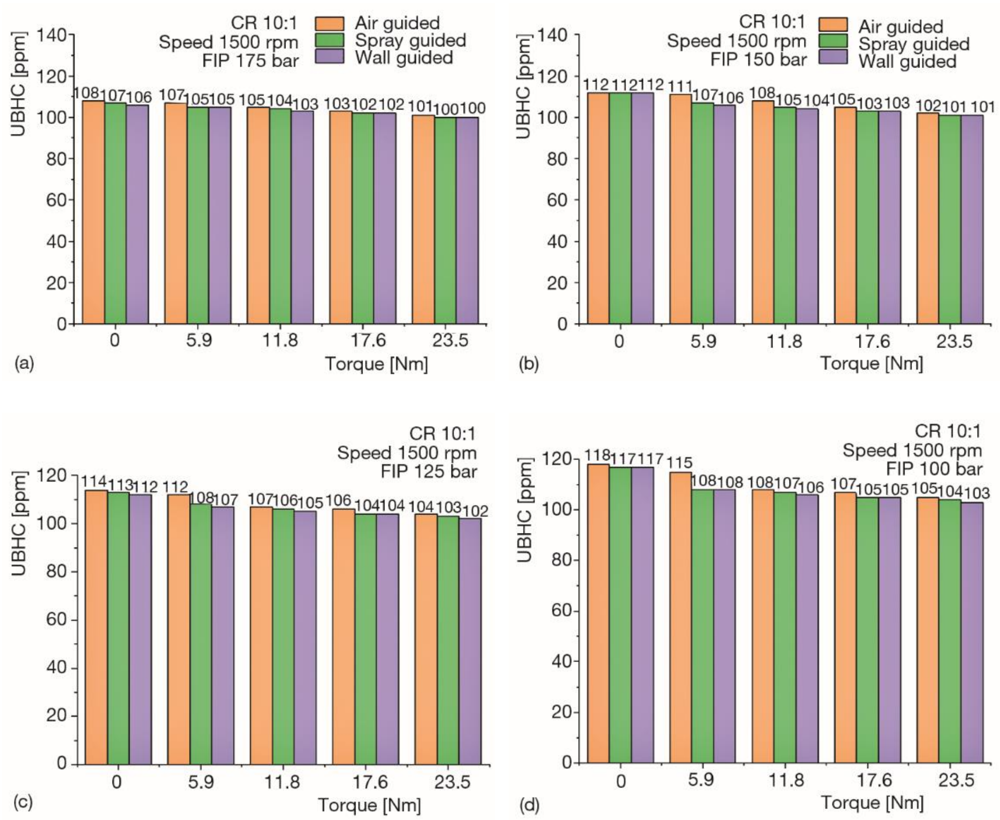

Figure 7. The UBHC emission at various loads and FIP at specified operating conditions
(*for color image see journal web site*)

The CO emission variation

The CO emission is because of lack of O_2 quantity at the time of burning of mixture. Figure 8 shows the variations of CO emission at various loads and FIP. At higher FIP, less CO emission was observed due to finer atomization, improved mixture formation before the initiation of combustion. No cross rich mixture zones in the modified CC volume with diminished wall wetting possibility, further lower flame quenching. The HC and CO emissions were reduced. Also, the early FI timing diminished the rich mixture zones due to dispersion of fuel particles with air-flow vortex during intake stoke. Lower FIP results in week mixture formation and incomplete combustion leads to CO formation. Like HC emission, CO emission is lower for wall guide mode than air guided and spray guided combustion modes for all torque values.

Soot emission variation

Soot emission is due to over rich mixture or moderately evaporated fuel. As discussed in CO emission, early FI timing diminishes the possibility of wall wetting, thereby it reduces the rich mixture zones over the cylinder surface. Figure 9 show the variations of Soot emission at various loads and FIP. Similar to CO emission, soot emission also reduced with an increase of engine torque. Higher FIP results in reduction of soot emission, because the

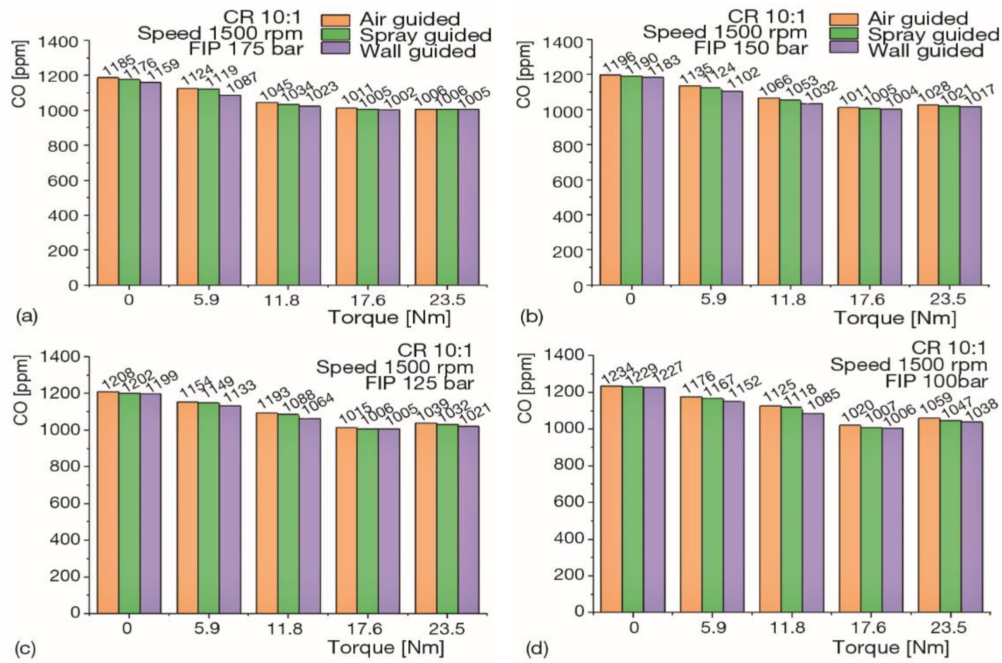

Figure 8. The CO emission at various loads and FIP at specified operating conditions
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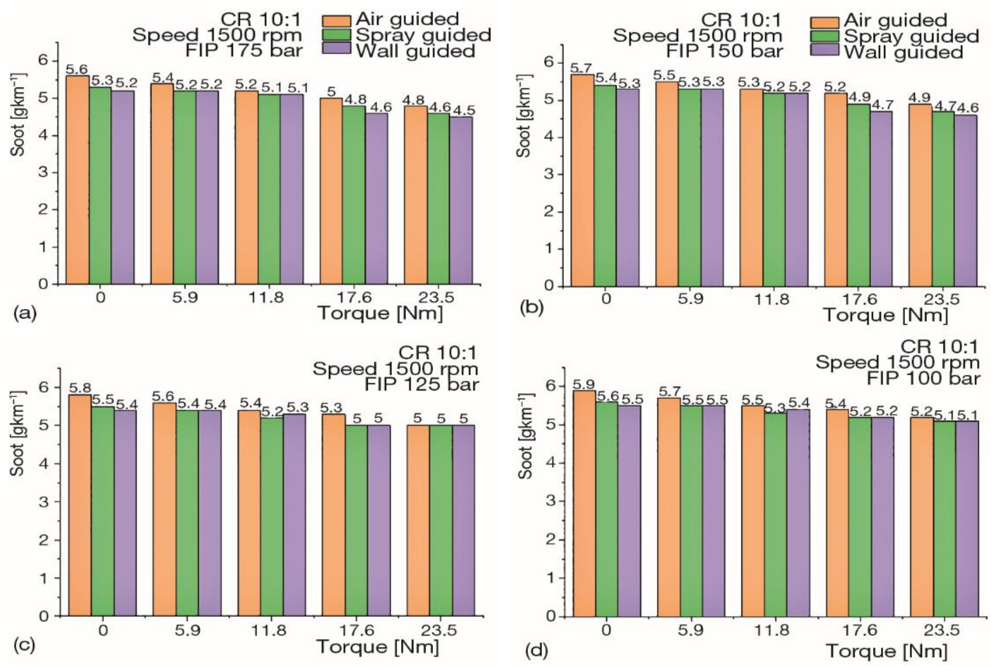

Figure 9. Soot emission at various loads and FIP at specified operating conditions
(*for color image see journal web site*)

higher FIP provide better fuel atomization and likely unvaried dispense of fuel into the CC, thereby no cross-rich zones inside the CC which results in better mixture formation and complete combustion. Also, the wall guided combustion mode shows the reduction in Soot emissions than other combustion modes due to effective turbulence effect with pent roof structure of piston crown which results in better mixing with rich zones.

Conclusions

The GDI engines for spray-guided, wall-guided, and air-guide combustion chamber geometry were tested towards the specific fuel consumption and emission characteristics at various loads under the optimized operating conditions. From the test results, the following points were summarized.

- With the increase of torque/load and FIP from 100 to 175 bars, the SFC was decreased irrespective of the type of combustion mode. The SFC was reduced by about 10.3% and 13.3% in wall-guided combustion mode compare with spray-guided and air-guided combustion modes.
- The NO_x emissions were reduced upto 4.9% for lower and medium loads and it increases towards higher load/torque values. From wall guided combustion mode, HC emissions were reduced by 3.7% and 4.7%, CO emissions were reduced by 2% and 3.3%. Soot emissions were reduced by 6.12% and 10.6% when compare with spray guided and air guided combustion modes.

From the previous points, it is concluded that the wall guided combustion chamber geometry (pent roof piston crown structure and a small bowl on one side of pent roof) GDI engine at 150 bar FIP shows better fuel economy and reduced emissions than other combustion modes.

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