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Effect of Piston Bowl Geometry on Smoke Emission for Two Cylinder Genset Engine

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ABSTRACT

The two cylinder naturally aspirated genset engine with mechanical in-line fuel injection system has an ever growing high demand in Indian market due to its simplicity in design, robust structure and portability; with engine meeting stringent CPCB II norms mandatorily. The approach for attaining smoke norms for diesel genset engine is twofold process; influence of engine parameters on smoke and control of smoke emission through that parameter. The Piston bowl geometry is the best suitable parameter that influences smoke emissions owing to seldom complex change for specific injector nozzle and washer.Four pistons with different bowl geometry, termed as Option Baseline, Option A, Option B and Option C are developed by modifying the parameters typically associated with bowl volume change. The Options are experimentally tested for various nozzle configurations and washer thickness.Results showed that Option A provides the least smoke level of 4.291 FSN while for remaining three Option the smoke level shoots up at 1st mode by a wide margin of 81.05 %, 39.75 % and 25.54 % respectively above 4.291 FSN. The change in injection timing for Option A further yielded the deterioration in smoke level but negligible on comparing with rest Options.

KEYWORDS

FSN, BTDC, Washer, Nozzle configuration

1. INTRODUCTION

Diesel generator outcomes the inefficiency caused by electric generation through thermal power plant, hydraulic power plant, nuclear power plant to cater the lag between electricity production and consumption in the most portable and effective way. In India the trend is to manufacture genset in simplest possible way with advanced technology. Now this trend is accompanied by CPCB II norms imposed by government of India stating that every genset manufactured and sold within country should strictly follow these norms. CPCB II norms were made effective from 1st January 2014. As per these norms the genset is categorized in three divisions as per power rating; up to 19 kW, 19 kW to 75 kW and 75 kW to 800 kW. The norms are

Table 1. CPCB II Emission Norms

Power category	Emission Limit (g/kW-hr)			Smoke limit (per meter)
	NO_X + HC	CO	PM	
up to 19 kW	≤ 7.5	≤ 3.5	≤ 0.3	≤ 0.7
19 kW to 75 kW	≤ 4.7	≤ 3.5	≤ 0.3	≤ 0.7
75 kW to 800 kW	≤ 4	≤ 3.5	≤ 0.2	≤ 0.7

2. LITERATURE REVIEW

The piston bowl configuration has significant effect on combustion during compression stroke [1]. Torroidal shape of piston bowl results in better swirl ratio. The number of holes in injector [2] is crucial in deciding the smoke level. Large number of holes leads to increase in chance of overlapping of sprayed jets causing generation of rich fuel zone area and ultimately rise in smoke level. Simultaneously large number of holes with smaller diameter increases the air utilization during the start of combustion but further the air utilization is poor. The effect of bowl diameter with respect to bore diameter in terms of bowl diameter to bore diameter ratio, the ratio is to be kept minimum to reduce smoke emission also high ratio is acceptable provided that swirl ratio is very high. Lower d/D ratio increases NOx emissions and decreases soot emissions [3], since the combustion is maximum at TDC. Re-entrant type of combustion chamber [4], with narrow entrance creates high swirl ratio and increases air-fuel mixing within cylinder also deep chambers are preferred over shallow. Spray angle and the point of spray hitting the bowl area enhance fuel mixing capability. Lip radius [5], affects the HC and smoke emissions by reducing the fuel flow out of cavity and also prevents thermal stress. Torroidal radius affects the smoke emissions and generates high cylinder pressure.

Bowl diameter affects the mixing of air-fuel, increase in diameter leads to uniform mixing of air with fuel [6]and causing high peak pressure owing to good combustion during late expansion stroke. This deteriorates the performance of engine. The strategy to follow to reduce smoke level and achieve CPCB II norms for engine below 19 kW power ratings include selection and change in combustion bowl geometry and compression ratio [7]. Furthermore selection of injector nozzle, hydraulic through flow, number of holes with optimized NTP. The reduction of high pressure pipe length and diameter leads to reduction injection duration since time require to travel from pump end to injector end and volume flow decreases respectively. Reduction in number of spray holes leads to restriction in flow causing increase in injection duration. Reduction in hole diameter causes restriction to flow increasing injection duration and decrease of droplet size. The centre line of fuel spray must hit the bowl at lip area or at entry to ensure homogeneous mixing and uniform combustion, the centre line is ensured by nozzle tip protrusion. If injection timing is advanced then spray angle is to be decreased and protrusion is to be increased. Narrower piston bowl has high effect on soot formation [9], since it has large unburnt fuel air mixture. Soot formation decreases with increase in centre depth. The lip area is significant since smaller lip area holds lesser quantity inside bowl leading to reduced soot emissions. In presence of high temperature and fuel rich zone fuel originated from hydrocarbons has a strong tendency to form soot particles [10]. In early stage of combustion large amount of soot is oxidized owing to oxygen rich zones. The aromatic hydrocarbon compounds generated are converted in to particles. These particles further coagulate and forms solid soot particles. Torroidal radius [11], significantly influence soot emission, bowl with large torroidal radius provides proper combustion and depress soot accumulation inside cylinder as compared to bowl with smaller torroidal radius. For Mexican hat type piston bowl with large depth [12], NOx formation rose due to proper mixing of fuel and air with proper swirl and high heat generated increased NOx formation while for a shallow piston bowl NOx reduced comparatively. Advanced timing ultimately leads to large time available for fuel injection in ignition delay period. This rises maximum heat release due to mixing of fuel and air in pre combustion phase, raising the temperature and contributing to NOx formation. Early inlet valve closing [13], leads to reduction in soot level while increases exhaust gas temperature. Advancing valve closing to before bottom dead centre leads to reduction in air trapped, furthermore, degrading volumetric efficiency and decreasing the maximum cylinder pressure at end of compression stroke. This strategy is efficient for light load conditions, increase in load diminishes the results.The level of smoke is considerably declined by utilizing late exhaust valve closing [14]. It also significantly affects the volumetric efficiency. This technique needs a lot boost of intake air, since fresh oxygen trapped inside cylinder is expelled back to intake manifold. It achieves a better trade-off between Smoke and NOx.

3. THEORETICAL STUDY OF PISTON BOWL

In case of direct injection diesel engine the piston bowl itself acts as a combustion chamber. The shape of bowl profile plays a significant role in combustion. The re-entrancy of bowl provides a platform for complete burning of fuel-air heterogeneous mixture. The terminology [9], associated with bowl shape is given below,

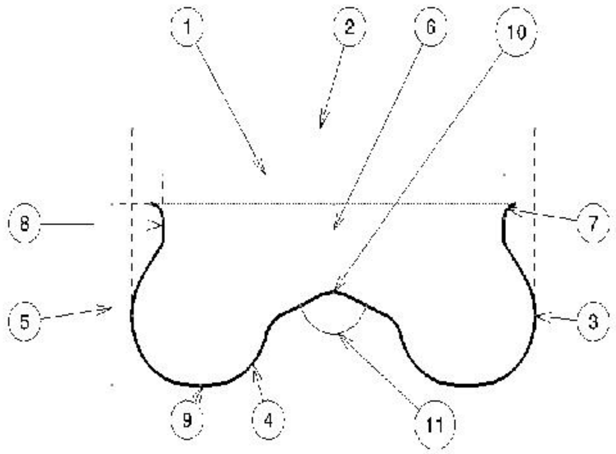


Fig 1: Piston Bowl Terminology

Table 2. Piston Bowl Terminology

1	inner bowl diameter (d)
2	outer bowl diameter (D)
3	outer bowl/Torroidal radius (R_O)
4	inner bowl radius (R_I)
5	bowl depth (b.d)
6	bowl cone depth (b.c.d)
7	lip radius (R_L)
8	lip length (S1)
9	bottom bowl length (S2)
10	pip length/radius (S3/R3)
11	cone angle

3.1. REFERENCE RANGE

The reference range for modification assists to restrict the change in parameter with respect to Smoke norms. The permissible range of parameters [4], [9], of piston bowl for 2-cylinder naturally aspirated genset engine are expressed as follow,

Table3. Reference parameter range for Piston Bowl

Parameters	Range (mm)
outer bowl dia [D]	48.00 - 51.00
inner bowl dia [d]	43.00 - 49.00
bowl radius	4.00 - 8.00
Cone Angle	130^0 - 140^0
bowl centre depth [b.c.d]	5.00 - 8.00
bowl depth (b.d)	12.00 - 18.00
lip radius (R_L)	0.5 - 1.5
lip length (S1)	1.00 - 5.00
bottom bowl length (S2)	2.00 - 9.00
inner bowl dia/ outer bowl dia [d/D]	0.85 - 0.90
outer bowl dia to bore dia [D/bore dia]	0.55 - 0.59
bumping height	1.05 - 1.26

4. MODIFIED PISTON BOWL GEOMETRY

The detailed specification of piston bowl to understand the necessary changes implemented in various piston bowls are expressed in terms of increase or decrease in percentage with respect to Option Baseline.

Table 4.Values of Parameter for Piston Options

Piston Option	Option A	Option B	Option C
bowl volume	9.17	2.50	-5.83
C.R	-5.83	-1.66	4.12
outer bowl dia (D)	-11.43	-12.86	-12.32
inner bowl dia (d)	-16.83	-15.45	-15.45
bowl radius (R_I)	40.00	56.00	34.00
bowl radius (R_O)	40.00	56.00	34.00
cone angle	-11.43	-14.59	-14.59
bowl centre depth (b.c.d)	25.00	6.67	6.67
bowl depth (b.d)	40.83	39.17	29.17
lip radius (R_L)	0.00	-100.00	-100.00
lip length (S1)	0.00	-27.20	-27.20
bottom bowl length (S2)	-25.33	-100.00	-100.00
pip length (S3)	0.00	-6.75	-6.75
d/D	-6.10	-2.97	-3.56
D/bore dia	-11.43	-12.86	-12.32

Note: All the Values are in terms of percentage

4.1. OPTION BASELINE

This is a baseline piston option , the smoke level is checked for this particular option. On the basis of literature reviewed for selection of suitable bowl geometry and test result obtained, various modified bowls are developed.

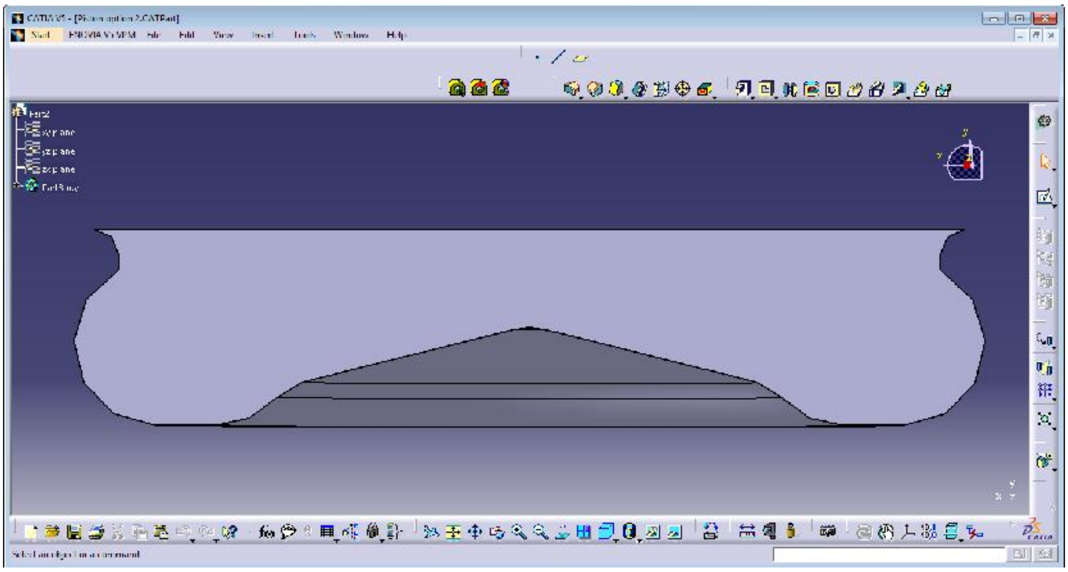
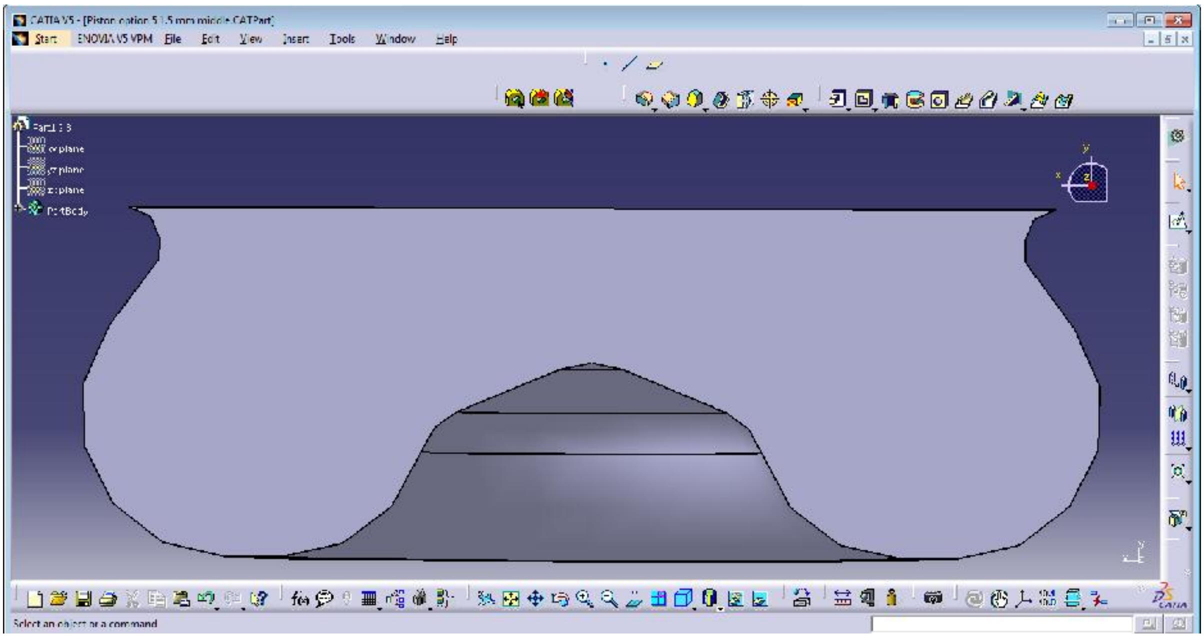


Fig 2: Option Baseline

4.2. OPTION A

The bowl volume is increased which has led to decrease in compression ratio. Lip radius lip length and pip length are kept constant to eliminate any change in emission on account of them.



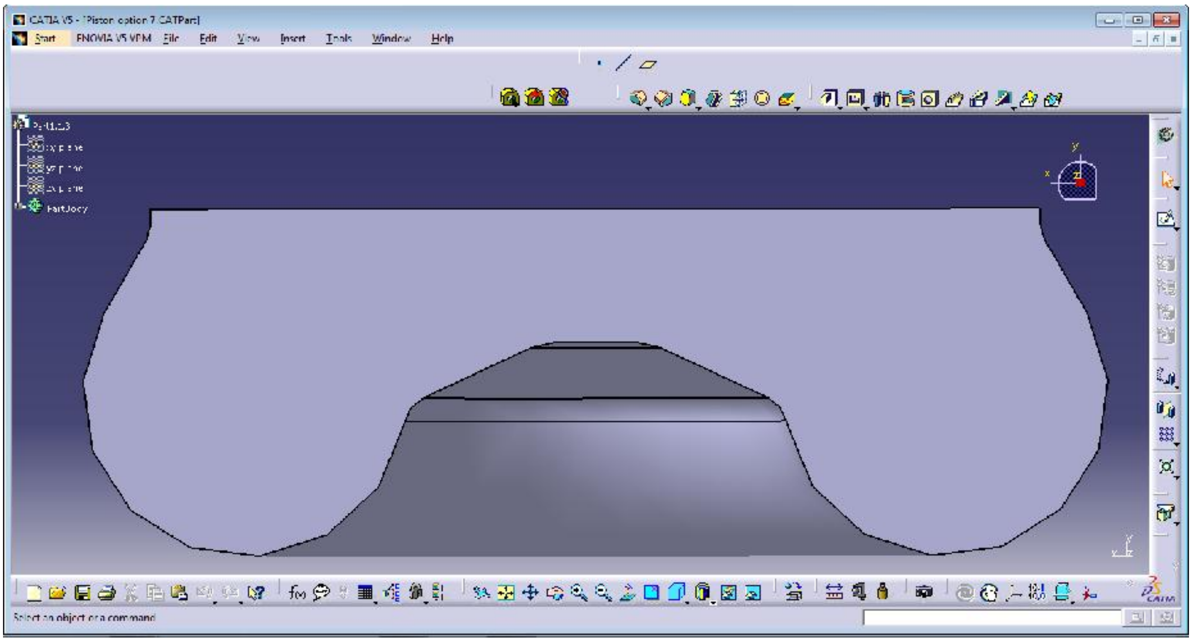
Select an object or a command

Fig 3: Option A

4.3. OPTION B

Comparative to Option Baseline, lip radius, lip length, pip length, bottom bowl length and cone angle are reduced to study the effect of these parameters on smoke limit which are missing in Option A.

The change in bowl volume and compression ratio is minute against changes observed in Option A. Among the entire options, bowl radius is kept maximum for this option.



Select an object or a command

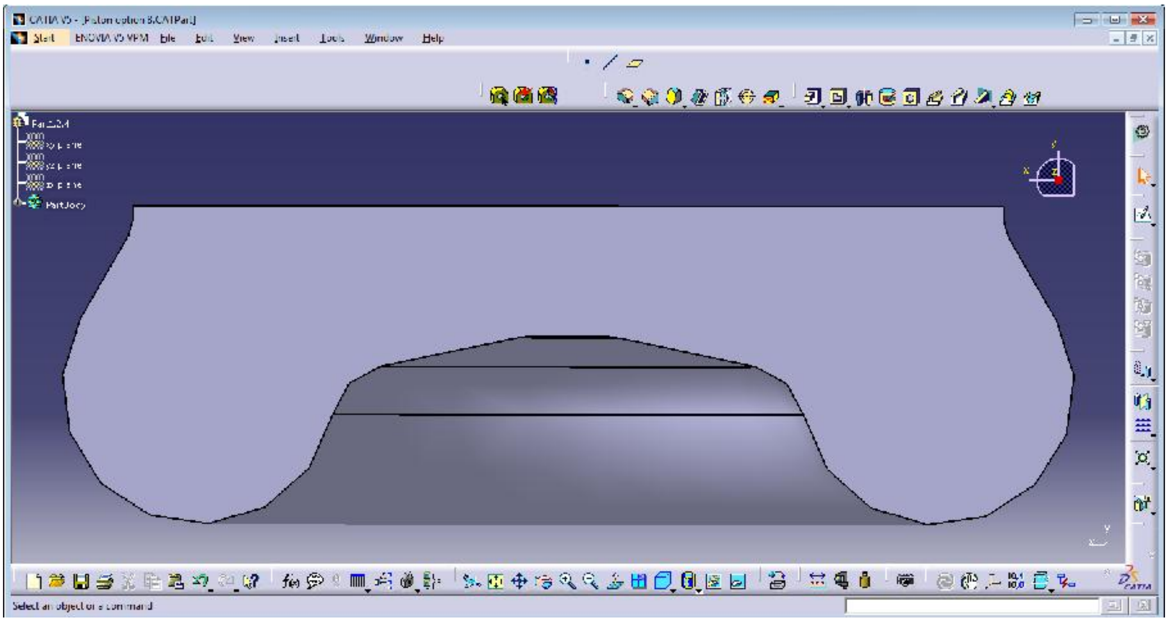
Fig4: Option B

4.4. OPTION C

Against the approach for Above Options, this Option C has decreased bowl volume and the highest compression ratio amongst all options.

Lip radius, lip length, pip length, bottom bowl length and cone angle are similar to Option B.

Other major changes involve increase in bowl radius, bowl depth and bowl centre depth as compared to Option Baseline but lesser than Option A and Option B



Select an object or a command

Fig 5: Option C

4.5. OVERLAP OF PISTON OPTIONS

The all Piston Option from Option A to Option D are overlapped to understand their particular change in specific parameter, either increased or decreased with respect to Option Baseline parameter. The Option Baseline, Option A, Option B and Option C are represented by Pink, Blue, Green and Red colours respectively.

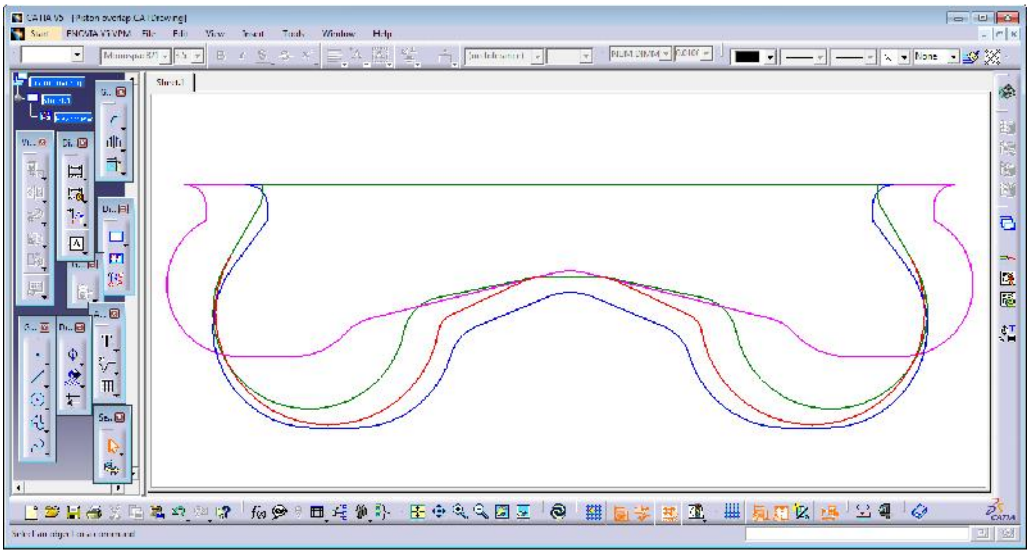


Fig 6: Overlap of Piston Options

5. TESTING INSTRUMENTS AND TEST CYCLE

5.1. TEST SETUP

The block diagram of test setup enables to picturized the arrangements of testing instruments involved during testing of smoke level.

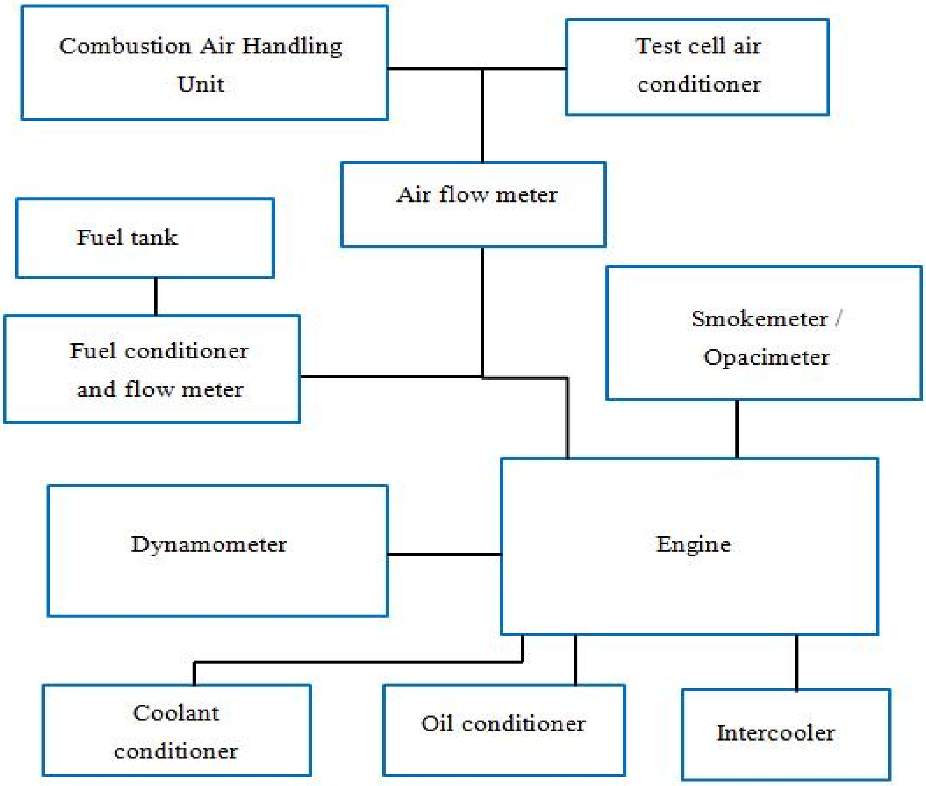


Fig 7: Block Diagram of Test Setup

Table 5.Technical specification of engine

Power ratings	10 kVA
	8 kW
Max power	10.5 kW
	13.99 HP
Rated speed	1500 rpm
No of cylinders	2
Cubic capacity	1.2 litres
Bore*stroke	87*100 mm
Aspiration	Naturally aspirated
Piston bowl	Re-entrant type
Cooling system	Liquid cooled
Application	Genset

5.2. CHARACTERISTICS OF INSTRUMENTS

5.2.1. SMOKEMETER

The Smokemeter used is of AVL and its model is 415SG002. The characteristics of AVL Smokemeter helps to determine the minimum error involved while undergoing test in form of repeatability and also certain other parameters as mentioned below,

Table 6. Characteristics of Smokemeter

Parameters	Values
Measurement Range	0 - 10 FSN
Detection Limit	0.002 FSN
Resolution	0.001 FSN
Repeatability	0.005 FSN + 3% of measured value

5.2.2. COMBUSTION AIR HANDLING UNIT (CAHU)

The SIERRA based instrument is used for CAHU system. the accuracy of instrument for temperature, pressure and relative humidity is mentioned below,

Table 7. Characteristics of CAHU

Parameters	Values
Steady state accuracy (temperature)	±1.0 0C
Steady state accuracy (relative humidity)	±5.0 %
Steady state accuracy (pressure)	±1.0 mBar
Transient accuracy (pressure)	±1.0 mBar

5.2.3. FUEL CONDITIONING UNIT

The fuel conditioning unit of HORRIBA with model FQ2200CR is used for conditioning the temperature and fuel flow rate of fuel system. The temperature control stability of this instrument is 0.05^0 C. This ensures that during the trial the minimum deflection of inlet fuel temperature is by 0.05^0 C which ensures minimum error during trial.

5.3. TEST CYCLE

The testing cycle is pre-dominantly based on ISO 8178. The 5-mode cycle is illustrated for 2-cylinder, 10.5 kW at 1500 rpm as mentioned below,

Table8. Test cycle for 10.5 kW engine

Mode	Engine Speed	Load (Nm)	Weighting Factor
1	1500	67	0.05
2	1500	50.25	0.25
3	1500	33.5	0.3
4	1500	16.75	0.3
5	1500	6.7	0.1

6. EXPERIMENTAL RESULTS

The experimental testing is conducted for static injection timing of 15 degree BTDC (before top dead centre), H.P.P (high pressure piping) of 6x1.8x550 mm and tappet clearance of 0.2 mm for intake valve and 0.6 mm for exhaust valve. The injector configuration is defined by number of holes x cone angle x hole through flow (for 30 seconds). Testing trials are conducted for Piston Options by varying injector configuration and washer thickness.

6.1. OPTION BASELINE

The Option Baseline is tested for nozzle configuration of 8x146x525 and 6x146x350 and for washer thickness of 3mm, 3.5 mm and 4 mm.

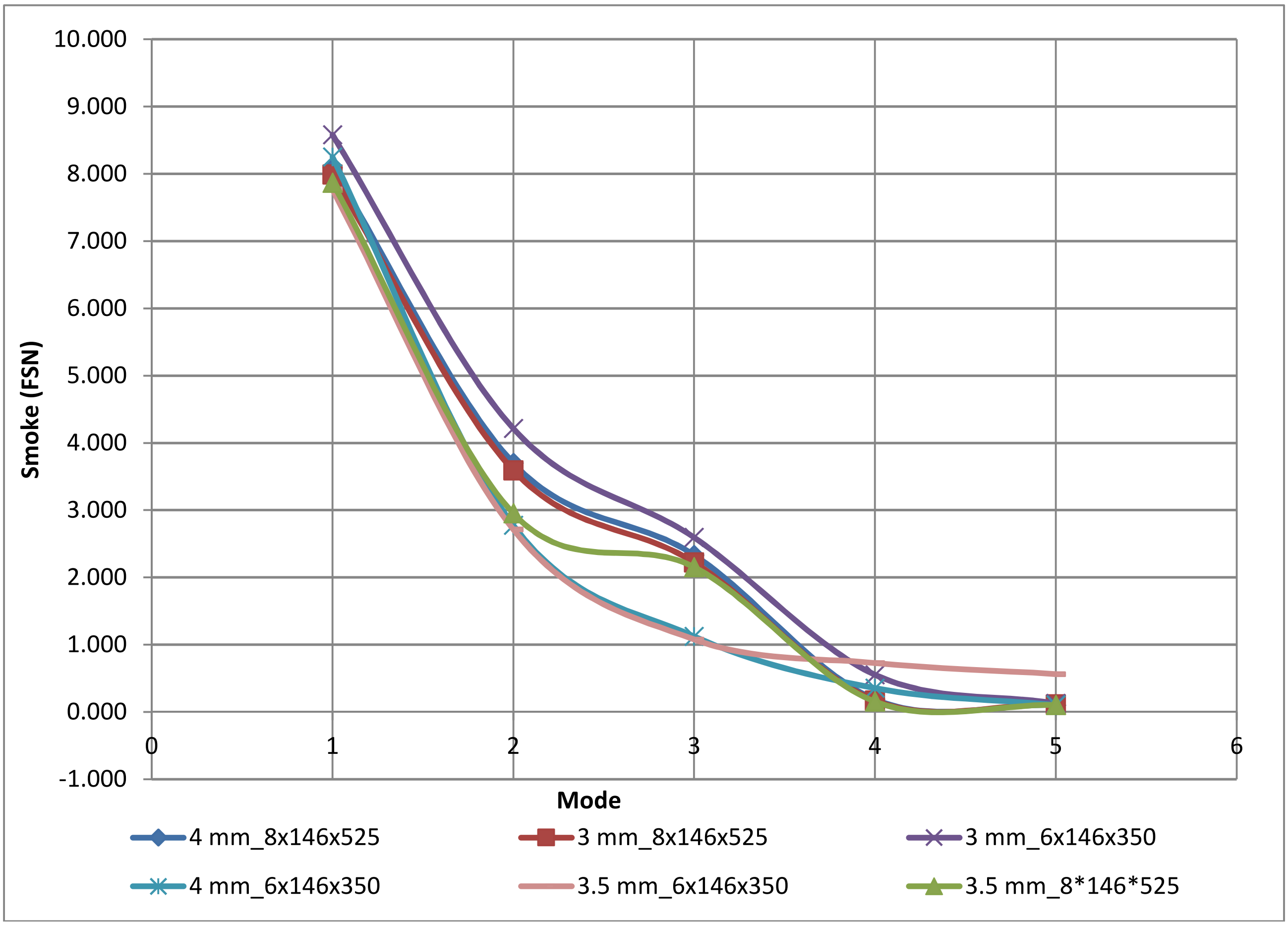


Fig 8: Mode Vs Smoke for Option Baseline

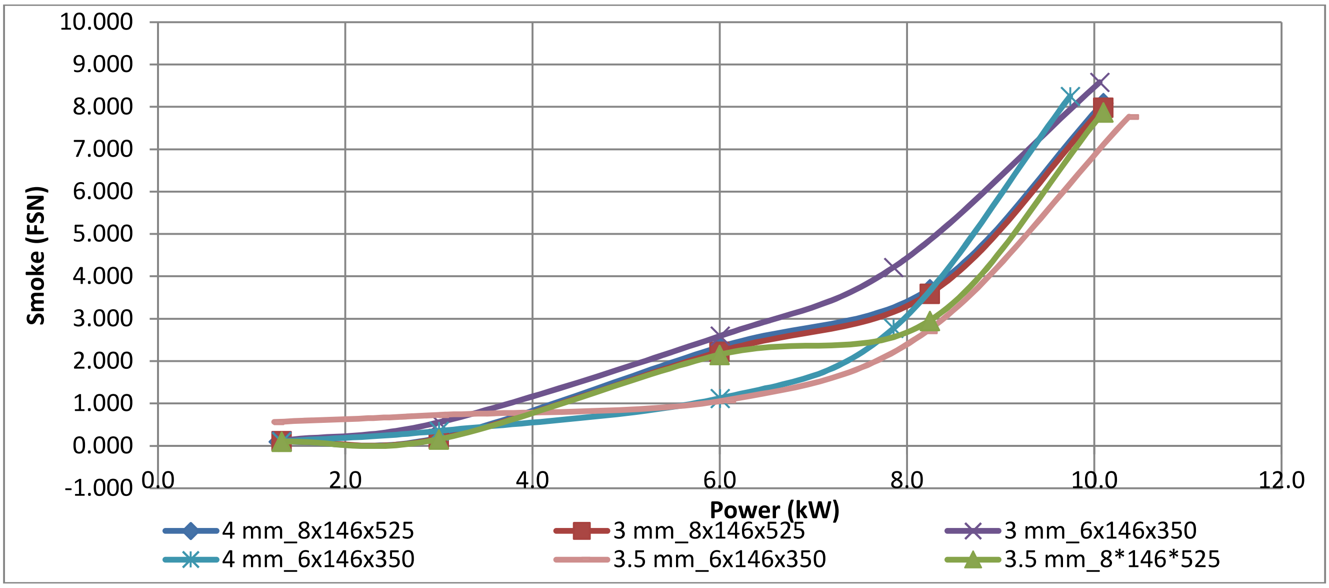


Fig 9: Power Vs Smoke for Option Baseline

6.1.1. INTERPRETATION OF GRAPHS

The level of smoke for 1st and 2nd mode is above 2.2 FSN while 3rd, 4th and 5th mode are below 2.2 FSN. The minimum smoke level of 7.769 FSN is obtained at 10.4 kW power corresponding to 66 Nm of torque. The best combination of Static injection timing, washer thickness and nozzle configuration for Option Baseline is 15 degree BTDC, 3.5 mm and 6x146x350 respectively. The worst possible smoke limit of 8.581 FSN is obtained for washer thickness of 3 mm that has deteriorated the maximum power to 10.1 kW.

6.2. OPTION A

The trial for Option A is for nozzle configuration of 6x146x350and for washer thickness of 1mm, 2mm, 3mm, 3.5 mm and 4 mm.

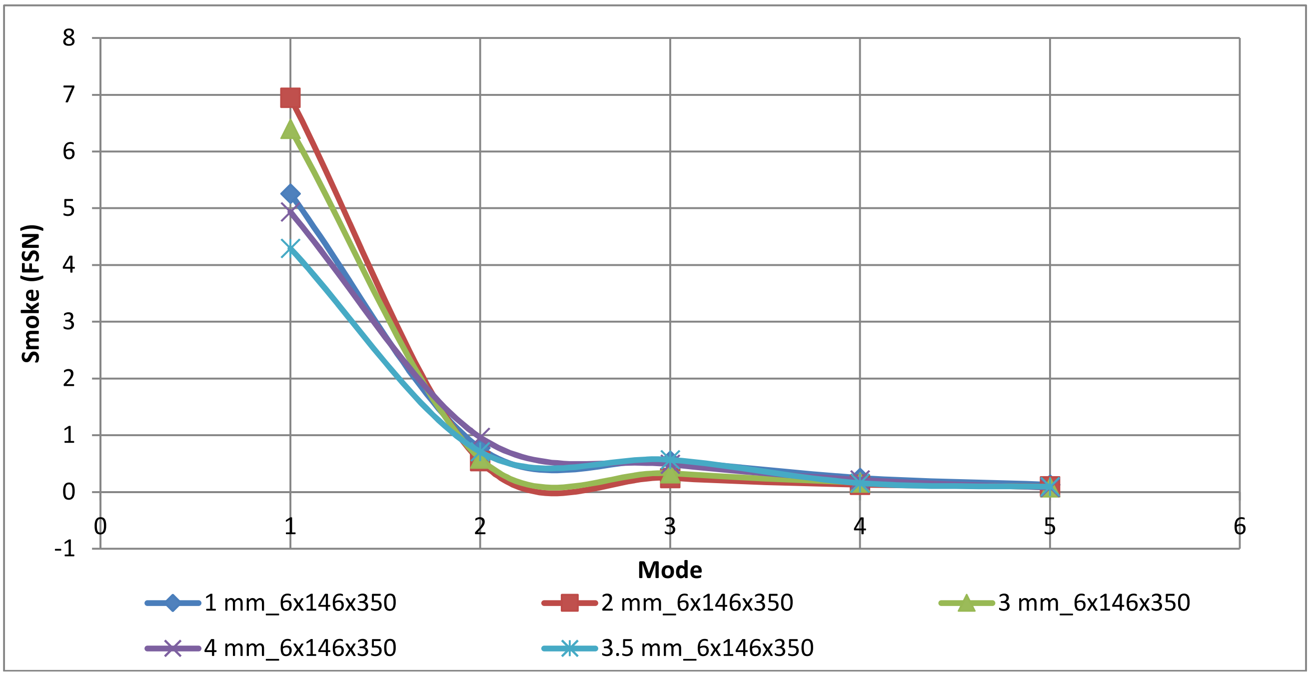


Fig 10: Mode Vs Smoke for Option A

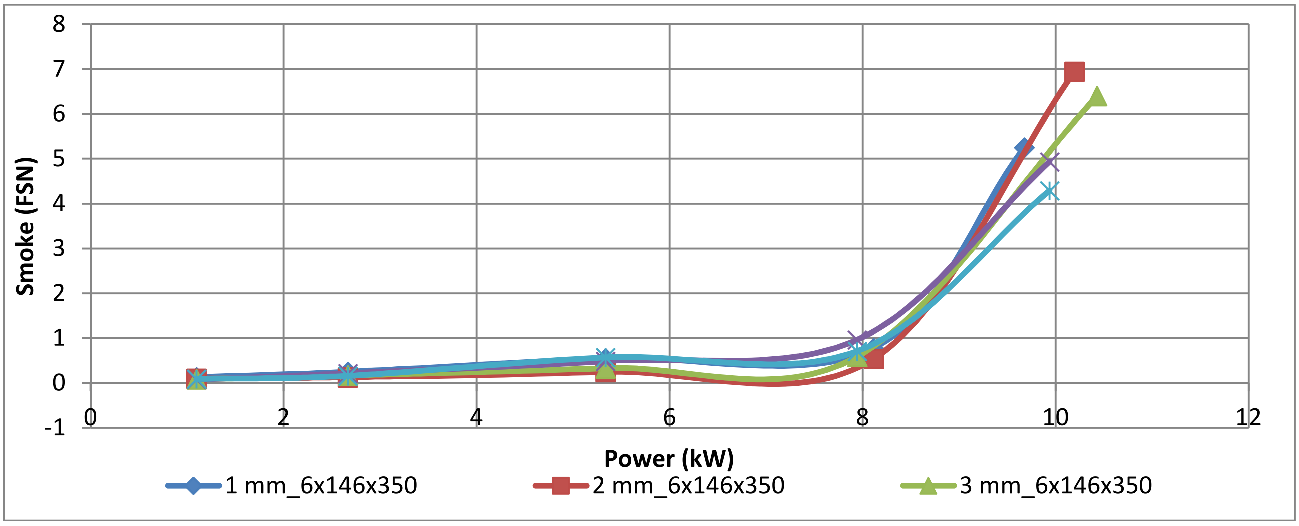


Fig 11: Power Vs Smoke for Option A

6.2.1. INTERPRETATION OF GRAPH

For Option A expect at mode 1, the smoke level at all mode is below 2.2 FSN. The best smoke level of 4.291 FSN is achieved for washer thickness of 3.5 mm, nozzle configuration of 6x146x350 and static injection timing of 15 degree BTDC. The maximum power achieved is 10.5 kW. The nozzle angle swing results indicate that 146 degree angle is best followed by 143 degree and 150 degree respectively for washer thickness of 2 mm, 2.5 mm, 3 mm and 3.5 mm.From this trial it is concluded that, henceforth washer thickness from 2 – 3.5 mm and nozzle cone angle of 146 degree is finalized.

6.2.2. TRIALS FOR STATIC INJECTION TIMING

The trials for injection timing swing of Option A comprises testing the option for three particular injection timings of 11 degree BTDC, 13 degree BTDC and 17 degree ATDC of which the earlier two have retarded timing and latter one has advanced timing,for nozzle configuration of 6x146x350 and washer thickness of 3.5 mm. This trials enables to observe the influence of change in static injection timing on smoke level.

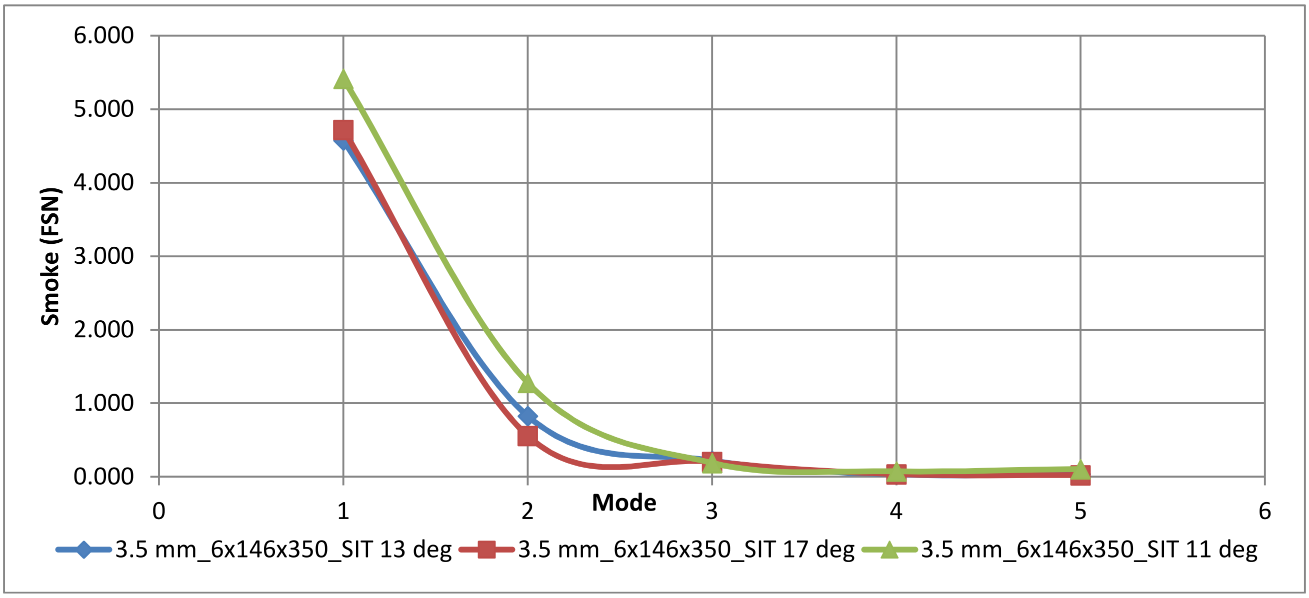


Fig 12: Mode Vs Smoke for Timing swing of Option A

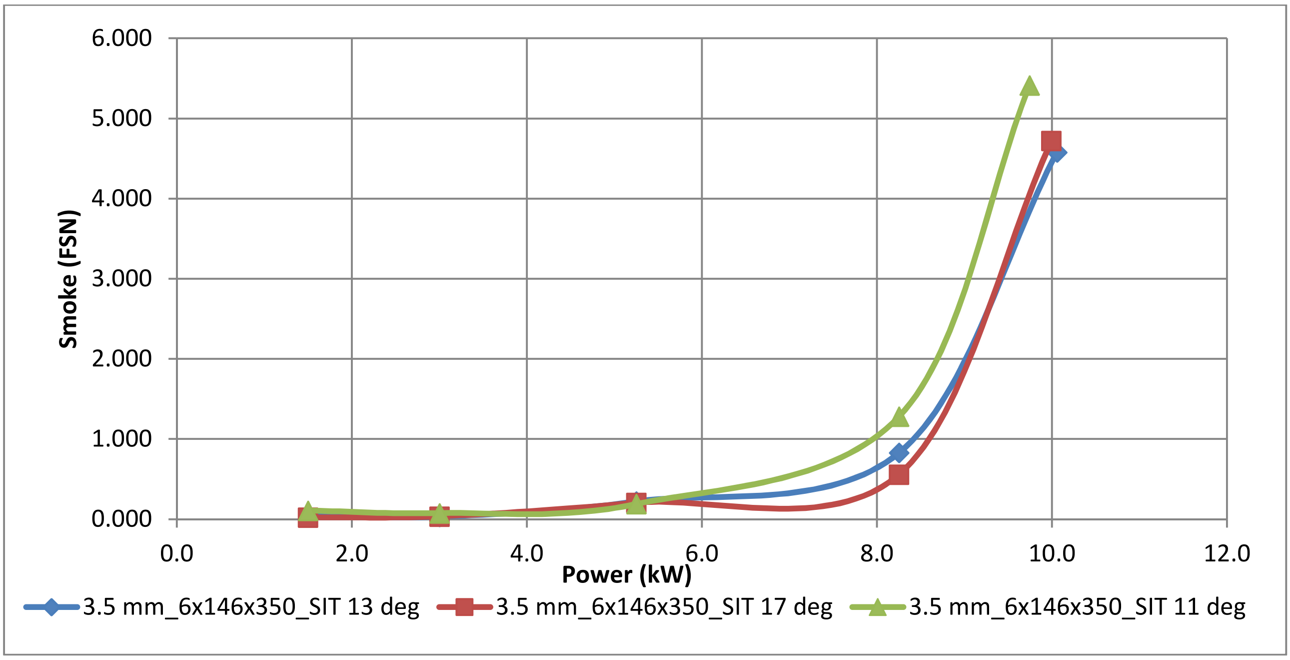


Fig 13: Power Vs Smoke for Timing swing of Option A

6.2.3. INTERPRETATION OF GRAPH

For an identical combination of washer thickness and nozzle configuration, the smoke level for 11 degree BTDC, 13 degree BTDC and 17 degree BTDC injection timing at 1st mode is 5.416 FSN, 4.578 and 4.722 against 4.291 FSN for 15 degree BTDC which signifies that advancing or retarding the timing has deteriorated the smoke level marginally. It is clear that best smoke level is obtained for 15 degree BTDC timing but the margin of deterioration for two degree advance and retard is 6.68% and 10.04% respectively.

6.3. OPTION B

The Option B is tested for nozzle configuration of 6x146x300 and 6x146x350 and for washer thickness of 2mm, 3mm, 3.5 mm and 4 mm.

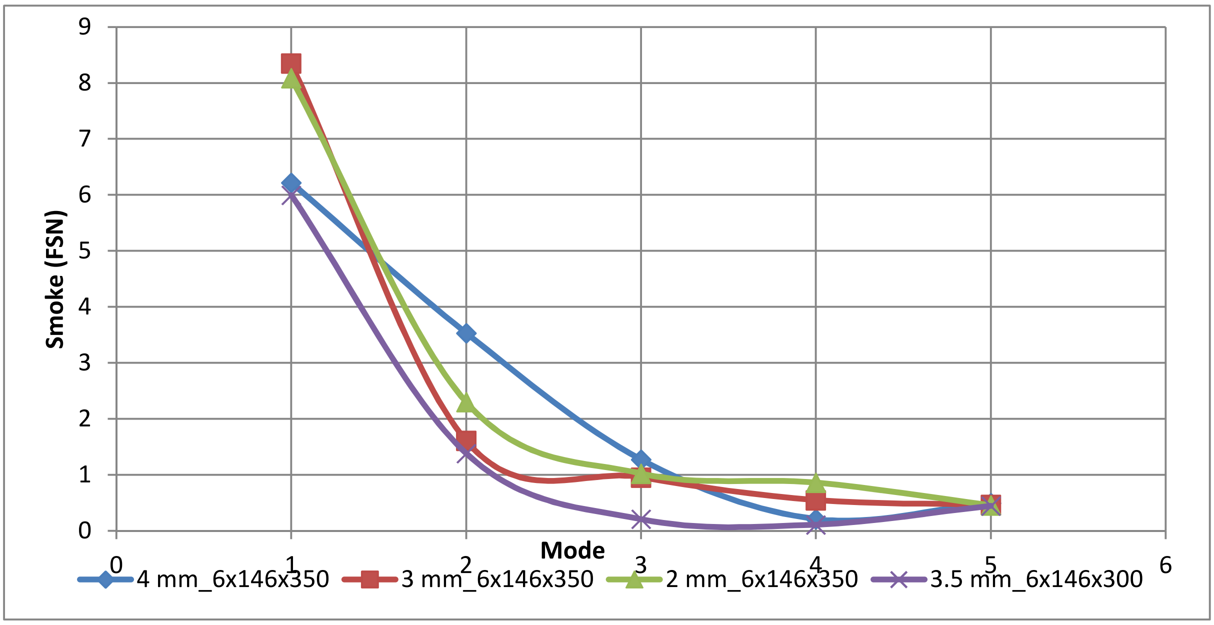


Fig 14. Mode Vs Smoke for Option B

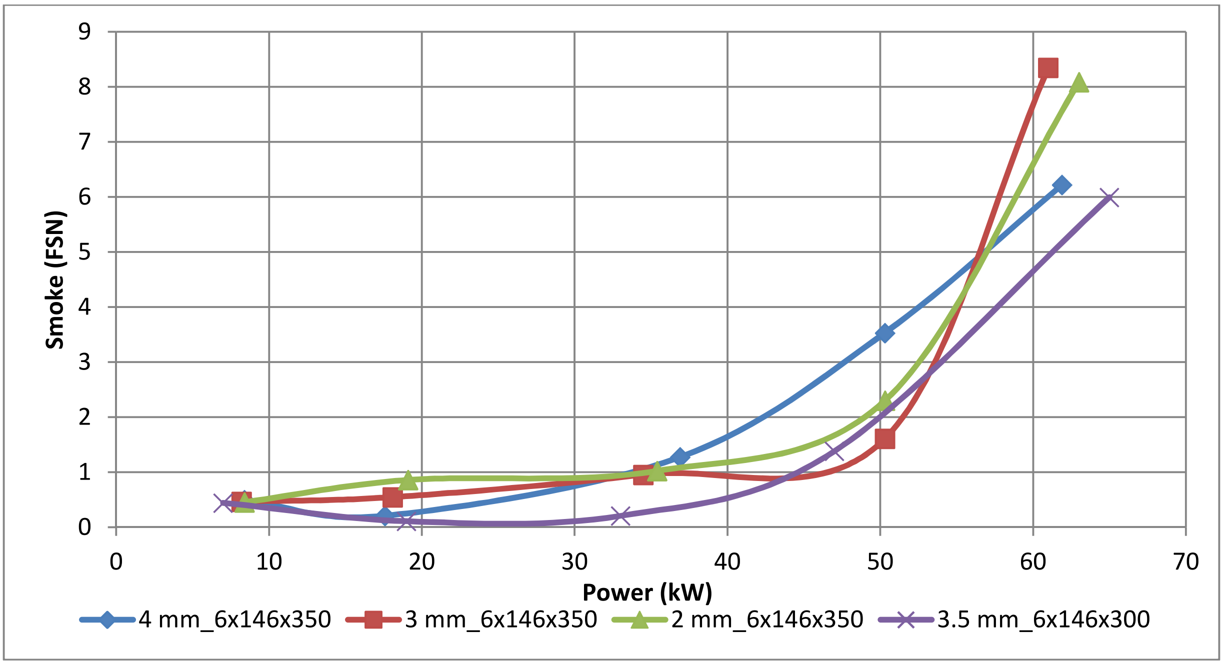


Fig 15: Power Vs Smoke for Option B

6.3.1. INTERPRETATION OF GRAPH

The smoke results obtained are very diverse for washer thickness. The smoke level for 2 mm and 4 mm washer thickness at 1st and 2nd mode are above 2.2 FSN whereas for 3 mm and 3.5 mm smoke level is above 2.2 FSN for 1st mode only. Least smoke level of 5.997 FSN is obtained for 10.2 kW power for 15 degree BTDC static injection timing and washer thickness of 3.5 mm. The margin of smoke level above permissible limit is higher than Option A and lowers than Option Baseline.

6.4. OPTION C

The Option Baseline is tested for nozzle configuration of 8x150x525, 8x146x525 and 6x146x350 and for washer thickness of 2mm, 3mm and 4 mm.

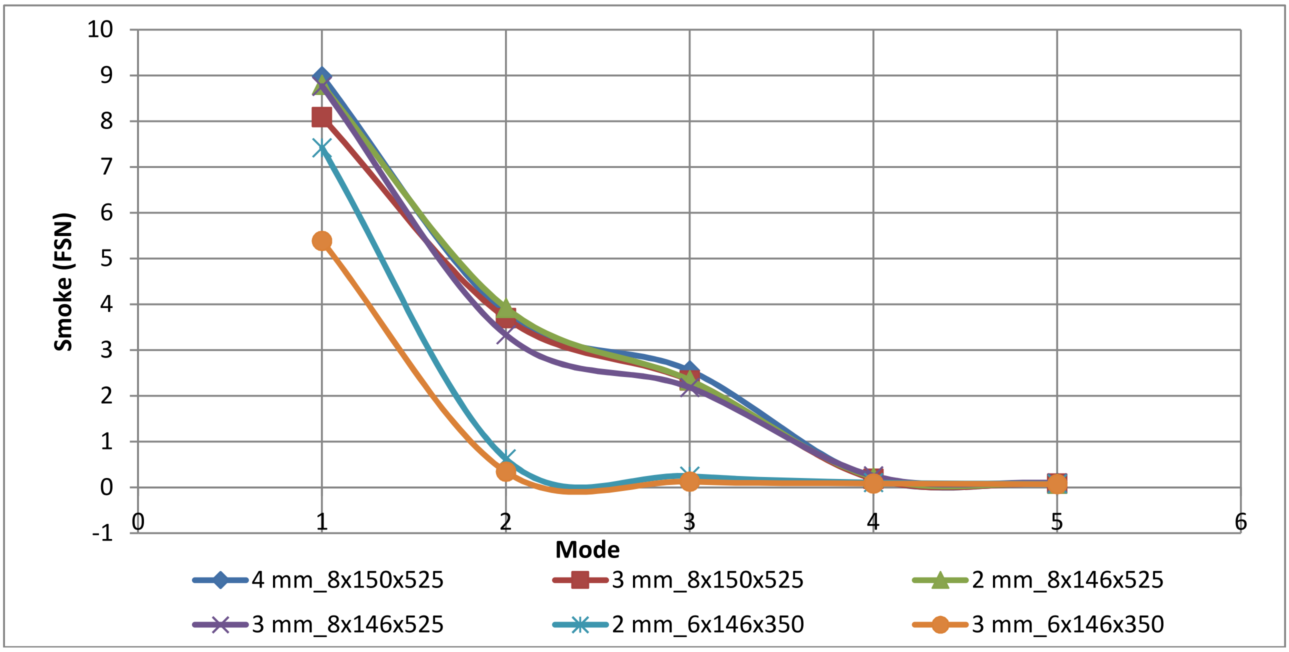


Fig16: Mode Vs Smoke for Option C

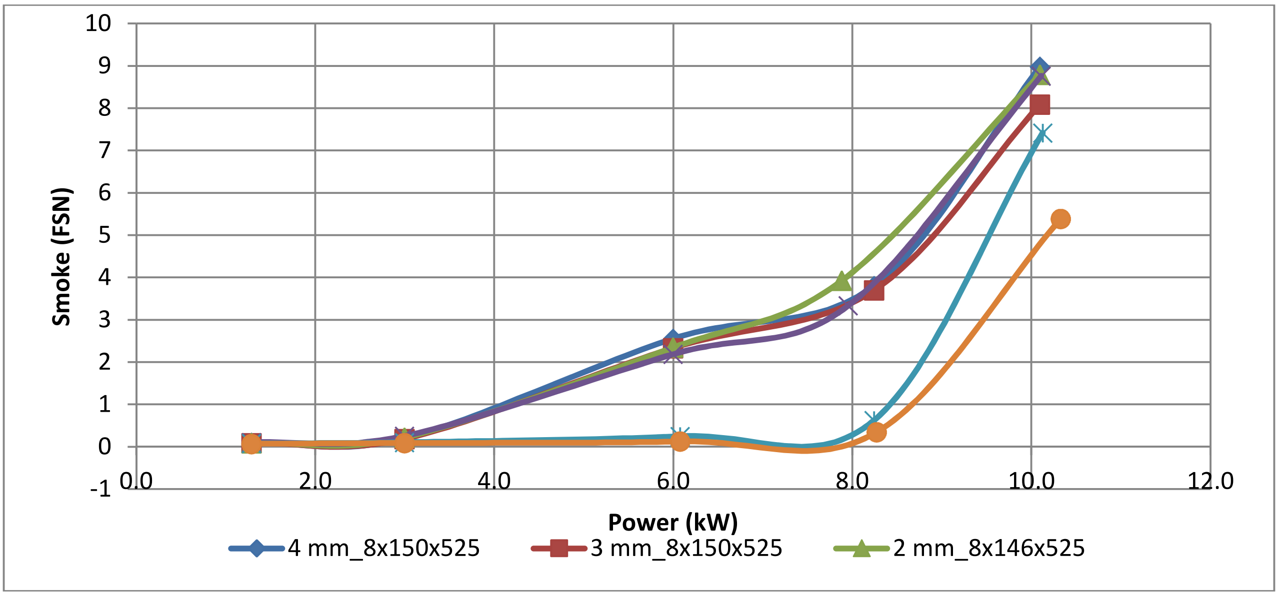


Fig 17: Power Vs Smoke for Option C

6.4.1. INTERPRETATION OF GRAPH

The best smoke of 5.387 FSN is achieved for 3 mm washer thickness and 6*146*350 nozzle configuration. The maximum power achieved is 10.3kW greater than Option B but lesser than Option A and Option Baseline.

For nozzle angle of 146 degree and , 2 mm and 3 mm washer has smoke level above 2.2 FSN at mode 1 only, whereas for 150 degree nozzle angle the smoke level is above 2.2 FSN at mode 1st, 2nd and 3rd mode respectively.

From this trial again it is evident that washer thickness within range of 2 mm to 3 mm for 146 degree cone angle is best suitable.

7. CONCLUSION

1) The best results are obtained for Piston Option A, whereas comparatively for same nozzle configuration, Piston Option B and Option C showed an increase of 39.75 % and 25.54 % respectively.
2) The inner diameter to bore diameter ratio of 0.5 against Option Baseline ratio of 0.58 is effective as the smoke is reduced considerably.
3) The cone angle and bowl depth of Option A is greater, so the volume was highest and compression ratio is least among all Options.
4) The pip length has adversely affects the smoke level in case of Option B and Option C. Owing to lesseffective in uniform bulk mixing in the region of bowl centre depthdue to reduction in pip length
5) The lip radius positively influences the smoke limit, contradictory to it Option B and Option C lacks this parameter and their limit of smoke has risen up.
6) The injector cone angle of 146 degree with 3 mm for Option C and 3.5 mm for Option B and Option C respectively, followed by cone angle of 143 degree, provided that washer thickness is 2 mm, 3 mm and 3.5 mm.
7) The injector cone angle of 150 degree is least effective in reducing the smoke level for all Options.
8) Static injection timing advance and retard by 2 degree for Option 5 has risen up the smoke level by 6.68 % and 10.04 % respectively against Static injection timing of 15 degree before top dead centre.
9) The power achieved by Option A for 3.5 mm washer with injector configuration of 6x146x350 is 10.5 kW, for rest all trials power achieved is less than 10.5 kW.

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REFERENCES

[1] Shafie, Mohamad N A. and Said, Muhamad M F. “Cold Flow analysis on internal combustion engine with different piston bowl configurations”, Journal of Engineering Science and Technology, 2017, Vol. 12, No. 4,ISSN: 1823-4690, pp. 1048-1066.

[2] Hielscher, Klaus. Brauer, Maximillan. Baar, R. “Reduction of soot emissions in diesel engines due to increased air utilization by new spray hole configurations”, Automotive and Engine Technology, Springer Journals, 2016, Vol. 1, Issue. 1-4, pp. 69-79.

[3] Gafoor, Abdul C P and Gupta, Rajesh. “Numerical investigation of piston bowl geometry and swirl ratio on emission from diesel engines”,Energy Conversion and Management, ELSEVIER, 2015, Vol. 101, pp. 541-551.

[4] Dimitriou, Pavlos. Wang, Weiji.Peng.Zhijun. “A Piston Geometry and Nozzle Spray Angle Investigation in a DI Diesel Engine by Quantifying the Air-Fuel Mixture”, International journal of spray and combustion dynamics, 2005, Vol. 7, No. 1, pp. 1-24.

[5] Quazi, Muzaffar A. Singh, Shakti K. Jadhao, Mangeshkumar. “Effect of piston bowl shape, swirl ratio and Spray Angle on Combustion and Emission in Off Road Diesel Engine”, SAE Technical Paper, 2015, 2015-26-0142: DOI 10.4271/2015-26-1042.

[6] Taghavifar,Hadi. Khalilarya,Shahram. Jafarmadar,Samed. “Engine structure modifications effect on the flow behaviour, combustion, and performance characteristics of DI diesel engine”, Energy Conversion and Management, ELSEVIER,2014, Vol. 85, pp. 20-32.

[7] Bhat P. Nalawade S. Achari S. Subramanian, Ashwin K. Pawar N. Gayen H. Marathe H. "Strategies to Meet CPCB Stage - II Emission Norms for Naturally Aspirated Diesel Genset Engines", SAE Technical Paper, 2013, 2013-26-0134: URL. https://doi.org/10.4271/2013-26-0134.

[8] The Gazette of India, Extraordinary, Part II, Section 3, Sub-section (i), Ministry of Environment and Forests (2013).

[9] Mobasheri, Rauf and Peng, Zhijun. "Analysis of the Effect of Re-Entrant Combustion Chamber Geometry on Combustion Process and Emission Formation in a HSDI Diesel Engine", SAE Technical Paper,2012, 2012-01-0144: URL. https://doi.org/10.4271/2012-01-0144.

[10] Sheikh, S., Gokhale, N., Thatte, V., Gandhi, N. et al., "Efficient Approach for Optimization of Piston Bowl Shape, Compression Ratio and EGR for DI Diesel Engine", SAE Technical Paper, 2011-24-0013, 2011, https://doi.org/10.4271/2011-24-0013.

[11] Zhu Y. Zhao H. Melas D. Ladommatos N. "Computational Study of the Effects of the Re-entrant Lip Shape and Toroidal Radii of Piston Bowl on a HSDI Diesel Engine's Performance and Emissions", SAE Technical Paper, 2004, 2004-01-0118:URL. https://doi.org/10.4271/2004-01-0118.

[12] Liu,Renshan and Zhang. Chao. “A Numerical study of NOx reduction for a DI Diesel injection with complex geometry”, Journal of Energy Resources Technology, American Society of Mechanical Engineers, 2004, Vol. 126, Issue. 1, pp. 13-19.

[13] Zammit J. Mcghee M. Shayler P. Law T. Pegg I. “The effects of early inlet valve closing and cylinder disablement on fuel economy and emissions of a direct injection diesel engine.” Energy, ELSEVIER, 2015, Vol.79, pp. 100-110.

[14] Kim,Jaeheun. Park, Stephen S. Bae,Choongsik. “The effects of late intake valve closing and different cam profiles on the in-cylinder flow field and the combustion characteristics of a compression ignition engine”, Journal of Automobile Engineering, Institute of Mechanical Engineers, 2017, pp. 1-11.

[15] Stone, Richard. Introduction to Internal Combustion Engines, SAE International, third edition, 1999, pp. 216-265.