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Enhancement of Engine Performance and Reduction of Emissions by changing Piston Geometry

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ABSTRACT

Internal combustion engines have been a relatively inexpensive and reliable source of power for applications ranging from domestic use to large scale industrial and transportation applications for most of the twentieth century. DI Diesel engines, having the evident benefit of a higher thermal efficiency than all other engines, have served for both light- duty and heavy-duty vehicles. In DI diesel engines, swirl can increase the rate of fuel-air mixing. Swirl interaction with compression induced squish flow increases turbulence levels in the combustion bowl, promoting mixing. It is evident that the effect of geometry has a negligible effect on the airflow during the intake stroke and early part of the compression stroke. But when the piston moves towards Top Dead Centre (TDC), the bowl geometry has a significant effect on air flow thereby resulting in better atomization, better mixing and better combustion. In CI engine piston shapes on crown is flat and concave combustion chamber, with this geometry we have been running the engine. But here air fuel ratio mixture cannot mix properly. To avoid this we make piston geometry changes. The main object of this project is to investigate the performance technique to enhance the air swirl to achieve betterment in engine performance and emission in a direct injection (DI) single cylinder diesel engine. In order to achieve the swirl intensities in the cylinder, changes on the piston crown has been selected. To increase the swirl, series of experiments were conducted like 2 toroidal grooves piston, 4 toroidal grooves piston, 6 toroidal grooves piston on piston crown and compare the results with normal piston values

Key words: engine performance, toroidal grooves on piston

1. Introduction

The in-cylinder fluid motion in internal combustion engines is one of the most important factors controlling the combustion process. It governs the fuel-air mixing and burning rates in diesel engines. The fluid flow prior to combustion in internal combustion engines is generated during the induction process and developed during the compression stroke .Therefore, a better understanding of fluid motion during the induction process is critical for developing engine designs with the most desirable operating and emission characteristics.

Dr S.L.V.Prasad [1] et al. 2013 investigate the technique to enhance the air swirl to achieve betterment in engine performance and emission in a direct injection (DI) single cylinder diesel engine. In order to achieve the different swirl intensities in the cylinder, three design parameters have been selected: the cylinder head, piston crown, and inlet manifold. In order to research the intensification of swirl in the

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cylinder, series of experiments are conducted by making straight grooves in the cylinder head. In this work three different configurations of cylinder heads i.e. in the order of number of grooves 1, 3, 6 are used to intensify the swirl for better mixing of fuel and air and to enhance the performance of the engine. An attempt is made in this work with different number of channels on the cylinder head of the diesel engine. A number of channels of size 16x3x2 mm are arranged on the cylinder head depends on the locally available technology

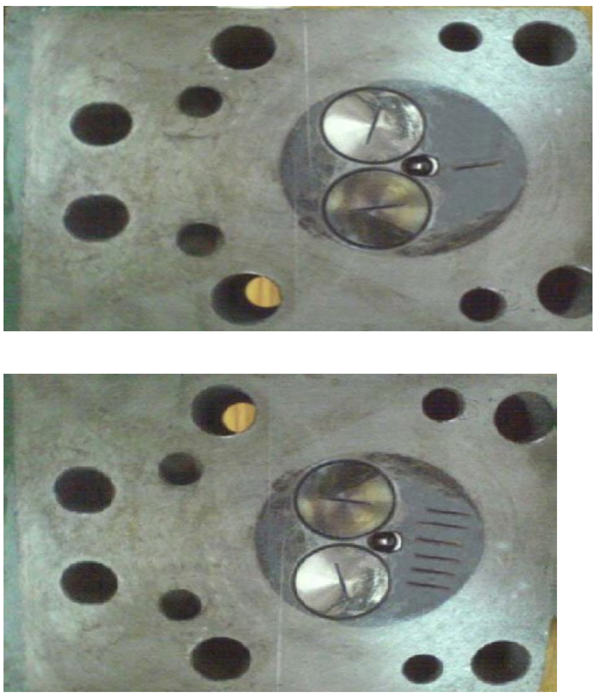

Figure.1 Cylinder head

He finally concluded that brake thermal efficiency of CH3 is increased by about 6.9% when compared to normal engine at 3/4 of the rated load.

B.V.V.S.U.Prasad [2] et al. 2010 concerns the effect of swirl induced by re-entrant piston bowl geometries on emissions in a diesel engine, and specifically focuses on a single cylinder, 7.5 kW constant-speed engine. The emission test results of two configurations of the selected engine are reported. The second configuration which has a slight lyre-entrant combustion chamber and a sac-less injector was found to yield lower emissions. In order to understand the effect of re-entrance and injector change on emissions, detailed, three- simulations of the in-cylinder processes were conducted. The effect of chamber geometry and injector change was studied using unfired and fired simulations. Simulation of closed valve part of the cycle in the two configurations revealed that average swirl and turbulence levels around TDC of compression were higher for the baseline case than for the modified geometry .Increased surface area, presence of a large central projection and insufficient re-entrance were identified as the reasons for the modified geometry yielding poor results. Combustion simulations revealed that the reduction in emissions observed during experiments is mainly due to the change in the injector rather than change in the piston bowl geometry, thus indicating scope for optimization of bowl geometry. Several piston bowl geometries with varying levels of re-entrance and different heights of central projections were simulated. A highly re-entrant piston bowl and without a central projection was found to be the best for swirl around TDC. Combustion simulations were carried out using the selected geometry and injection timings were optimized to keep NOX levels below those of the baseline case. An injection timing of8.6° CA BTDC was found to be optimum since it led to a 27% reduction in NOX emissions and 85% reduction in soot levels as compared to the baseline configuration.

Dr.G .Prasanthi [3] et al 2013 studied about influence of the air swirl in the cylinder upon the performance and emission of a single cylinder diesel direct injection engine by using diesel on volume basisis presented. The intensification of the swirl is done by cutting grooves on the crown of the piston. In this work three different configurations of piston i.e. in the order of number of grooves6,9,12 are used to intensify the swirl for better mixing of fuel and air

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and their effects on the performance and emission are recorded.

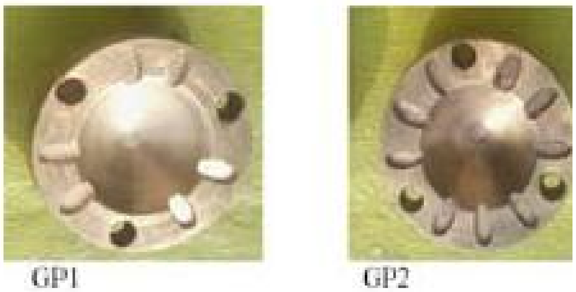

Figure.2 Piston with rhombus grooves

From the investigation, it is evident that out of all pistons configurations tested in the single cylinder D.I diesel engine ,piston with nine grooves i.e. GP2 gives better performance in all the aspects. The following conclusions are drawn based on the effect of air swirl in the cylinder at 3/4 of the rated load when compared to normal engine.

• The brake thermal efficiency is increased by about 6.9%., The improvement in brake specific fuel consumption is about8.8%., With higher turbulence in the combustion chamber, the reduction in the ignition delay is about 7.3%,The smoke emission in the engine is reduced by about5.9%,The maximum reduction in NOx emissions is about 1.8%, The maximum reduction in HC emissions is about 2.83%, The carbon monoxide emissions are found to be reduced by about 11.7%.

Vaibhav Bhatt [4] et al 2014 investigated combustion efficiency of CI Engine can be increased by creating turbulence, by designing intake system, by designing combustion chamber. A good swirl promotes fast combustion to improve the efficiency. So in this present work a study about influence of air swirl in the combustion chamber upon the performance and emission of a diesel engine is presented. In order to achieve different swirl intensity to improve combustibility of combustible mixture in the cylinder, three different configurations of piston i.e. in the order of number of grooves 4, 8 and 12 are used to intensify the swirl for the better mixing of fuel and air and their effect on the performance and emissions

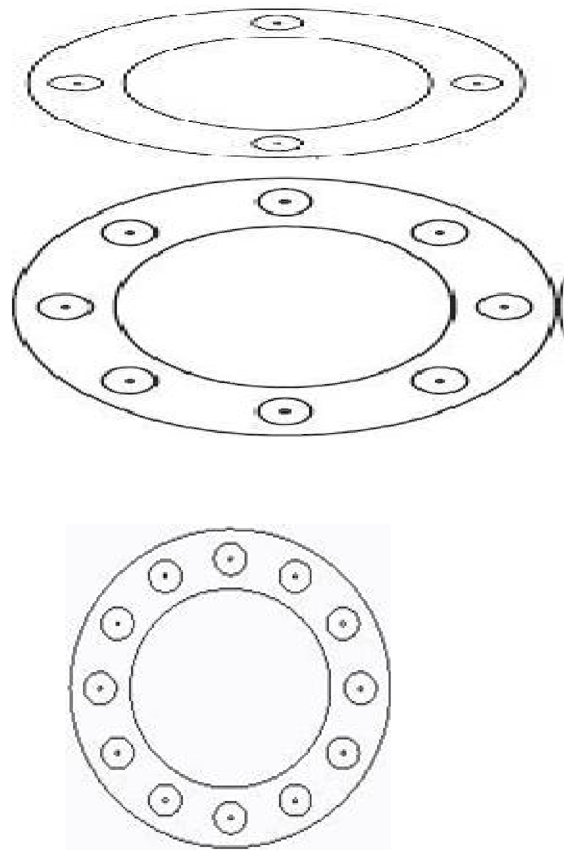

Figure.3 Piston with circular grooves

NP 4 – Normal Piston with 4 grooves

NP 8 – Normal Piston with 8 grooves

NP 12 – Normal Piston with 12 grooves

From the investigation, it is evident that out of all pistons configurations tested in the single cylinder D.I diesel engine, piston with grooves i.e. NP 12 gives better performance in all the aspects. The following conclusions are drawn based on the effect of air swirl in the cylinder at full load when compared to normal engine.

Fuel consumption for NP 12 configuration is lowest among all piston configurations.

The improvement in mechanical efficiency is about 14.8% ,The brake thermal efficiency is

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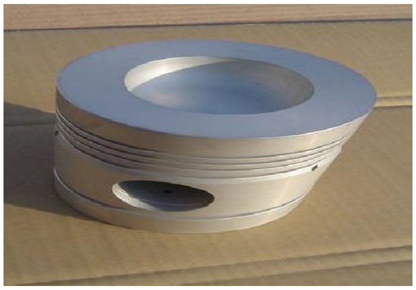
increased by about 7.2% ,The improvement in brake specific fuel consumption is about 6.9% ,The maximum reduction in exhaust gas temperature is about 10.4% ,The maximum reduction in NOx emissions is about 1.5%. The hydrocarbon emissions are found to be reduced by about 3.1%.The Carbon monoxide emissions are found to be reduced by about 10%.

Wendy Hardyono Kurniawan [5] et al have investigate the effect of piston crown inside the combustion chamber of a 4-stroke D.I. automotive engine under the motoring condition by using CFD simulation. The analyses are dedicated to investigate the outcome of the piston shape differences to the fluid flow and turbulence characteristics for air-fuel mixture preparation in the terms of swirl and tumble ratio, turbulence kinetic energy, turbulence dissipation rate and turbulence viscosity along the degree of crank angle occurred inside the engine. Two different piston bowls for certain engine speeds were considered to be compared to evaluate the parameters produced during intake and compression stroke.

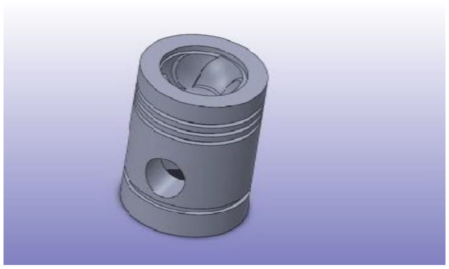
Dr. K. Hemachandra Reddy [6]et al. has worked to investigate the effect of piston bowl shape on NOx emission from DI Diesel engine by CFD prediction. For this analysis Toroidal Bowl (TB) and Central Lip Squish Bowl (CLSB) have been used. From this analysis it is concluded that NO emission with TB is higher than that of CLSB. The analysis investigates that the CLSB is better than TB.

Katsuhiko miyamoto [7]et al have worked on the improvement of combustion by means of changes in piston shapes. By using squish piston the thermal efficiency is forced up to reduce the fuel consumption and increase the combustion speed with low engine loading and that squish reduces the knock at high engine loading. During full load condition, it was confirmed that not only reduced knock but also piston shape forced up the improvements in volumetric efficiency. The benefits were evaluated experimentally and through the analysis of in-cylinder flows. From this can understand that the effects of the heights and shapes of squish areas and of the effects of the parts of a squish piston's crown outside the squish areas was gained.

I.J.Patel[8] et al 2014 experimented on the piston bowl by cutting three spiral grooves on inner surface of hemispherical bowl and slight increasing in bowl diameter. The spiral grooves increase the air capacity and slight reduce the compression ratio as well as make homogeneous mixing of air and fuel .This experiment is done on Kirloskar AV1 water cooled, natural aspirated direct injection diesel engine with pure diesel. In experiment it is observed the fuel consumption and NOx reduction by 0.1Kg per hour 8.82%. respectively.



Standard piston



Modified piston

Figure .4 Piston with spiral grooves on the inner surface

2. Methodology

The present paper attempted to evaluate the engine performance and emission characteristics of normal piston and toroidal shape of 2 grooves, 4

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grooves, and 6 grooves with diesel and compare normal piston.

3. TOROIDAL SHAPE OF THE PISTON:

In mathematics, a toroid is a doughnut-shaped object, such as an O-ring. It is a ring form of a solenoid. Its annular shape is generated by revolving a plane geometrical figure about an axis external to that figure which is parallel to the plane of the figure and does not intersect the figure. When a rectangle is rotated around an axis parallel to one of its edges, then a hollow cylinder (resembling a piece of straight pipe) is produced.

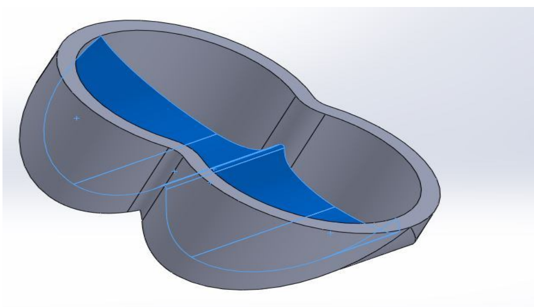

Figure .5 Toroidal shape

4.VERTICAL MILLING CENTER:

Milling is the machining process of using rotary cutters to remove material from a work piece advancing in a direction at an angle with the axis of the tool. It covers a wide variety of different operations and machines, on scales from small individual parts to large, heavy-duty gang milling operations. It is one of the most commonly used processes in industry and machine shops today for machining parts to precise sizes and shapes.

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Figure. 6 Vertical Milling Centre

5. EXPERIMENTAL SET UP:

The experimental set up consists of engine, an alternator, hydraulic load system, fuel tank, exhaust gas measuring digital device and manometer.

Fig 7. Experimental Setup of the Test Engine

Fig. 8. Hydraulic Dynamometer with manometer

7. ENGINE SPECIFICATIONS:

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The engine is single cylinder four stroke Diesel engine

Model	- HMT04
Make	- Kirloskar Oil Engine Ltd.
Ignition System	- Compression Ignition
Cylinder Type	- Vertical
Cooling	- Water cooled
Bore	- 0.0875 m
Stroke	- 0.11 m
Compression ratio	- 17.5:1
Speed	- 1500 RPM
Rated Power	-15 HP at 3000 – 5000 RPM
Piston Diameter	- 0.0874 m

8. EXPERIMENTAL PROCEDURE

Before starting the engine, the fuel injector is separated from the fuel system. it is clamped on the fuel injection pressure tested and operates the tester pump. Observe the pressure reading from the dial. At which the injector starts spraying. In order to achieve the required pressure by adjusting the screw provided at the top of the injector .This procedure is repeated for obtaining the various required pressures.

As first said, diesel alone is allowed to run the engine for about 30 min, so that it gets warmed up and steady running conditions are attained. Before starting the engine, the lubricating oil level in the engine is checked and it is also ensured that all moving and rotating parts are lubricated.

The various steps involved in the setting of the experiments are explained below

1. The Experiments were carried out after installation of the engine
2. The injection pressure is set at 220 bar for the entire test.
3. Precautions were taken, before starting the experiment.
4. Always the engine was started with no load condition
5. The engine was started at no load condition and allowed to work for at least 10 minutes to stabilize.
6. The readings such as fuel consumption, speed of the engine, cooling water flow rate, manometer reading, emission testing etc., were taken as per the observation table.
7. The load on the engine was increased by 20% of FULL Load using the engine controls and the readings were taken as shown in the tables.
8. Step 3 was repeated for different loads from no load to full load.
9. After completion of test, the load on the engine was completely relieved and then the engine was stopped.
10. The results were calculated as follows.

The above experiment is repeated for various loads on the engine .The experimental procedure is similar as foresaid. While starting the engine, the fuel tank is filled in required fuel proportions up to its capacity. The engine is allowed to run for 20 min, for steady state conditions, before load is performed.

9. EMISSION TESTING:

INDUS model PEA205 is a 5-gas analyzer meant for monitoring CO, CO_2, HC, O_2 and NO in automotive exhaust. It meets OIML Class-I specifications. CO, CO_2 and HC (Hydrocarbon residue) are measured by NDIR technology and O_2 and NO by electrochemical sensors. It is also supplied as a 4-gas analyzer which can be upgraded easily to 5-gas version by the addition of an NO sensor. It has many control features to prevent faulty measurements. A built-in dot matrix printer is provided to print out a hard copy of the results. It conforms to CMVR 115/116 and is certified by ARAI, Pune.

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Fig 9: Five Gas Analyzer

10. PISTONS BEFORE AND AFTER TESTING:

Piston images of normal iston:

Figure10 . Image of normal piston

2-groove piston before and after testing:

Figure 11. Images of 2 grooves piston

4-groove piston before and after testing:

Figure12. Image of 4-groove piston

6-groove piston before and after testing:

Figure13 . Image of 6-groove piston Dimensions of toroidal groove:

11. SIZE OF TOROIDAL GROOVE:

Diameter of toroidal groove : 10 mm

Depth of toroidal groove : 5 mm

Length of toroidal groove : 16 mm

12. FORMULAE:

The parameters that are determined at different loads are as fallows

1. Brake Power,

$$B.P=\frac{W*N*0.746}{1500}Kw..................(1)$$

Where, W=Load in kgs

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N=Engine speed in RPM

2. Total Fuel Consumption,

$$\text{T.F.C}=\frac{30\times0.826\times3600}{1000\times t}\ \text{kg/hr}\ldots\ldots\ldots\ldots(2)$$

T.F.C=Total Fuel Consumption in kg/hr

Specific gravity of diesel = 0.826

t = Time taken for 30 c.c fuel, seconds

3.Brake Specific Fuel Consumption,

$$=\frac{\text{T.F.C}}{\text{B.P}}\ \text{kg/kw-hr}\ldots\ldots\ldots\ldots(3)$$

4. Brake Thermal Efficiency

$$\eta_{bth}\ \%=\frac{\text{B.P}}{\text{T.F.C}\times\text{Cv}}\times100\%\ldots\ldots\ldots\ldots(4)$$

Where, B.P=Brake power in kW

T.F.C=Total fuel consumption in kg/hr

Cv=Calorific value of fuel=42500KJ/Kg k

5. Indicated power, I.P= B.P+F.P…………(5)

Where ,B.P=Brake power in kW

F.P=Frictional power in kW which is obtained from graph

6. Mechanical Efficiency

$$\eta_{mech}\ \%=\frac{\text{B.P}}{\text{I.P}}\times100\%\ldots\ldots\ldots\ldots(6)$$

7. Indicated Thermal Efficiency

$$\eta_{ith}\ \%=\frac{\text{I.P}}{\text{T.F.C}\times\text{Cv}}\times100\%$$

Table 1: Readings for normal piston:

S.no.	Load in kg	Speed Rpm	v_f c.c	Time sec	Exhaust gas temp
1	0	1500	30	253	110
2	1	1500	30	178	140
3	2	1500	30	140	170
4	3	1500	30	122	190
5	4	1500	30	109	220
6	5	1500	30	96	250

Table 2 Readings for 2-groove piston:

S.no.	Load in kg	Speed Rpm	v_f c.c	Time sec	Exhaust gas temp
1	0	1500	30	256	130
2	1	1500	30	158	170
3	2	1500	30	116	210
4	3	1500	30	105	225
5	4	1500	30	88	255
6	5	1500	30	75	280

Table 3: Readings for 4-groove piston:

S.no.	Load in kg	Speed Rpm	v_f c.c	Time sec	Exhaust gas temp
1	0	1500	30	247	110
2	1	1500	30	166	160
3	2	1500	30	136	180
4	3	1500	30	113	210
5	4	1500	30	92	240
6	5	1500	30	82	270

Table 4: Readings for 6-groove piston:

S.no.	Load in kg	Speed rpm	v_f c.c	Time sec	Exhaust gas temp
1	0	1500	30	194.3	120
2	1	1500	30	155	180
3	2	1500	30	126	200
4	3	1500	30	103	235
5	4	1500	30	85	270
6	5	1500	30	69.5	300

13. RESULTS& DISCUSSIONS

Table 5 : Engine performance results for normal piston:

S.no.	Load kg	T.F.C Kg/hr	B.P Kw	F.P Kw	I.P kw	B Kg
1	1	0.501	0.746	2.3	3.046	0.
2	2	0.637	1.492	2.3	3.792	0
3	3	0.731	2.238	2.3	4.538	0
4	4	0.818	2.984	2.3	5.284	0
5	5	0.929	3.73	2.3	6.03	0

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Table 6: Emission test results for normal piston:

S.no.	Load Kg	HC ppm	CO %	NO_x ppm
1	1	1567	3.591	533
2	2	1581	3.613	688
3	3	1595	3.635	885
4	4	1603	3.649	1121
5	5	1617	3.690	1352

Table 7: Engine performance results for 2-groove piston:

S.no.	Load kg	T.F.C Kg/hr	B.P kw	F.P Kw	I.P kw
1	1	0.564	0.746	1.5	2.246
2	2	0.769	1.492	1.5	2.992
3	3	0.850	2.238	1.5	3.738
4	4	1.014	2.984	1.5	4.484
5	5	1.19	3.73	1.5	5.23

Table8 : Emission test results of 2-groove piston:

S.no.	Load Kg	HC ppm	CO %	NO_x Ppm
1	1	1575	2.940	394
2	2	1576	2.930	520
3	3	1576	2.927	552
4	4	1575	2.818	425
5	5	1574	2.854	486

Table 9: Emission test results of 4-groove piston:

S.no.	Load Kg	HC ppm	CO %	NO_x ppm
1	1	1508	2.719	420
2	2	1518	2.747	485
3	3	1526	2.743	619
4	4	1534	2.758	724
5	5	1543	2.797	859

Table 10: Engine performance results for 6-groove piston:

B.P kw	F.P Kw	I.P kw	BSFC Kg/kw-hr	η_{bth} %	η_{mech} %	η_{ith} %
1	0.575	0.746	1.65	2.396	0.771	10.99
2	0.708	1.492	1.65	3.142	0.475	17.85
3	0.866	2.238	1.65	3.888	0.387	21.89
4	1.049	2.984	1.65	4.634	0.352	24.095
5	1.284	3.73	1.65	5.386	0.344	24.61

BSFC Kg/kw-hr	η_{bth} %	η_{mech} %	η_{ith} %
0.756	11.20	33.21	33.73
0.515	16.43	54.98	32.96
0.380	22.30	59.87	37.25
0.340	24.93	66.55	37.45
0.320	26.55	71.32	37.23

Table 11: Emission test results of 6-groove piston:

S.no.	Load Kg	HC ppm	CO %	NO_x Ppm
1	1	1527	2.823	424
2	2	1540	2.806	501
3	3	1546	2.820	573
4	4	1554	2.825	670
5	5	1561	3.031	776

Table 12.Comparison on Emission test results of normal, 2-groove, 4-groove, 6-groove pistons:

Piston	Load	HC	CO	NO_x
Normal	5	1617	3.690	1352
2-groove	5	1574	2.854	486
4-groove	5	1543	2.797	859
6-groove	5	1561	3.031	776

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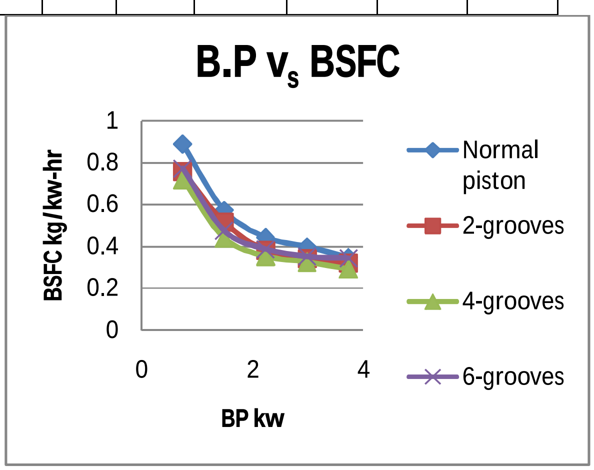
Table13: Comparison of brake thermal efficiencies with all the pistons:

S.no.	Load	B.P	η_{bth}-NP	η_{bth} -2	η_{bth} -4	$\eta_{bt\,h}$-6
1	1	0.746	9.16	11.20	11.76	10.99
2	2	1.492	14.08	16.43	19.26	17.85
3	3	2.238	18.03	22.30	24.027	21.89
4	4	2.984	20.25	24.93	26.085	24.095
5	5	3.73	23.25	26.55	29.04	24.61

Graph 1 : Brake power v_s Brake thermal efficiency

Table 14: Comparison of brake specific fuel consumption with all the pistons:

S.no.	Load	B.P	BSFC -NP	BSFC -2	BSFC -4	BSFC -6
1	1	0.746	0.888 4	0.756	0.720	0.771
2	2	1.492	0.573	0.515	0.440	0.475
3	3	2.238	0.444	0.380	0.353	0.387
4	4	2.984	0.399	0.340	0.325	0.352
5	5	3.73	0.347	0.320	0.292	0.344

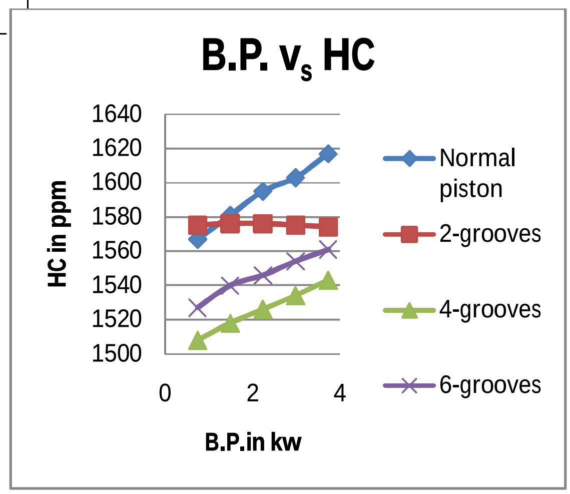


Graph 2: Brake power v_s Brake specific fuel consumption

15.. **EMISSION CHARACTERISTICS:**

Table16:Comparison of hydrocarbons with all the pistons:

S.no.	Load	B.P	HC-NP	HC-2	HC-4	HC-6
1	1	0.746	1567	1575	1508	1527
2	2	1.492	1581	1576	1518	1540
3	3	2.238	1595	1576	1526	1546
4	4	2.984	1603	1575	1534	1554
5	5	3.73	1617	1574	1543	1561



Graph 3 : Brake power v_sHydrocarbons

Table 16: Comparison of carbon monoxide with all the pistons:

S.no.	Load	B.P	CO-NP	CO-2	CO-4	CO-6
1	1	0.746	3.591	2.940	2.719	2.823
2	2	1.492	3.613	2.930	2.747	2.806
3	3	2.238	3.635	2.927	2.743	2.820
4	4	2.984	3.649	2.818	2.758	2.825
5	5	3.73	3.690	2.854	2.797	3.031

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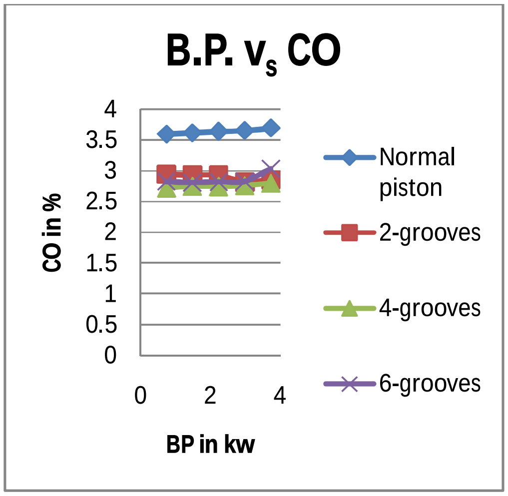
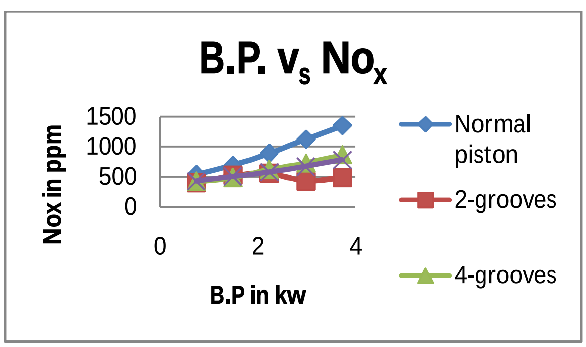


Graph 4: Brake power v_sCarbon monoxide

Table 17: Comparison ofnitrous oxide with all the pistons:

Load	B.P	NO_X-NP	NO_X-2	NO_X-4	NO_X-6
1	0.746	533	394	420	424
2	1.492	688	520	485	501
3	2.238	885	552	619	573
4	2.984	1121	425	724	670
5	3.73	1352	486	859	776



Graph 5 : Brake power v_sNitrous oxide

15. CONCLUSIONS

The geometry of the piston is modified by accommodating grooves in the piston surface to induce turbulence by means of swirl motion of charg. Based on the experimental work on a single cylinder diesel engine with swirl piston the following conclusions are drawn.

1. The brake thermal efficiency at 17.5:1 compression ratio, 220 bar pressure and advanced injection timing at full load is found to be maximum value of 23.25% for normal piston and 29.04 % for 4-groove piston. Thereby it has been a rise of 19.93 % at full load condition of the engine for 4-groove piston when compared to normal piston.
2. The brake specific fuel consumption is found to be 0.347 kg/kw-hr for normal piston and 0.292 kg/kw-hr for 4-groove piston. There has been a decrease of 0.055 kg/kw-hr, the brake specific fuel consumption obtained for 4-groove piston is less when compared to normal piston.
3. The exhaust gas temperature is found to be 250^0C for normal piston and 270^0 for 4-groove piston.There is an increase in 20^0C, and then exhaust gas temperature obtained for 4-groove piston is more.
4. The CO emissions for normal piston are 3.690 %vol and2.797%vol for 4-groove piston There has been a decrease in 0.893 % in CO emissions when compared to normal piston.
5. The HC emissions for normal piston are 1617 ppm and1543 ppm for 4-groove piston.Hence with the use of 4-groove piston there has been a considerable decrease of 74 ppm in HC emissions.
6. The NO_x emissions for normal piston are 1352 ppm and859 ppm for 4-groove piston. Hence with the use of 4-groove piston there has been a considerable decrease of 493ppm in NO_x emissions .From the above it can be concluded that with the use of swirl piston there has been an improvement in brake thermal efficiency and decrease of exhaust gas temperature and decrease of brake specific fuel consumption and have remarkable decrease in exhaust emissions of CO,HC and increase in NO_x .

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