EFFECT OF FUEL INJECTION SYSTEM ON SMOKE EMISSION FOR TWO CYLINDER GENSET ENGINE

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***Abstract*— The two cylinder genset engine with mechanical injection system is gaining popularity in Indian market due to its simplicity in design, robust structure and eco-friendly aspect. The injection system and its ancillaries are pivotal for controlling exhaust smoke emissions. To achieve smoke limit as per CPCB II norms, it is first essential to understand the effect of injection system parameters on exhaust smoke and then utilizing this perception of parameters to achieve control on smoke level. Piston bowl geometry is experimentally tested for different combination of washer, nozzle configuration, injection timing and high pressure piping. The trials indicate wide fluctuations in smoke level for change in injection timing and high pressure piping for different washer thickness. Minimum smoke level of 0.158 m^{-1} is achieved for pressure piping, washer and nozzle configuration of 1.5x550 mm, 2 mm and 5x143x285 respectively whereas for remaining combinations smoke level at first mode of test cycle is high.**

Keywords— High pressure piping; Injection timing; Nozzle configuration; Washer.

I. INTRODUCTION

The trend followed in India is to manufacture genset in simplest possible way with environmental friendly technology. To ensure this environmental friendly nature of technology, government of India has introduced CPCB II (Central Pollution Control Board) norms with effect from 1st January 2014. It is mandatory that every genset manufactured within territorial boundary or imported from outside has to follow these norms strictly. According to these norms the genset is categorized in three power category as per power rating; up to 19 kW, 19 kW to 75 kW and 75 kW to 800 kW.

Owing to the concept of simplicity in development and environmental friendly, to achieve it in terms of emission norms, mechanical in-line fuel system plays a significant role as well as it is the crux of combustion phenomenon. The major parameters of injection system involves injection pressure, nozzle hole diameter, number of holes, spray cone angle, amount of fuel injected per stroke, nozzle tip protrusion. There is limit to increase mechanical in-line injection pressure, pressure beyond 750 bars cannot be achieved and as per Indian market minimum limit for nozzle diameter is 0.14 mm so, the combination of remaining parameters is to be ensured for ample atomization, air utilization and bulk mixing with air.

TABLE I. 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

II. LITERATURE REVIEW

The injection timing has a significant role in emission [1] so, a relation is established between end of injection timing, speed and CO emission. At low speed, as the end of injection with respect to BTDC advances the CO emission rises on the contrary at the medium speed of engine CO emission gradually lowers as the injection closing is advanced with respect to BTDC. At higher speed for retarded end of injection with respect to BTDC, CO emission rises. The increase in combustion pressure advances the combustion phenomenon and increase in advanced timing raised the in-cylinder pressure [2]. Smoke and HC emissions are reduced on increasing the injection pressure but at the compensation of rise in NOx emissions. At low load condition CO emissions hardly depends on injection pressure. On contrary at high load CO emissions reduced at high injection pressure. Compared to 160 Mpa injection pressure smoke emissions depend upon 80 Mpa pressure. Injection timing and injection [3], plays a pivotal role in determining performance and emissions. Early injection timing allows smoke emission and higher NOx emission than late injection timing. Increasing fuel injection pressure leads to better mixing of air-fuel mixture causing to higher heat release and peak pressure, increase in pressure causes decrease in droplet diameter hence atomization and evaporation at quicker rate. Increasing the spraying pressure produces fast mixture formation; reduce in delay time and release of maximum energy. By advancing the fuel injection timing [4], the peak pressure in cylinder is closer to the BTDC causing a rise in effective pressure and torque inside the engine. Injection timing plays a pivotal role in influencing brake specific fuel consumption (BSFC). Retarding the timing leads to rise in BSFC. The advancing strategy is useful in case of reducing the exhaust gas temperature. Retarding timing even though influence BSFC but it leads to a considerable rise in exhaust temperature. The smoke emission from exhaust is decreased as the injection timing is advanced [5]. The advancing limit is up to 6 degree BTDC beyond it smoke level further increased for four cylinder indirect injection engine while NOx emission increased even though smoke level raised beyond 6 degree BTDC. This situation raised due to abrupt mixing of the evaporated air and fuel. This timing is considered as optimal for emission purpose. The BSFC is optimally increased by increasing the retardation timing of injection [6] whereas the CO emission gradually rises from no load to full load condition. Retardation in injection timing causes the unburned hydrocarbons to rise heavily. The NOx emission increases successively with the rise in loading conditions. For a multi fuel injection system [7], the length of full injection pipe has a huge effect on fuel supply at injector end. Altering the length causes a change in pressure wave whose effects are high on pump side comparatively to the injector side. This leads to fluctuation in pressure through a wide range of 12 Mpa to 125 Mpa. Short length pipes are preferable for accuracy of fuel supply during injection as such they cause reduced pressure waves and amplitude at the forefront of injector. To meet CPCB II emission norms [8], it is essential to select suitable injector nozzle configuration in form of number of holes, cone angle and hole through flow along with optimized nozzle tip protrusion. Altering the pressure of pipe leads to change in time require for pressure waves of fuel flow to travel from pump end to injector end. Change in diameter of pipe leads to variation in obstruction of fuel flow causing a variation in amount of fuel flow through nozzle. Change in injection timing influences the chemical delay during combustion. Re-entrant type of piston bowl [9], is most effective in reducing the smoke emission from engine. The soot formation majorly occurs during the initial stage of combustion and after this phase it is oxidized during the expansion stroke particularly. The advanced injection timing leads to accumulation of evaporated fuel and creates longer ignition delay. This delay is responsible for high pressure and temperature inside cylinder and efficient burning of fuel and hence reduction in smoke emission. An approach is proposed which includes studying the effect of injection timing and nozzle tip protrusion on re-entrant piston bowl [10]. A trade-off between NOx and soot emission is established and indicates that advancing the injection timing for a constant washer and re-entrant piston bowl declines the smoke intensity. The soot formation and oxidation theory for re-entrant piston bowl diesel engine [11], states that the major reason behind soot formation is fuel rich and oxygen deficient zone and high temperature inside cylinder. The smoke formation is dominant during the initial stage of combustion later this depletes due to oxidization of smoke.

The smoke emissions are controllable [12], if number of holes are increased and diameter of holes are decreased. Numbers of holes are increased further then oxygen deficient zone are formed adjacent to tip area of each hole leading to rise in smoke emissions.

It is clear from literature that injection timing, nozzle configuration, high pressure and washer thickness is necessary to determine the level of smoke for a genset engine. Further it is necessary to comprehend the effect of fuel injection and piston bowl geometry on smoke emissions separately so, during trials it is essential to keep either of the parameters constant and then successively conduct further trials.

III. Piston Bowl Geometry

The piston bowl in case of diesel engine acts as a combustion chamber for atomization, better mixing, evaporation and combustion of air-fuel mixture. The re-entrant profile of bowl is pivotal as fuel sprayed by injector circulates in profile of piston bowl and it positively influences the smoke level [9], [10]. In this paper the parameters of piston bowl geometry are kept constant particularly to observe only the influence of fuel injection system on exhaust smoke so, diagram and parameters of piston bowl kept constant are mentioned below. The parameters of piston bowl under experimental condition are expressed in terms of range as below,

TABLE II. RANGE OF PISTON BOWL PARAMETER

Bowl parameters	**Range in 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

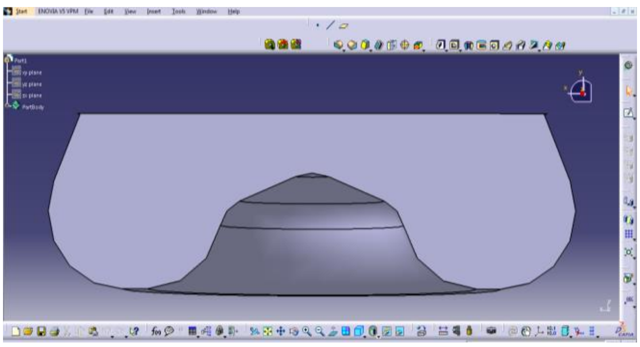


Fig. 1. Re-entrant profile of Piston Bowl

IV. Testing Instruments and Test Cycle

A. *Technical specification of engine*

The technical specifications of genset engine are mentioned below in tabular format.

TABLE III. 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 liters
Bore*stroke	87*100 mm
Aspiration	Naturally aspirated
Piston bowl	Re-entrant type
Cooling system	Liquid cooled
Application	Genset

B. *Test setup*

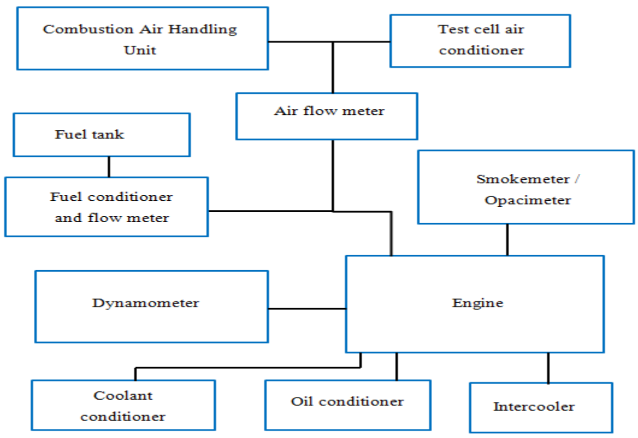


Fig. 2. Block Diagram of Test Setup

The diagram representation in form of blocks enables to visualize the arrangements of testing instruments involved during performing trials for smoke level.

C. *Characteristics of Iinstruments*

1) *Opacimeter*

The AVL Smokemeter is of model 4390G004. The characteristics of AVL Opacimeter helps to find out minimum detection limit, resolution involved while performing trials as mentioned below,

TABLE IV. CHARACTERISTICS OF OPACIMETER

Parameters	Values
Measurement range	0 to 10 m^{-1}
Detection limit	0.0025 m^{-1}
Resolution	0.001 m^{-1}
Zero stability	0.25 m^{-1}/30 min

2) *Combustion Air Handling Uunit (CAHU)*

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

TABLE V. CHARACTERISTICS OF CAHU

Parameters	Values
Steady state accuracy of temperature	±1.0 ^{0}C
Steady state accuracy of relative humidity	±5.0 %
Steady state accuracy of pressure	±1.0 mBar
Transient accuracy of pressure	±1.0 mBar

3) *Fuel conditioning unit*

The fuel conditioning unit of HORRIBA unit having model number FQ2200CR is necessary for conditioning fuel flow rate of fuel system as well as temperature. The stability of temperature control is 0.05^0 C. Minimum errors are induced during trial since the deflection in temperature of fuel inlet is meagre by 0.5^0 C.

4) *Test cycle*

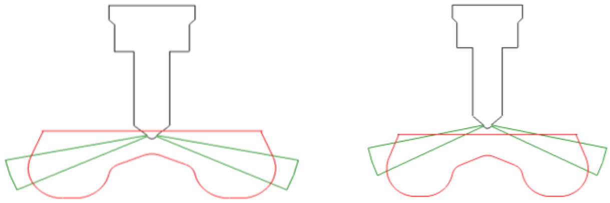
The standard used for testing of cycle is ISO 8178. The test interval for each mode is five minutes. The five modes represent the loading condition for engine at constant speed of 1500 rpm. The loading conditions are applied in ascending order 10%, 25%, 50%, 75% and 100% of maximum load that engine sustains, for this genset engine it is 67 Nm. A weighting factor is assigned to each mode in terms of percentage. Before initiating trials engine is warmed up for 20-30 minutes to satisfy pre-experimental working conditions. During warmup air temperature and pressure is maintained at 21^0 C and 1 bar respectively. Coolant inlet and outlet temperature after warmup is 60^0 C and 85^0 C respectively while relative humidity and exhaust back pressure is kept at 40% and 45 mm of Hg respectively.
The 5-mode test cycle is mentioned below,

TABLE VI. TEST CYCLE OF 10.5 KW ENGINE

Mode	Engine Speed (rpm)	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

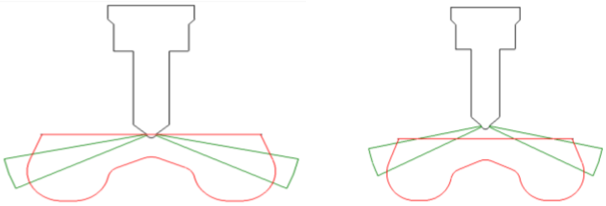
V. Fuel Spray Inside Bowl

The washer thickness available range varies from 1 mm to 4 mm as it ensures Nozzle tip protrusion from 4 mm to 1 mm respectively. The cone angle of 143 degree for 2 mm washer is best suitable since, at Top dead centre and 25 degree after top dead centre (for extreme case) the spray hits the torroidal area and lip section of piston bowl which is highly desirable. For 3 mm washer fuel spray is not that effective but comparatively it is better than 1 mm and 4 mm. This converges our trial typically for 143 cone angle and washer of 2 mm and 3 mm thickness only.



a) TDC b) 25 degree BTDC

Fig. 3. Fuel spray for 2 mm washer



a) TDC b) 25 degree BTDC

Fig. 4. Fuel Spray for 3 mm washer.

VI. EXPERIMENTAL RESULTS

The trial for different combination of static injection timing, high pressure piping and nozzle configuration is conducted for various thickness ranging from 1mm to 4 mm. nozzle is defined in terms of number of holes x cone angle x hole through flow (per 30 seconds) while pressure piping are expressed in form of outer diameter x internal diameter x length of piping. The static injection timing [15], for Baseline test is 15 degree BTDC while washer thickness and Nozzle configuration is 3.5 mm and 6x146x350. The static injection timing is summation of hydraulic lag and actual injection timing. In this case hydraulic lag is 7 degree. The value of smoke is 1.96 m^{-1} and power achieved is 10.5 kW at 1500 rpm for baseline trial.

A. *Trials for 2 mm washer*

1) *Trials for static injection timing*

Experimental trials are conducted for static injection timing ranging from 8 degree BTDC to 16 degree BTDC with progression of two degrees. This trial enables to confirm the injection timing.

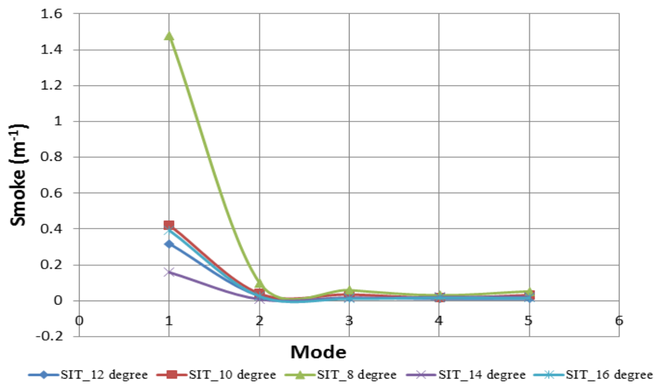


Fig. 5. Mode Vs Smoke

The Fig. 5 indicates that for static injection timing of 10 degree, 12 degree and 16 degree, smoke level varies between 0.3 m^{-1} to 0.4 m^{-1} at 1st mode. These values obey CPCB II norms but the minimum value of smoke is desirable as in case of 14 degree timing for which smoke limit is way below 0.2 m^{-1}. A wide margin at 1st mode for 8 degree injection timing is observed indicates high deterioration of exhaust smoke level due to retardation.

2) *Trials for high pressure piping*

For a constant static injection timing of 14 degree BTDC and Nozzle configuration of 6x143x285, the inner diameter and length of high pressure pipes are varied to comprehend the effect of piping change alone.

The smoke level rises due to increase in piping length and diameter since they alter the time required by fuel to reach injector end and resistance to flow of fuel from pump end. The piping parameter of 2.4x600 mm, 1.8x650 mm and 1.5x460 mm violates CPCB II norms as the range of smoke level attained is from 0.9 m^{-1} to 1.5 m^{-1} while 1.5x550 mm provides the least smoke level below 0.2 m^{-1} as indicated in Fig. 6.

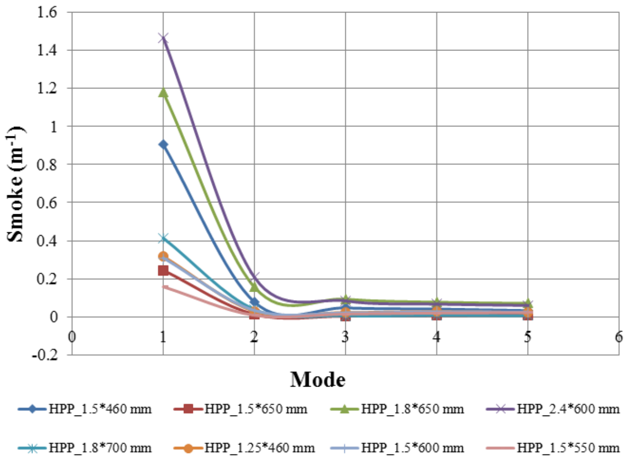


Fig. 6. Mode Vs Smoke

3) *Trials for nozzle configuration*

Nozzle trials are conducted to observe the change in smoke level when the flow injected in terms of cm^3 (cubic centimeter) is varied gradually. The nozzle having hole through flow of 325 cm^3, 350 cm^3 and 355 cm^3 are under observation.

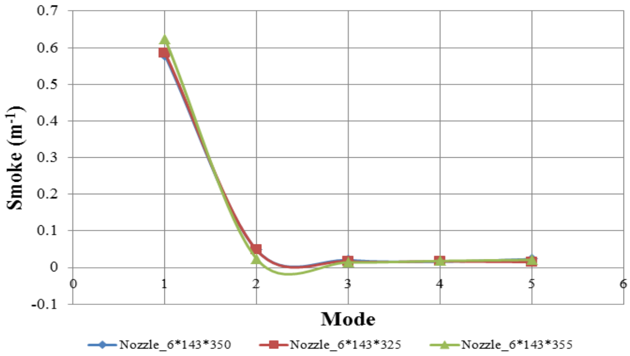


Fig. 7. Mode Vs Smoke

It is clear that as the volume of fuel injected per second increases the smoke level rises gradually this is due to the fact that the fuel is supplied in ample beyond necessarily required which leads to incomplete combustion. The Fig. 7 indicates that smoke level rises successively in order for 325 cm^3, 350 cm^3 and 355 cm^3 for 2 mm washer thickness and 1.5x550 mm pressure pipes.

B. Trials for 3 mm washer

1) Trials for static injection timing

The piping parameter of 1.25x550 mm, 1.5x550 mm and 1.5x600 mm are tested for 3 mm washer and injection timing of 14 degree BTDC. This piping indicates better results for 2 mm washer, so this trial enables to observe their effect on 3 mm washer.

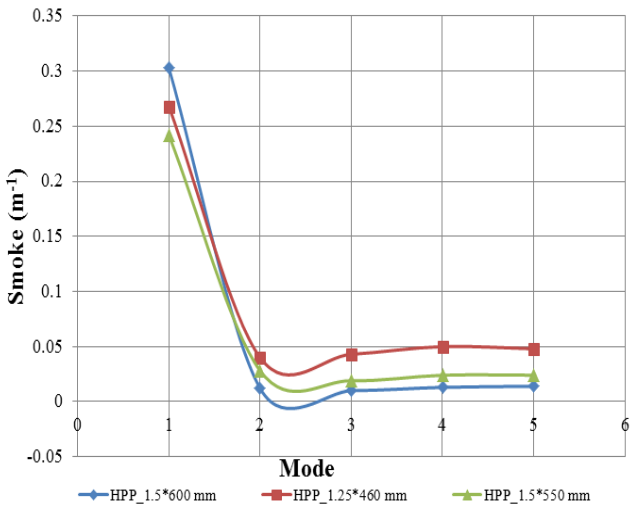


Fig. 8. Mode Vs Smoke

For 3 mm washer the value of smoke level for piping configuration of 1.5x550 mm and 1.5x600 mm is higher than its counterpart of 2 mm washer whereas for 1.25x460 mm configuration it is considerably low as indicated in Fig. 8.

VII. Conclusion

The effect of fuel injection system on smoke emission and the influence of various combinations of injection system is summarized as given below,

1) For a constant piston bowl parameters, the best results are obtained for 2 mm washer with nozzle configuration of 6x143x285 and 1.5x550 mm pressure piping.

2) The static injection timing of 14 degree BTDC is most suitable for 2 mm and 3 mm washer as the smoke level is minimum while for 8 degree BTDC the level is deteriorated.

3) For a washer thickness of 2 mm and 3 mm, the cone angle of 143 degree induces desirable fuel spray inside piston bowl.

4) The smoke level gradually rises for increase in supply of fuel for a constant hole number and cone angle of injector nozzle.

5) For a 14 degree BTDC timing, fuel flow of 9.5 cubic centimeters per second is desirable.

6) Decrease in piping inner diameter and length raises the smoke level slightly comparatively increases in both parameters negatively influences the smoke concentration whereas the level of smoke is median for constant diameter and change in length.

7) Excluding pressure piping values of 1.5x460, 1.8x650 and 2.4x600, rest all have smoke limit at 1st mode below 0.7 m^{-1}.

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