Performance Enhancement of a Single Cylinder Diesel Engine Through Optimization of Piston Combustion Bowl Geometry

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Abstract — The fuel consumption in the world is increasing at alarming rate as well as fossil fuels prices are increasing day by day and they will deplete within few decades. Hence the efficient combustion design is a key factor to current research and development of I.C engine. The piston bowl geometry, nozzle position and spray behaviour plays a predominant role in the engine performance operated with diesel. The most important role of Compression Ignition (CI) engine combustion chamber is to enhance the fuel-air mixing rate (swirl) in short possible time. The turbulence can be guide by the shape of the combustion chamber hence there is necessity to study the combustion chamber geometry in detail. The engine design with re-entrant cavity with central pip produces longer spray volumes, wider spray spreading and also introducing bottom comer radius helps to disperse the fuel. Re-entrant combustion chamber (TRCC) gives better fuel economy, lower hydrocarbons (HC), smoke, carbon monoxide (CO), greater brake thermal efficiency (BTE) and maximum cylinder pressure it is attributed to higher in turbulence, swirl, hotter surface of the re-entrant chamber and availability of oxygen with diesel. The double dish combustion chamber and chamber with the bump ring increases the kinetic energy, air-fuel mixing rate than that of conventional combustion chamber.

Keywords: Compression Ignition (CI) Engine; Combustion Chamber Geometry; Re-Entrant Combustion Chamber; Fuel-Air Mixing; Swirl and Turbulence; Engine Performance Optimization;

I. INTRODUCTION

Due to increase in usage of diesel-fuel vehicles has brought attention to environmental pollution, hence rapid depletion of fossil fuels increase in fuel demand and cost of the fuel. The diesel engine technology is very important factor to enhance economic and environmental implications. One of the main goals of developing diesel engine is better fuel economy and environmental friendly [2].

Increased pollution awareness and the consequent introduction of stringent emission norms throughout world are forcing engine manufacturers to continue to investigate for strategies for reducing emissions. During fuel injection in diesel engines, the air motion plays a very important role in fuel-air mixing, combustion and emission formation, especially around the compression TDC. Along with air motion, spray characteristics such as injector-hole diameter, injection pressure, spray angle, and injection timing, also have a significant effect on diesel engine combustion [5].

Sensitive Industries is a diesel engine manufacturer based in Rajkot produces different platform of diesel engine and rating. This engine primarily for agriculture application to drive centrifugal water pump. From last few years a strong competition from Chinese made diesel engine for such application in India market. To compete this Chinese product as well to meet market demand, Sensitive Industries continuously putting a lot effort to improve on BSFC and cost to power ratio as potential sales and marketing strategy. Again a durable and reliable product will further enhance to tap the potential market and compete with domestic & Chinese company. Here below about the application details of the product which will go through the development as planed in this project.

The in-cylinder air motion in diesel engines is generally characterized by swirl, squish and turbulence, which have a major impact on air–fuel mixing and combustion. Swirl motion of the air is usually generated due to the design of the intake port. A good intake port design will generate higher swirl and help to improve combustion. When there is swirl in the in-cylinder air, the swirl–squish interaction produces a complex turbulent flow field at the end of compression. This interaction is much more intense in re-entrant combustion chamber geometries. Further, intensification of swirl and turbulence are observed around TDC of compression. Around this time, most of the in-cylinder air is compressed into a smaller diameter combustion chamber. Thus, by conservation of angular momentum, as the radius of rotation reduces, the speed of rotation increases. Intensification of turbulence is due to the highly turbulent squish and reverse-squish motions of the air near TDC of compression. Because of these, usually two peaks in turbulence, one just before TDC and the other just after the TDC, are observed. In re-entrant chambers, the intensification of swirl and turbulence are higher when compared to cylindrical chambers. This leads to more efficient combustion which in turn causes higher NOX emission and. less soot and HC emissions. Better air–fuel mixing and combustion are possible with injectors having smaller hole diameter and higher injection pressures. Higher injection pressures produce smaller droplets which evaporate faster and mix rapidly with air. However, wall impingement of the spray and vapour leads to flame quenching and high soot emissions. Thus, a longer spray path without wall impingement is desirable for better combustion and low emissions. The relative position of the axes of the piston bowl and injector with respect to the cylinder axis also plays a role in in-cylinder mixture motion and combustion. Injectors with finite sac volume at the tip are usually associated with large fuel droplets towards the end of injection which lead to higher HC and soot emissions. In re-entrant chambers, retardation of spray can be used to control [4].

The fundamental needs of sensitive to meet the market expectation as primarily fuel economy. To compete with market it also required to be more cost competitive. Hence the cost to power ratio to be reduced. Other secondary requirement is to make the engine more durable and reliable so extended warranty etc can offer to end customer as marketing strategy to compete with domestic product. Diesel engine having wide use in agriculture, transportation, power generation and many industrial appliances.

It also to be in mind with very minimum changes in design or material cost increase during the development. To cater the Genset market the same engine should capable with very minimum hardware changes or retrofit to meet the CPCB-2 norms in India. We mainly focus on development of air swirl. A higher swirl intensity will improves fuel and air mixture which impacts on combustion efficiency.

As We in Prakarsa also working to develop certain type of solution i.e. Exhaust gas recirculation valve either mechanical control or electronic control which also could be tried on the same engine as probable solution. This feasibility study is additional to the subject project and will only cover the probable option. The subject project objective will not be linked with the outcome of the results of Objective. Proposed Technical approach the increase in power the more fuel need to be burnt which can be done with increase in air flow i.e. by increase by cylinder capacity, volumetric efficiency. Also increase in BSFC also helps to get more power output.

II. BACKGROUND

Sensitive diesel company has developed a single cylinder base engine primarily for the agriculture market and have future plan to enter in Genset market with required modification in the engine to comply with CPCB-2 norms. The base engine current performance and its compatible water pump are still under development. Here, we have approached to the customer and offer possible solution to meet the market requirement.

A. *Engine component production scenario*

In an automobile industry the component production range includes 31% engine parts, 19% drive transmission and steering parts, 12% suspension and braking parts, 10% electrical parts, 12% equipment, 9% body and chassis, and 7% other parts. Based on this scenario we can predict the current importance of the engine parts that are heavily impacts the performance factor of the engine. In the automobile sector engine having more productionised parts so we decide to more focused on engine performance. And it also major impacts on the environment pollution, the world's government giving targets for each country to minimize pollution range per year which reduces the ozone layer of the earth.

B. *Application details*

The company is going to provide us an engine it applies to agricultural pump which having below specific conditions. As we are started with the collecting basic details of the engine from the customer at his end. The basic engine parameters and its application details are mentioned below. That includes engine load, rating in hp, fuel economy, service details, its maximum life, and engine details like cubic capacity, compression ratio, no of piston rings, speeds like fly up speed and idling speed. This data plays an important role before doing to start our actual work. After collecting the prioritised data we start to work on our important factor that is Piston.

Parameters	Application details
Gross load	20/12lps@10m duty head (3.7/2.25hp)
Rating	5/4/3hp@2600/2000/1800
Max torque	14/14.4/12Nm@2600/2000/1800
Fuel economy	265-275 g/kWhr
Oil service interval	250 hrs(soot loading)
Oil consumption	0.5% sfc
B50 life	1000hrs
Bore × stroke	82×65 mm
Cubic capacity	343cc
Compression ratio	16.95
Piston ring	4NOS
Fly up speed	2850 rpm (max.)
Idling speed	1600 rpm

Table 1: Engine parameter and application details

III. MATERIALS AND METHOD

The materials use for the piston is aluminium alloy with 4-5% silicon. This material is equivalent to KS1275. Piston was run 50hrs for endurance test and the results are interpreted. As piston reciprocate in the liner, hence to see the distortion effect, FEA on liner done, and modification suggested to increase liner thickness to 106 mm OD from 101 mm. Engine test shows improvement in performance. The piston will be designed in catia software with modified bowl in-line with addition required spares and engine test is carried out for the results.

Design of Piston for 85mm and 82 mm, Piston ring (top, 2nd, Oil ring) for both 85mmand 82mm Diameter piston. Here more engine component like Piston ring (top, 2nd, Oil ring) for both 85mmand 82mm Diameter piston, Cam shaft, Linear drawing for 85mm Diameter bore, Block to Linear dimensional specification for 85 Diameter bore, Injector and fuel pump are accommodated as per requirement.

Performance testing and optimization of all the rating as per table for agriculture application. Performance & emission feasibility study for CPCB -2 norms for 3.5Hp@1500 rpm.

The delivery of the research based on the trials likely 500hrs endurance test, B50 life of 1000hrs, durability, reliability, Oil consumption & Oil change interval, soot level, component wears & tears analysis and Failure mode study.

A. *Methodology*

The research method used in this study was experimental. This research encourages experiments to determine the influence of certain variables. First step is to define problem statement there after industrial visits are most important factor comes in picture for actual output and feedback of the engine. We prefer some literature survey for basic study of the available material. Then we go for piston bowl design calculation and modelling of some different piston bowls based on our functionality. For the theoretical purpose apply CFD analysis in order to get the effect of bowl geometry on in-cylinder airflow, air/fuel mixture distribution and soot formation. If results satisfy the output requirement of swirl will allow to go for manufacturing if fails then we go for repeating the cycle.

This study focuses on the experimental results of the engine output. The final output if testing satisfy the results on the other hand go with repetition of cycle. The below results are self- explanatory and accompanied. Learn from our results and was did the progress from design stage to final step until we get the satisfactory outcomes from the experiment. At the last we compare the readings taken at initially and after doing modification with our modified piston. For this everyone should know the diesel engine operation cycle of Suction-Compression-Expansion-Exhaust. Swirl plays vital role at the end of suction stage and during the compression stage.

B. Engine photographs

Here below are some photos of the entertained engine that we are going to work and have to increase the performance and specific fuel consumption. It will implies for agricultural as well as Genset application. So it will be required to optimize the engine and pump hardware accordingly for the different application as well. Examine the effect of a variation in the intake–generated air swirl and in-blow swirl ratio, on combustion and exhaust emission. As additional project scope as feasibility study of the engine to find suitable solution for Genset application meeting CPCB-2 emission norms. The feasibility will cover possibilities of max rating @1500 rpm and suitable hardware to meet required level of emission by means of engine hardware improvement or with external solution like EGR system. Primarily focused on the fuel economy. It gives clear title idea for the research that work to impinge our focus as mentioned above how important to improve the specific fuel consumption not only for money it also enhance the atmosphere our nature and limited resource of energy also. Practically we takes readings before and after the piston bowl modifications on below shown actual engine, the main factor is air-fuel mixture and it enhanced by the air swirl motion of the engine

Fig. 1: Actual engine photographs

C. Current base engine performance simulation

From the customer input data, as mentioned in table 2 a simulation runs to understand the engine performance. The below parameter as received from customer taken as input to the simulation. The objective of the development is to improve the power output from same weight of the engine i.e. 10-15% more power at the speed range 1500 to 2600 rpm. Performance testing and optimization of all the rating as per below details for agriculture application. For the simulation takes external professional CFD support for the output results mentioned in table 3. We get clear idea about the volumes involved in the cycle and air flow in the cylinder which suitably helps for the design stage of the piston.

Parameter	Symbol	Unit	Value	Parameter	Symbol	Unit	Value
Bore	D	mm	82	SOI	SOI	Deg	1
Stroke	L	mm	65	Inj Duration	DOI	Deg	13.15
No of Engine Cylinder	n	nos	1	Blow By	Bb	kPa	0.95
Con Rod Length	Lcr	mm	118	Ambient Air Pressure		kPa	96
Bowl Volume	Vc	cc	18.5	Ambient Air Temperature		C	28
No of Valves	Nv	nos	2	Intake manifold pressure	w	bar	0.94
Gasket Compression Thickness	Tg	mm	0.95	Intake manifold Tem	T1	m/s	303
Gasket dia	Dg	mm	82.5	Specific Heat	Y	m2	1.3
Piston Top land Step length	Lsp	mm	10	Specific Heat constant Volume	Cv	kj/kgk	0.713
Piston top land eff dia	Dt	mm	82	Specific Heat constant pressure	Cp	kj/kgk	1.005
Valve dia	Dv	mm	32	Fuel calrific Value	CV	kj/kgk	42500
Valve depth (Piston + head)	Dd	mm	1.5	Fuel Density	Fd	kg/m3	820
Swirl Ratio			1	Air Fuel ratio			18.61
Piston Protrusion	Lp	mm	0.5	Fuel Rate	Fr	kg/hr	1.092
Inlet valve open@BTDC	IVO	Deg	15	Rated Speed	N	rpm	2600
Inlet valve close@ABDC	IVC	Deg	38	Timer travel at rated Speed		deg	5
Exhaust valve open@BBDC	EVO	Deg	42	Air Flow Rate	Fb	kg/hr	20
Exhaust valve close@ATDC	EVC	Deg	11	Frictional HP	FHP	kW	10
Engine Stroke			2	Heat dissipation loss		kw	6
Volumetric Eff (Calculated)			0.78	Swirl factor		kw	0.9753

Table 2: Engine performance simulation – Input

Parameter	Symbol	Unit	Value	Parameter	Symbol	Unit	Value
Cylinder Volume	Vs	cc	343	Effectice expansion ratio@EVO	ER	nos	10.86
Total Clearance Vol	V2	cc	26	Temperature after expansion @EVO	T5	K	1081.5
Volume @BDC	V1	cc	369	Pressure after expansion@EVO	P5	bar	1.323
Compression Ratio	CR	nos	14.2	Temperature after expansion @BDC	T6	K	983.74
Effective Swept vol @IVC	Vs'	cc	316	Exhaust Stack temperature	T7	C	594.48
Effective Compression ratio	CR'	nos	13.1	Heat Addition	Qa	kJ	2167.8
Compression Pressure	P2	bar	25.7	Heat Rejection	Qr	kw	0.415
Effective Compression Pressure	P2"	bar	24.4	Thermal Efficiency	Πτη%	%	0.441
Compression Temp	T2	K	656	Indicated power	Pihp	kW	5.682
Fuel burnt in constant volume	mfcv	kh/hy	0.1	Rated Power	P	KW	4.03
density of air		m	871	Rated Power (Corrected)	P	kW	4.021
Temp after constant vol heat addition	T3	m	871	Rated Power (Corrected)	P	Hp	5.401
Air Flow		kg/hr	20.3	Engine Torque (Corrected)	T	N	14.774
Pressure at 1st Heat add	P3	Bar	30.8	BSFC@2600 (Corrected)	BSFC	gm/kWhr	271.56
Temperature after heat addition 2nd	T4	N	2876	BSFC@2600	BSFC	gm/hphr	199.73
Pressure	P4	N	30.8	BMEP	BMEP	bar	5.418
Effective Specific Heat ratio	Y	kJ/kg K	1.4	Volumetric Efficiency	ηv%	%	74.971
Effective Specific Heat ratio	Y	kJ/kg K	1.3	Injection Qty/st		mm3/.st	17.073
Volume after constant pres heat additi	V4	CC	30.9	Injection Qty/st		mg/st	14
Volume after expansion	V5	CC	336	ignition delay			0.907
Cut off Rtaio	ρ	nos	3.3	Static timing			15.6
Pressure Ratio	pr		1.26	Bore/Stroke Ratio			1.262

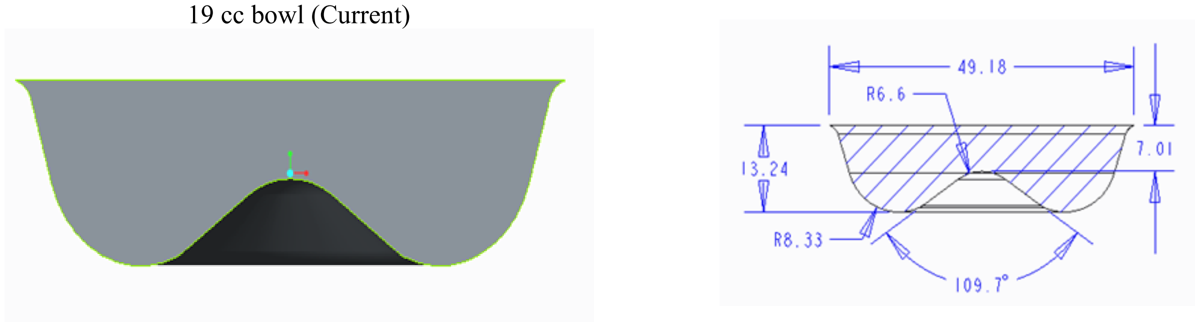
Table 3. Engine performance simulation - Output

Here above are some outputs from the performance simulation that we study for our cylinder and piston bowl volume requirement. Considering the above output data we develop some bowl shapes for the test results.

IV. Bowl Shapes

Below are different bowl shapes designs was manufactured at Rajkot head and at Kolhapur head. For the design purpose will use catia V5 software. The performance test result will compare with base engine test result. The current base engine performance simulation plays important role while the design of bowl to consider the volume threats. Accomplishment of the design based on the CFD simulation. The output test results for those new pistons with modified bowl shapes are mentioned below.

19 cc bowl (Current)



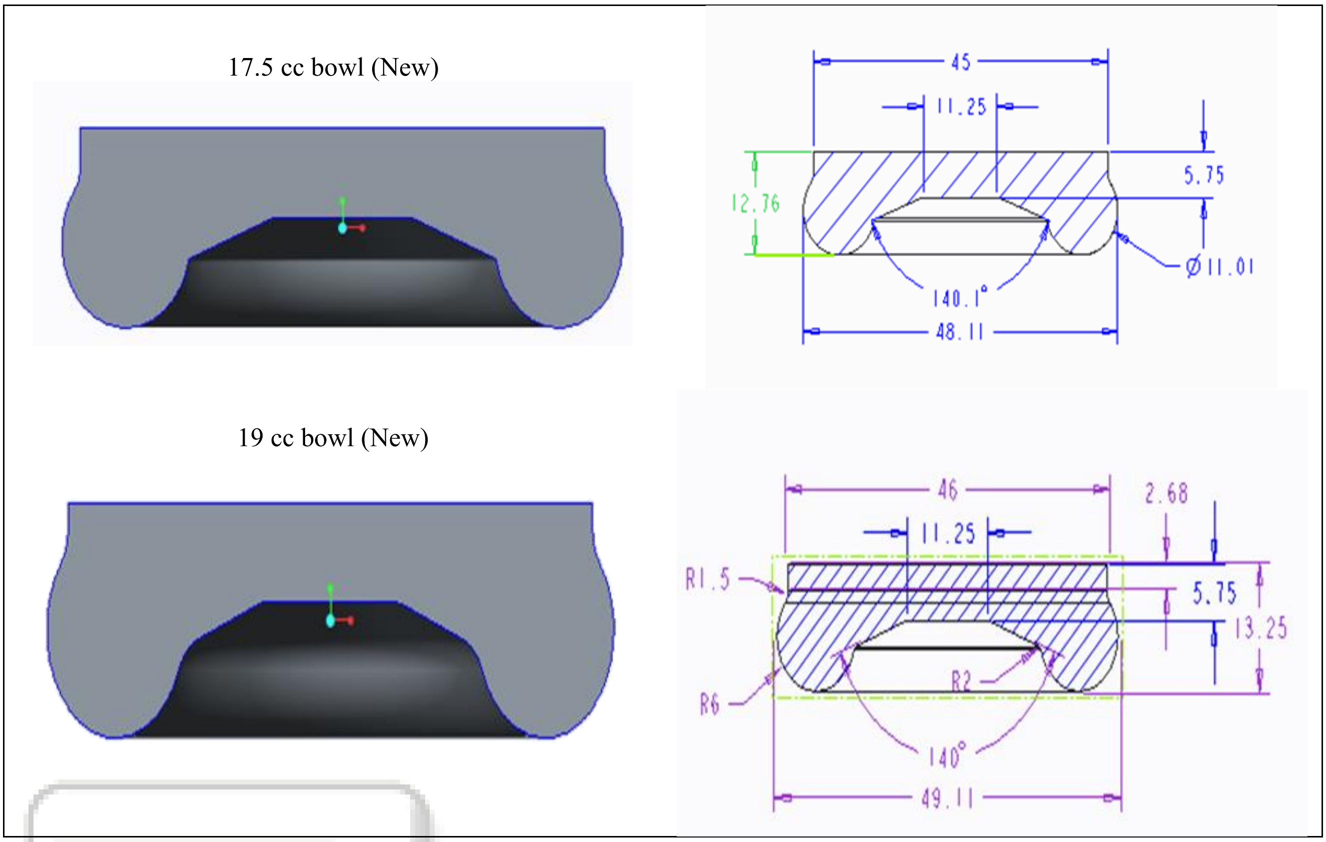


Fig. 2: Different Bowl Shapes

A. *Equations*

According to the inputs from customer and output results of simulation we use mathematical method to find indicated power, brake power, heat dissipation from the piston head and axial thickness of piston rings which are involved in heat dissipation.

Design input consideration

1) Cylinder bore = 82 mm;
2) Stroke = 65 mm;
3) Maximum gas pressure = 6 N/mm2;
4) Indicated mean effective pressure = 0.75 N/mm2;
5) Mechanical efficiency = 80%;
6) Fuel consumption = 0.15 kg per brake power per hour;
7) Higher calorific value of fuel = 42 × 103 kJ/kg;
8) Speed = 3000 r.p.m.
9) Piston head or crown
10) The thickness of the piston head or crown is determined on the basis of strength as well as on the basis of heat dissipation and the larger of the two values is adopted.
11) We know that the thickness of piston head on the basis of strength,
12) $t_H=\sqrt{\frac{3pD^2}{16\sigma_t}} = \sqrt{\frac{3x6x82^2}{16x35}}$ =15.7 say 16 mm
13) Taking σt for cast iron = 38 MPa = 38 N/mm2
14) Since the engine is a four-stroke engine, therefore, the number of working strokes per minute,
15) n = N / 2 = 3000 / 2 = 1000
16) and cross-sectional area of the cylinder,
17) $A=\sqrt{\frac{\prod D^2}{4}} = \sqrt{\frac{\prod x82^2}{4}}$ =72.67 mm^2
18) We know that indicated power,
19) $IP=\frac{P_m.L.A.n}{60} = \frac{0.75x6x.6500.1500}{60}$ =6kw
20) ∴Brake power, BP = IP × η_m = 12.27 × 0.8 = 9.8 kW ...(Q ηm = BP / IP)
21) We know that the heat flowing through the piston head,
22) H = C × HCV × m × BP
23) = 0.05 × 42 × 103 × 41.7 × 10–6 × 9.8 = 0.86 kW = 860 W
24) Taking C = 0.05
25) ∴Thickness of the piston head on the basis of heat dissipation,
26) $t_H=\frac{H}{12.56k(T_c-T_H)} = \frac{860}{12.56x46.6(220)}$ =6.7mm
27) Q For cast iron, k = 46.6 W/m/°C, and TC – TE = 220°C
28) Taking the larger of the two values, we shall adopt
29) t_H = 16 mm
30) Since the ratio of L / D is 1.25, therefore a cup in the top of the piston head with a radius equal
31) to 0.7 D (i.e. 70 mm) is provided.
32) Radial ribs
33) The radial ribs may be four in number. The thickness of the ribs varies from tH / 3 to tH / 2.

34) ∴ Thickness of the ribs, tR = 16 / 3 to 16 / 2 = 5.33 to 8 mm
35) Let us adopt tR = 7 mm Ans.
36) Piston rings

Let us assume that there are total four rings (i.e. nr = 4) out of which three are compression rings and one is an oil ring.
We know that the radial thickness of the piston rings,

t1 = 3.4

Taking pw = 0.035 N/mm2, and σt = 90 MPa
And axial thickness of the piston rings

t2 = 0.7 t1 to t1 = 0.7 × 3.4 to 3.4 mm = 2.38 to 3.4 mm

Let us adopt t2 = 3

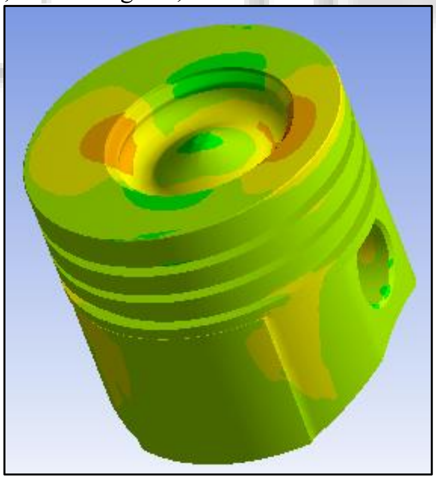
B. Proposed engine components and its features modification

For the current work the project requires multiple components to be work-out are as above in line with piston, ring pack, oil ring, camshaft, injector, fuel pump, linear. But we mainly focus on the piston and piston bowl modifications to develop the engine. To improve the sfc of engine it is necessary to customise all engine components. Keeping the piston rings thickness as per our calculations and piston bowl shapes with our new designed and manufactured bodies. The below mentioned our test results from our modified piston bowl shapes.

V. RESULTS AND DISCUSSION

A. FEA analysis of critical current engine parts

The structural study on the current engine part like piston, crankshaft, Cylinder liner and connecting rod done and the result shows further increase in power output without any modification is possible. No changes are required in crankshaft, connecting rod, block or Head.



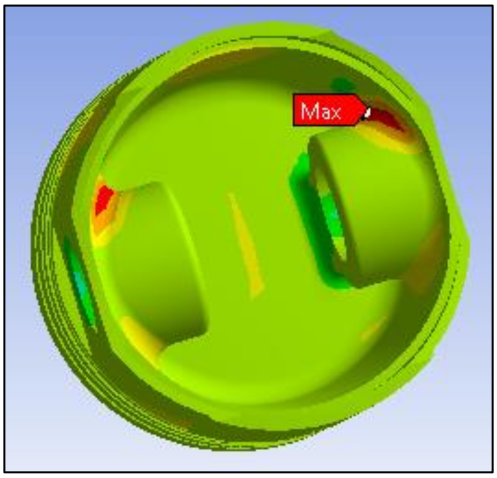


Fig. 3: Piston Structural Analysis

The design pressure considered 60 bar max and material is Aluminium alloy with 4-5% silicon. This material is equivalent to KS1275.Piston FEA results shows stress level at piston rim and pin bore area are within acceptable limit.

The piston was run for 50 hrs. Endurance test and found structurally satisfactory. No visual crack or abnormal wear found.

B. Liner stress analysis

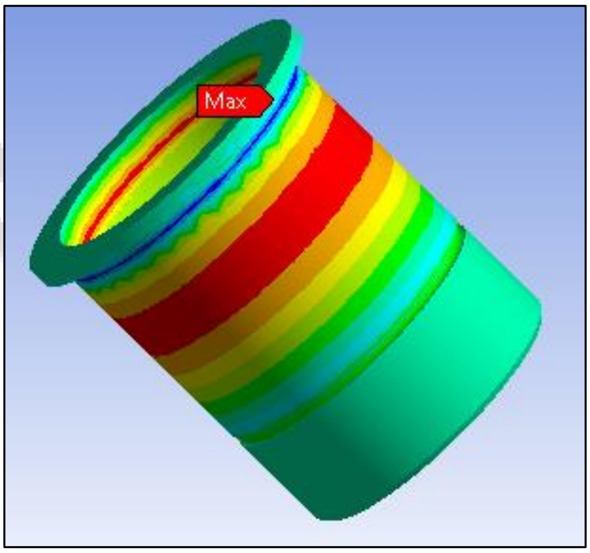


Fig. 4: Liner Stress Analysis

As piston reciprocate in the liner, hence to see the distortion effect, FEA on liner done, and modification suggested to increase liner thickness to 106 mm OD from 101 mm. Engine test shows improvement in performance.

C. Piston bowl optimization

There are five different type of bowl tested for different parameter like sfc, smoke, loading capacity etc. Here below is finalised for production. Improve the fundamental understanding of the effect of piston bowl geometry on diesel engine combustion, exhaust emissions, and other performance parameters.

Description	Old	Modified at primary	19 cc bowl (Current)	19 cc bowl (New)	17.5 cc bowl (New)
Bowl Volume cc	16	15	19	19	17.5
Compression Ratio	19cc	20	16.5	16.5	17

Cold Starting	7	9	5	3	3
Smoke starting	Very High	low	low	low	
Smoke at full Load	Very High	Yes	No	No	TBD
SFC@2200 Gm/kwhr	>320	>300	265	282	TBD
NOX	NA	NA	NA	NA	TBD

Table 4: Piston bowl optimization

D. Test result

As per the above table we manufactured piston with modified design and take some readings of the engine specific fuel consumption at different engine RPM as below. The above plan is tentative estimation of project task may changes based on actual activities etc.

bowl shapes	2000	2200	2600
Hemispherical bowl 14cc	310	325	356
Shallow 14 cc with centre mount	288	294	319
Shallow 14 cc re-entrant	301	306	326
Shallow 16.5 cc bowl	278	285	298
Shallow19 cc bowl	276	282	292
19.5 cc re-entrant	274	278	298

Table 5: Specific fuel consumption with different bowl shape at different engine rpm

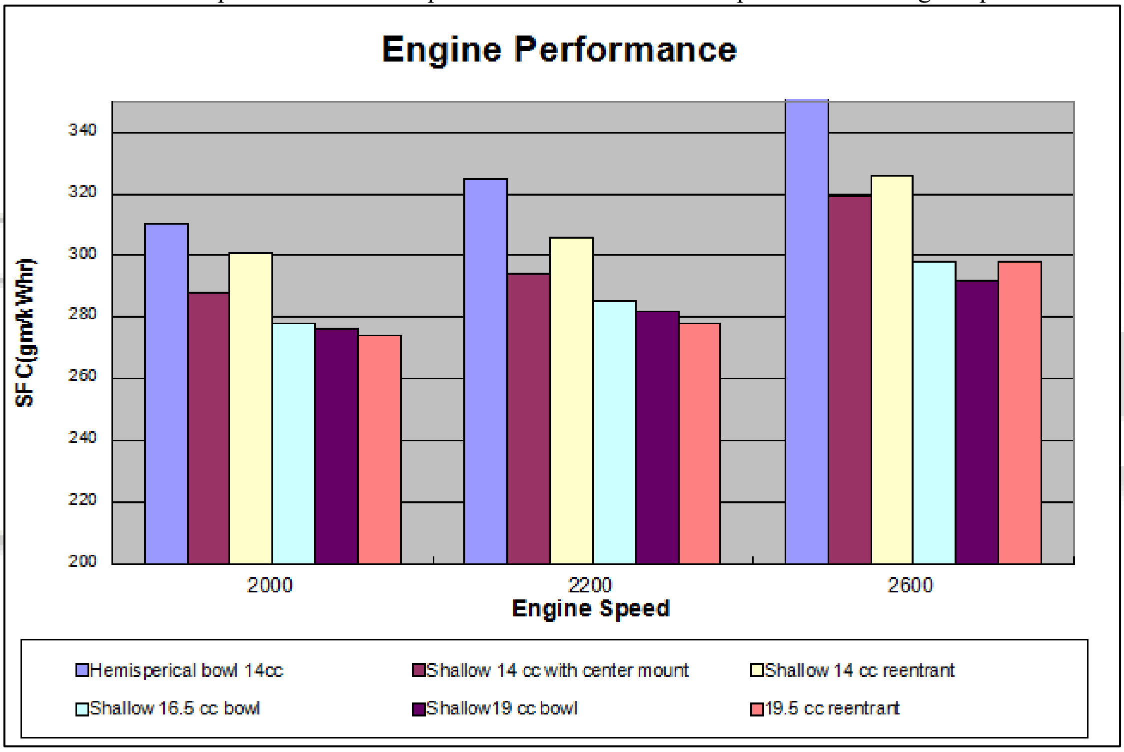


Fig. 5: sfc vs engine speed

Results: Increased in bowl volume is giving good improvement in SFC. 19 cc is found to be optimized bowl volume without any cold starting issue. Re-entrant shape is giving better SFC at lower engine speed and shallow bowl at higher speed. This is due to turbulence produced at higher speed due to re-entrant shape.

Bowl shapes	2000	2200	2600
+Shallow19 cc bowl with skirt	276	282	292
Shallow19 cc bowl without skirt	271	275	285

Table 6: Specific fuel consumption with different 19 cc shallow bowl with different skirt at rpm

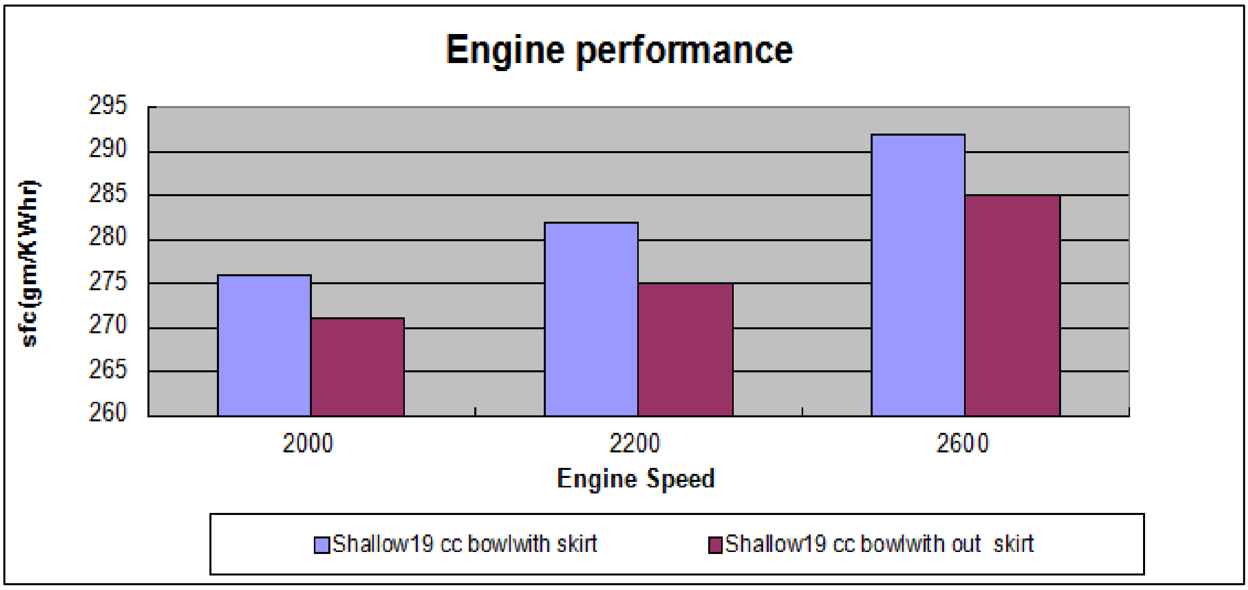


Fig. 6: sfc vs engine speed

Results: Lower skirt of the piston lower the viscous friction loss hence improves in SFC. Engine with lower skirt piston run for endurance for any wear in piston skirt, it is found that with CH4 oil grade no abnormal wear found on the skirt area both thrust and anti-thrust side.

bowl shapes	2000	2200	2600
Shallow19 cc bowl without skirt and 4 nos Ring	271	275	285
Shallow19 cc bowl without skirt and 3nos Ring	265	267	281

Table 7: Specific fuel consumption with different 19 cc shallow bowl numbers of rings at different engine rpm

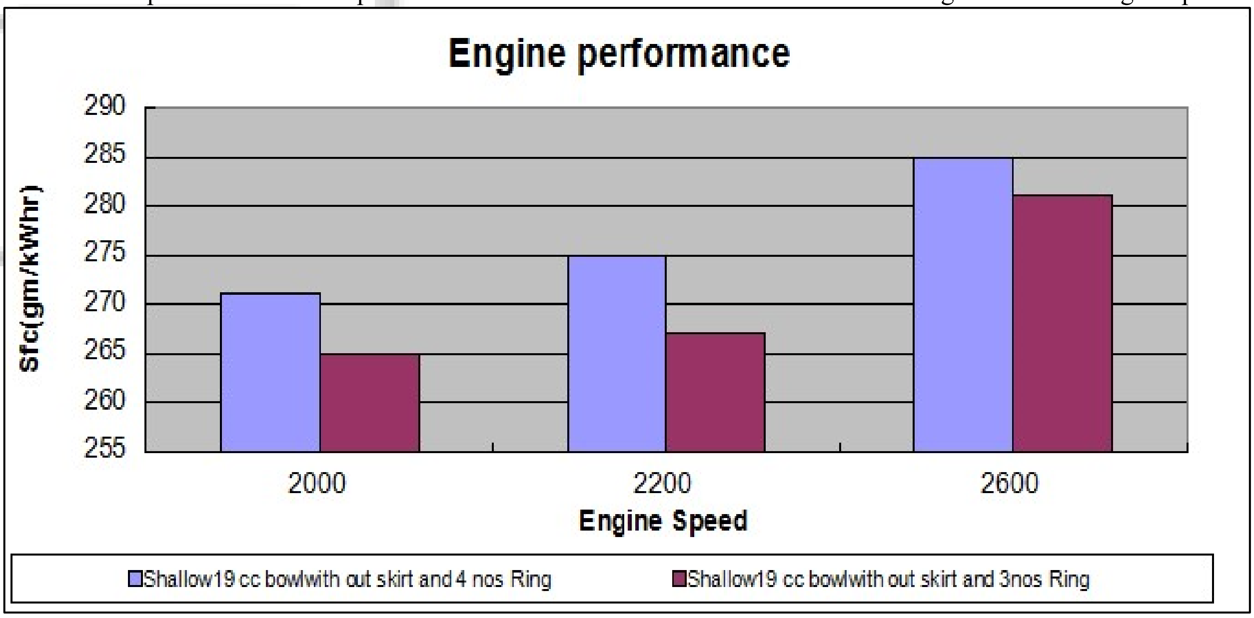


Fig. 7: sfc vs engine speed

Results: Less no ring and ring tension also lower the viscous friction loss hence improves in SFC. Engine with lower less piston rings run for endurance for any wear in piston skirt, it is found less no ring improve liner life.

bowl shape	2000	2200	2600
Shallow19 cc bowl without skirt and 3nos Ring	265	267	281
Shallow19 cc bowl without skirt and 3nos Ring & Effective compression ratio	261	264	278

Table 8: Specific fuel consumption with 19 cc shallow bowl with different CAM shaft impact of compression ratio at different engine RPM.

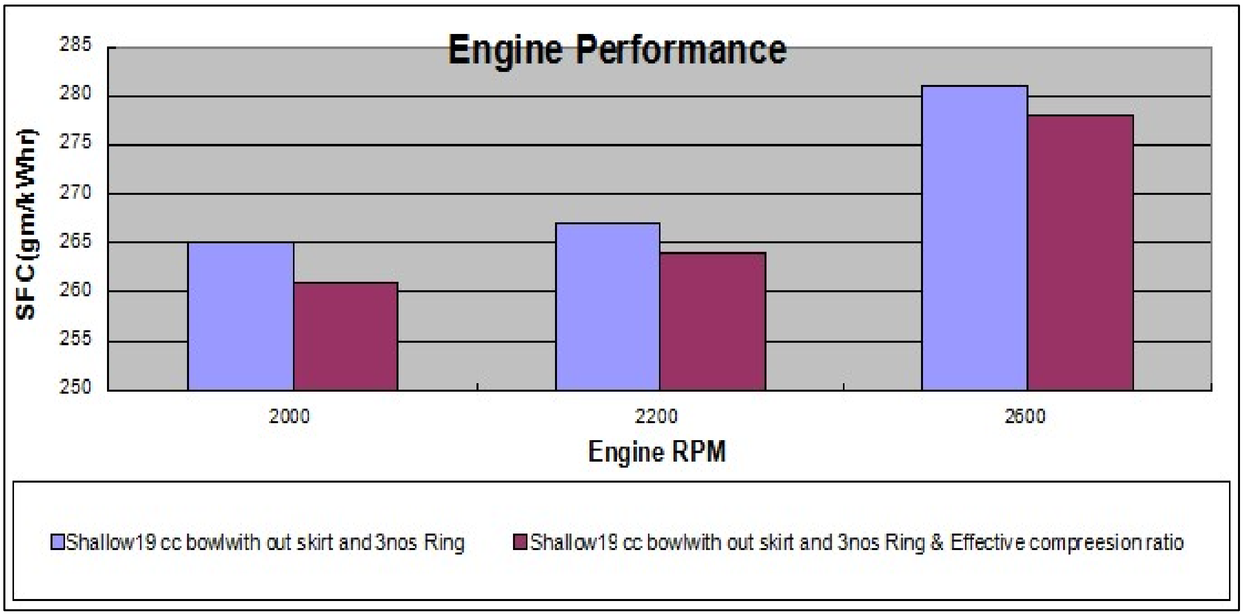


Fig. 8: sfc vs engine speed

Results: Valve timing controlled by camshaft. Intake valve closing impact on increase on compression ratio. Hence thermal efficiency improves.

bowl shape	2000	2200	2600
Shallow19 cc bowl without skirt and 3nos Ring & Effective compression ratio	261	264	278
Shallow19 cc bowl without skirt and 3nos Ring & Effective compression ratio with thick liner.	255	259	275

Table 9: Linear improvement so better piston and ring contact

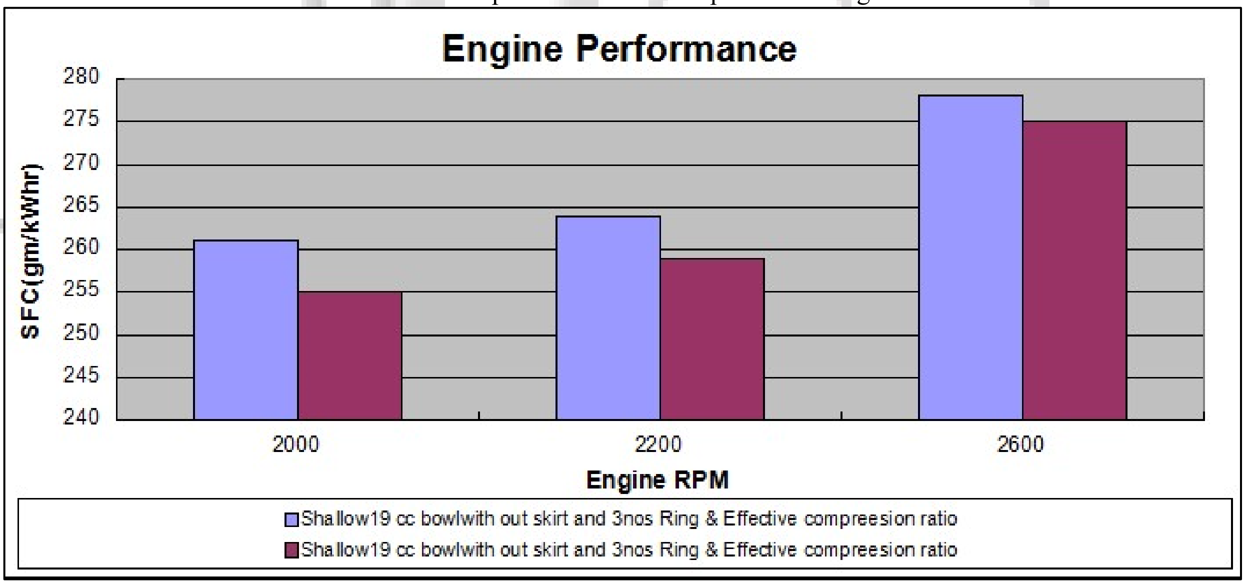


Fig. 9: sfc vs engine speed

Results: Linear improvement so better piston and ring contact hence better blow down losses. Hence lower in fuel consumption. As speed increases specific fuel consumption increases.

bowl shapes	2000	2200	*2600*
Shallow19 cc bowl without skirt and 3nos Ring & Effective compression ratio @3.5	255	259	*275*
Shallow19 cc bowl without skirt and 3nos Ring & Effective compression ratio@4	251	254	*272*

Table 10: Load optimization

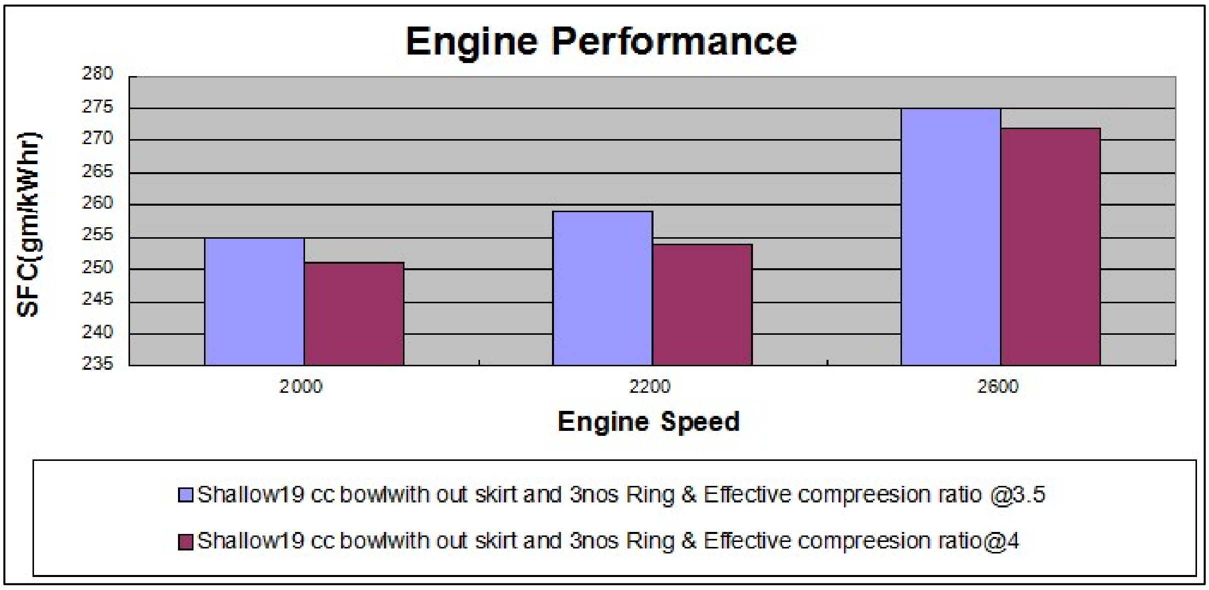


Fig. 10: sfc vs engine speed

Results: As diesel engine performance depend on the loading on the engine. Increase in load found to be better for fuel consumption. Hence the rated power declaration changed to get the benefit of lower fuel consumption at optimized load.

Different parameter	2000	2200	2600
Power Curve	2.98	3.3525	3.725
Sfc	251.7	255.0	271.1
Airflow	18	19.5	22
fuel flow	0.75	0.855	1.01
Air fuel Ratio	24	23	22

Table 11: Engine final performance curves with finalized hardware

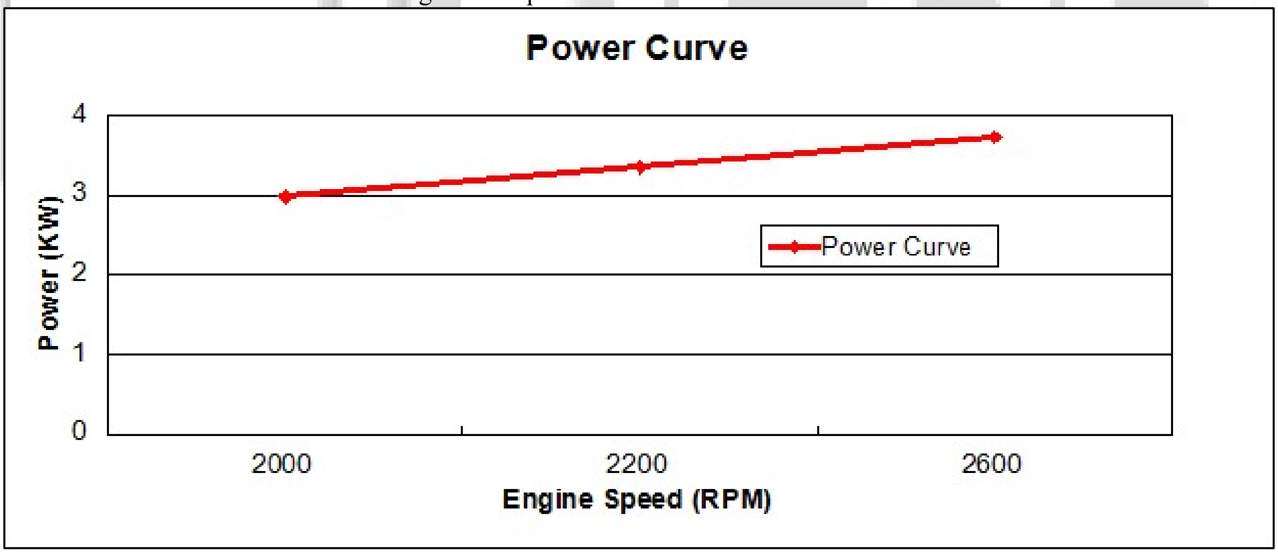


Fig. 11: Power (KW) vs engine speed

Result: According to table as RPM of engine increases power output and specific fuel consumption also increases

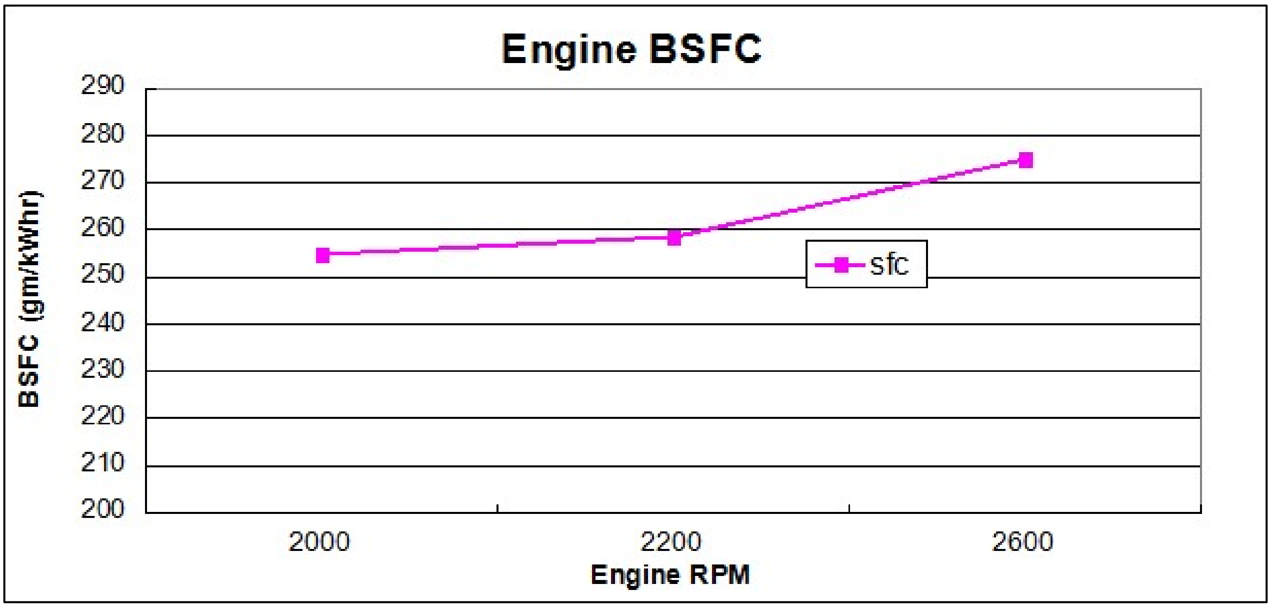


Fig. 12: Bsfc vs engine rpm

Result : Engine speed is directiy propotional to break specific fuel consumption.

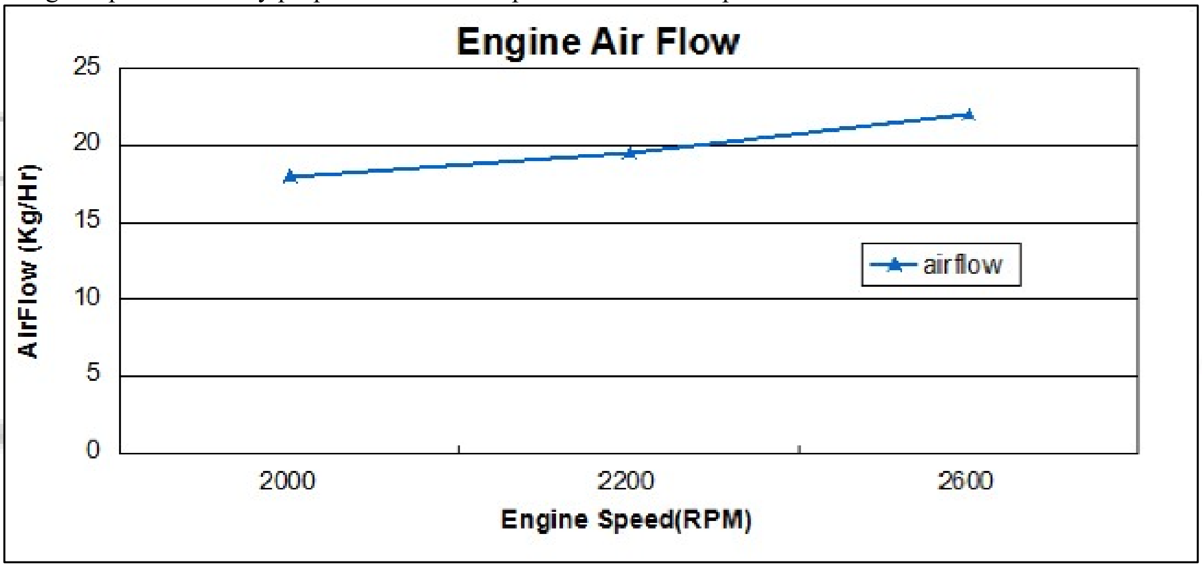


Fig. 13: airflow vs engine rpm

Result : Above graph shows when engine speed increases then air flow also increases.

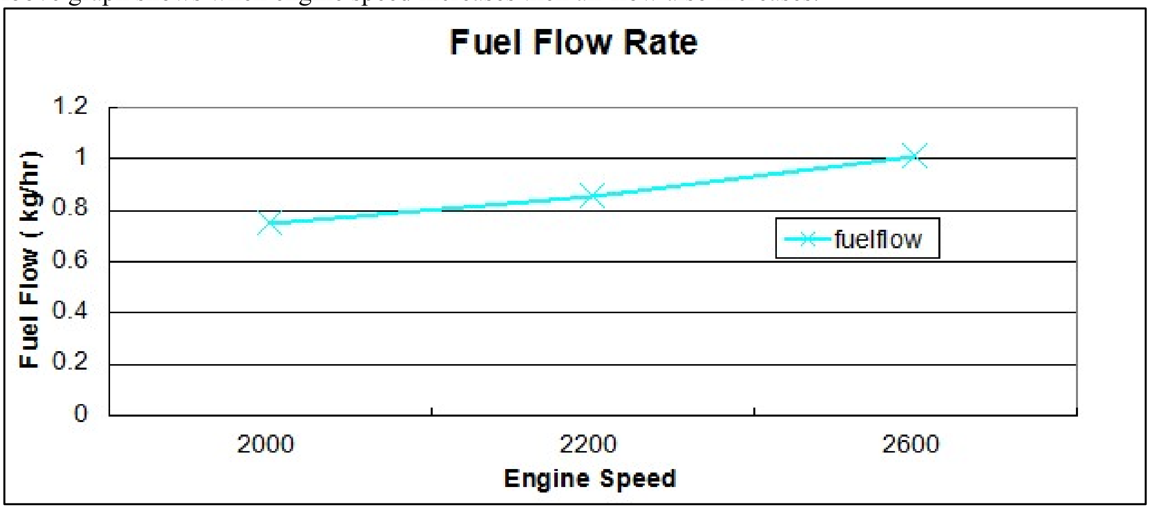


Fig. 14: fuel flow vs engine speed

Result : Increse in engine speed results increase in fuel flow.

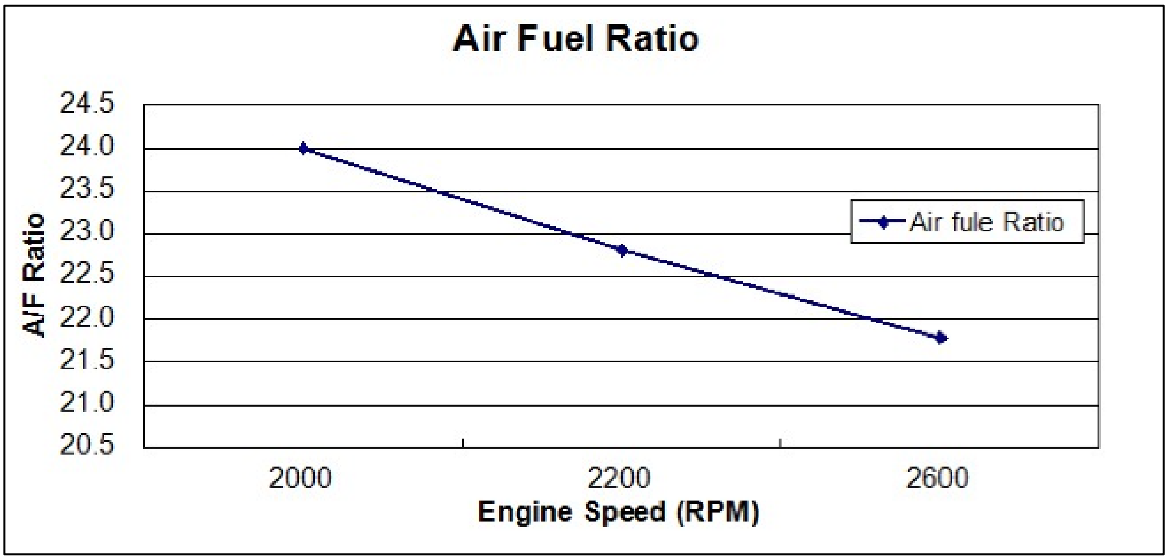


Fig. 15: air-fuel ratio vs engine speed

Result : Air-fuel ratio is inversly propotional to engine speed.

E. Oil consumption

In the conclusion of engine test it is calculated that whatever oil consumption in the engine is very important. It was affecting on the sfc of the same.

Design Contamination	Oil consumption in ml
Regular Piston, 4 Nos Ring, Linear, Oil	140
Regular Piston, 3 Nos Ring, Linear, Oil	75
Regular Piston, 3 Nos Ring, Modified Linear, CH4 Oil	32

Table 12: Oil consumption in ml

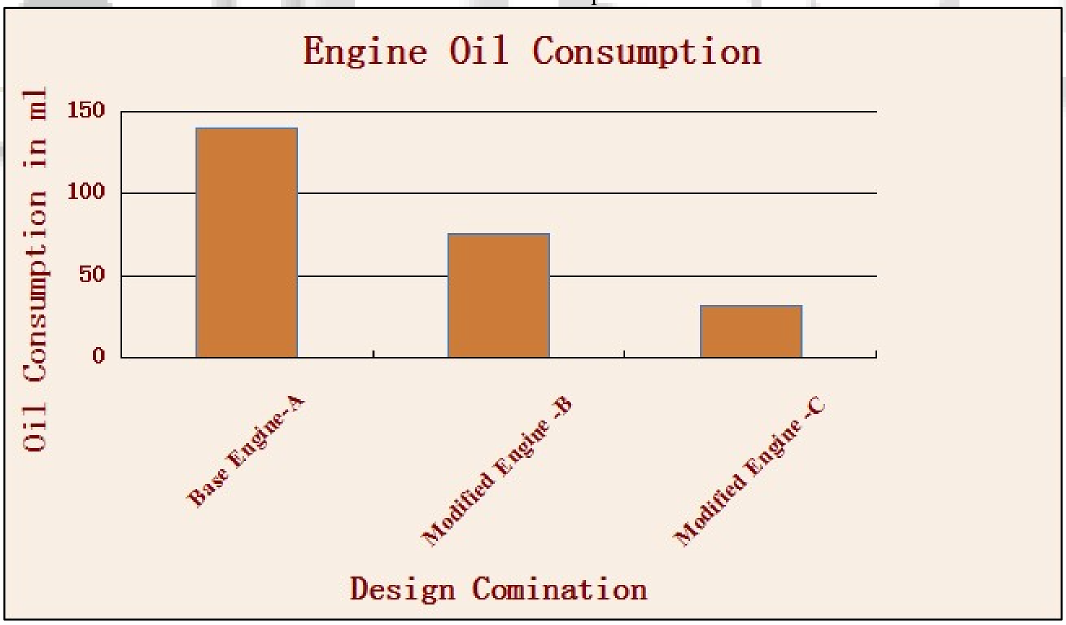


Fig. 16: Oil Consumption (Ml) Vs. Design Contamination

Results: The piston and its related component development improves the specific fuel consumption, oil consumption and smoke capacity. Hence with all this production released in the market successfully.

VI. CURRENT BASE ENGINE PERFORMANCE DETAILS

A base engine performance also taken at Customer end. This data is only referred to understand the effective part load zone of the engine. It shows that the engine need to be designed or suitable application need to be optimized as such that power drawn from the engine should not go below 90% of the engine rated power output to get maximum fuel economy benefit.

Hence it is suggested to design the pump perfectly optimized with engine output. Also as customer want the engine to be developed for two different application i.e. Agriculture pump and Genset application. So it will be required to optimize the engine and pump hardware accordingly for the different application as well.

% Load	RPM	Load	FC_ Time	Power (HP)	Power(kW)	Fuel Rate	SFC
100%	2616	4.2	135.15	5.4936	4.038	1092.120	270.5
88%	2594	3.7	157.5	4.7989	3.527	937.143	265.7
67%	2592	2.8	188.81	3.6288	2.667	781.738	293.1
46%	2590	1.95	232.34	2.52525	1.856	635.276	342.3
100%	2400	4.2	155.06	5.04	3.704	951.890	257.0
88%	2400	3.7	173.4	4.44	3.263	851.211	260.8
67%	2400	2.8	210.63	3.36	2.470	700.755	283.8
46%	2400	1.95	256.91	2.34	1.720	574.520	334.0
100%	2200	4.24	163.89	4.664	3.428	900.604	262.7
87%	2200	3.7	186.4	4.07	2.991	791.845	264.7
66%	2200	2.8	225.84	3.08	2.264	653.560	288.7
46%	2194	1.95	289.81	2.13915	1.572	509.299	323.9
100%	2000	4.2	186.28	4.2	3.087	792.356	256.7
88%	2000	3.7	207.81	3.7	2.720	710.264	261.2
67%	2000	2.8	281.5	2.8	2.058	524.334	254.8
46%	2000	1.95	324.56	1.95	1.433	454.770	317.3

Table 13: Current Base Engine Test Details

A. Engine performance simulation after possible changes

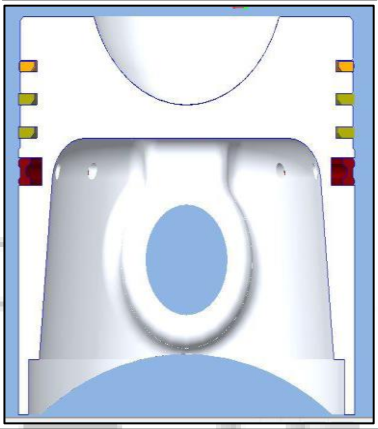
From the below analysis it shows there is possibility to get maximum power output 4.5 kW@2600 rpm and expected bsfc 224gm/kWhr. This needs some changes in the Piston, Piston Ring, Cylinder Liner, Camshaft, Injector and Fuel Pump.

Parameter	Symbol	Unit	Value	Parameter	Symbol	Unit	Value
Cylinder Volume	Vs	cc	343.3	Effective expansion ratio@EVO	ER	nos	15.432
Total Clearance Vol	V2	cc	18.5	Temperature after expansion @EVO	T5	K	916.47
Volume @BDC	V1	cc	361.9	Pressure after expansion@EVO	P5	bar	1.256
Compression Ratio	CR	nos	19.5	Temperature after expansion @BDC	T6	K	886.523
Effective Swpt vol @VC	Vs'	cc	331.4	Exhaust Stack temperature	T7	C	508.75
Effective Compression ratio	'CR'	nos	18.9	Heat Addition	Qa	kJ	2043.6
Compression Pressure	P2	bar	41.8	Heat Rejection	Qr	kw	0.123
Effective Compression Pressure	P2'	bar	39.7	Thermal Efficiency	η%s	%	0.502
Compression Temp	T2	K	731.5	Indicated power	Pihp	kW	5.925
Fuel burnt in costant volme	mfcv	kh/hy	0.1	Rated Power	P	kW	4.467
desity of air			1.111	Rated Power (Corrected)	P	kW	4.468
Temp after constant vol heat addition	T3	m	901.2	Rated Power (Corrected)	P	Hp	5.987
Air Flow		kg/hr	19.8	Engine Torque (Corrected)	T	N	16.379
Pressure at 1st Heat add	P3	Bar	46.5	BSFC@2600 (Corrected)	BSFC	gm/kWhr	224.32
Temprarure after heat addition 2nd	T4	N	2814.3	BSFC@2600	BSFC	gm/hpr	164.984
Pressure	P4	N	46.5	BMEP	BMEP	bar	6.006
Effective Specific Heat ratio	γ	kJ/kg K	1.4	Volumetric Efficency	ηv%	%	73.049
Effective Specific Heat ratio	γ	kJ/kg K	1.3	Injection Qty/st		mm3/.st	15.635
Volume after constant pres heat additi	V4	cc	22.7	Injection Qty/st		mg/st	12.821
Volume after expansion	V5	cc	350	ignition delay			0.812
Cut off Rtaio	ρ	nos	3.123	Statis timing			15.6
Pressure Ratio	pr		1.17	Bore/Stroke Ratio			1.262
				Engine Static Timing			27.522

Table 14: Engine performance simulation - Output

B. *Actual piston comparison*

Below in the images shown the old piston and new modified piston for test results.



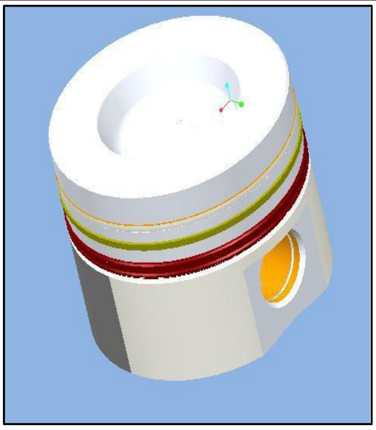


Fig. 18: New Piston

VII. CONCLUSIONS

It is found proper calculation of piston bowl volume, compression ratio, and cam valve timing, ring number, ring tension, skirt area etc. are very important to improve the specific fuel consumption and load capability.

As the engine operate on wide speed range from 1800 rpm to 2600rpm, hence swirl at high need to be properly designed otherwise will lead to thermal losses. Hence shallow and lower re-entrant shape is suitable for engine. 3no's ring is sufficient for the design as compression ratio lowed compared to earlier design hence lower pick cylinder pressure. As this type of piston does not have offset hence thrust load is lower which call for lower skirt length. Cam shaft, Injector and Pump also optimized to match the fuel injection match with Bowl design, injection jet velocity and timing.

VIII. FUTURE SCOPE

In the present research programmer, a conservative evolution of the compression ratio has been adopted. This was made for cold-start and production-feasibility requirements. Sufficiently low NOx emissions have been achieved to meet future emissions legislation, however, these were achieved with an increase in fuel consumption and extremely high levels of inlet-manifold pressures to compensate for the high EGR rates. The increased inlet-manifold pressures would result in extra parasitic losses, and hence fuelling to compensate for the turbine work and would therefore represent an additional fuel consumption increase. To minimize these drawbacks, a lower compression ratio could be implemented, allowing less retarded injection timings to be used and possibly lower levels of inlet-manifold pressures, which would both benefit fuel consumption.

The expected increase in HC and CO emissions as well as cold-start issues will need to be addressed, however, these may be easier to handle than implementing either a NOx after-treatment system or an advanced air system. The lower compression ratio could also extend the PCCI Operating region where diffusion combustion does not occur, offering confirmation of the simple NOx formation concept applicability over a greater proportion of the drive cycle. Other detailed investigations which could benefit the present work concern the effect of injection pressure, inlet-manifold temperature. Higher injection pressures could be beneficial to NOx emissions since they would increase the premix ratio and enable the injection timing to be retarded whilst maintaining an appropriate combustion phasing for fuel economy reasons. A more detailed analysis of the impact of inlet-manifold temperature on the combustion and its rate of pressure change would provide a more complete NOx formation concept. One of the main assumptions to justify load independency of the NOx formation concept was based on a stable rate of pressure change during the combustion, limited by the rate of propagation and combustion. The evidence for this is lacking at this stage and would benefit from investigations in computational fluid dynamics and ultimately in an optical engine to observe the combustion process under fully-premixed-charge combustion. The optical engine would also provide evidence of the level of mixing and potentially of the regions of NO or HC formation.

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