Experimentation and Characterization of CI Engine Performance with Design Modifications in Piston: By Inducing Turbulence

K. Srinivasa Raghavan[1], K Vijaya Kumar[2]
[1]*Mechanical Engineering Department*
MCET, Affiliated to Osmania University
HYDERABAD, INDIA
srinivasaraghavan343@gmail.com

J A Ranga Babu[2], P Shailesh[1]
[2]*Mechanical Engineering Department*
NIT Warangal, Deemed University
Warangal, INDIA
vijay701414@gmail.com

***Abstract*— Transportation sector plays a critical role in the world energy consumption. As the petroleum reserves are running at depleting rate, so that it defining the country economy and also these fuels are cause for many environmental issues. It is high time to improve the performance of engine and minimize the emission rates to the Environmental Protection Agency standards and also search for an alternative fuel to reduce the dependency on fossil reserves. The present work focusing on improves the performance of Compression Ignition Engine by doing modifications in combustion chamber and also analyses the performance and emission characteristics by conducting the experiments with diesel. A stirrer is introduced on the top of the piston so as to induce turbulence in incoming charge for enhancing the vaporization rate. whirling motion is created in combustible mixture by providing rotating blades in the cavity/bowl on the reciprocating piston head. The oscillatory motion of the connecting rod will cause to rotate the blade by an angle of 60°. This arrangement induces the turbulence in combustible mixture during engine operation, there by facilitating a better combustion performance. The effects of various operating parameters like induced turbulence, injection pressure, compression ratio, injection timing and diesel mixture proportion in combustion are investigated.**

Keywords*— *CI Engine, Piston, Stirrer, Turbulence, Diesel.

I. Introduction

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. The in-cylinder fluid motion is one of the most important factor that controlling the combustion process in IC engines. It governs the fuel-air mixing and burning rates in diesel engines. The charge would flow prior to combustion and turbulence will generated during the induction process and it would develop 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 for effective functioning and emission characteristics of the combustion process.

To obtain a better combustion with lesser emissions in direct-injection diesel engines, it is necessary to achieve a good spatial distribution of the injected fuel throughout the combustion chamber. Therefore, matching the combustion chamber geometry, fuel injection and gas flows are the most crucial factors for attaining a better combustion. In DI diesel engines, swirl can increase the rate of fuel-air mixing, reducing the combustion duration and retarded injection timings. Swirl interaction and squish flow increases turbulence levels in the combustion bowl causes for promoting charge mixing. Since the flow in the combustion chamber develops from interaction of the intake flow with the in-cylinder geometry, the goal of this work is to characterize the role of combustion chamber geometry on in-cylinder flow. Somender Singh et al. [1] studied the effect of design change by forming grooves or channels or passages through the squish areas which further enhance in-cylinder turbulence followed by multi flame front combustion. This provides a faster and efficient burn, with less loss of heat, through his design to improve turbulence in combustion chambers. Yang-Liang Jeng et al. [2] experimentally investigated the quality of the tumbling motion, especially for the engine with a bowl in piston. It is observed that, a small-scale vortex will be reserved inside the bowl in piston. In their further investigation the quality of the vertical flow in the axial plane with the generation of turbulence during the compression stroke is strongly recommended. Kern Y. Kang et al. [3] evaluated the turbulence characteristics of the tumble flow in a four valve engine by Laser Doppler velocimetry (LDV) and analyzed by means of turbulence intensity, integral time and length scales, and energy spectrum. The results showed that tumble causes to increase turbulence intensity during the compression stroke and its distribution to be homogeneous. Henrik W. R. Dembinski et al. [4] studied by applying different inlet port designs and valve seat making, swirl and tumble. To measure the in-cylinder flow around TDC, particle image velocimetry software was used to evaluate combustion pictures thereby mean swirl number was calculated. This offset survives during the compression and combustion. Jeong-Eue Yun et al. [5] A Studied the combined effect of swirl and tumble flow of intake port system in cylinder head. Since both swirl and tumble flows are together induced to the cylinder, in-cylinder flow pattern becomes very complicated so that it is difficult to decide this flow as one major pattern like swirl or tumble. Their study results are to find new evaluation index for in-cylinder flow characteristics instead of current swirl or tumble coefficient using a steady flow test rig on intake port systems. Yoshihiro Suzuki [6] proposed base line modifications in piston and cylinder liners to enhance mechanical and thermal loading on vital engine parts. Hard anodizing the piston head, reinforcing the piston head with SiC-whisker. He proposed a new process for improving the surface lubrication by introducing numerous finely distributed micro-pits. S.L.V. Prasad et al. [7,8,9] studied the influence of the air swirl in cylinder on the performance and emission of a diesel engine. In order to achieve the different swirl intensities in the cylinder, three design modifications were done on cylinder head, piston crown, and inlet duct. All the modifications are aimed to improve the turbulence in the charge. This intensification of the swirl is done by cutting grooves on the crown of the piston. B.V.V.S.U. Prasad et al. [10] In-cylinder air motion was studied in a number of combustion chamber geometries which produced the highest in-cylinder swirl and Turbulence Kinetic Energy (TKE) around the top dead centre was identified.

It is evident from the literature, that the bowl 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 for creating proper turbulence thereby resulting better mixing and better combustion.

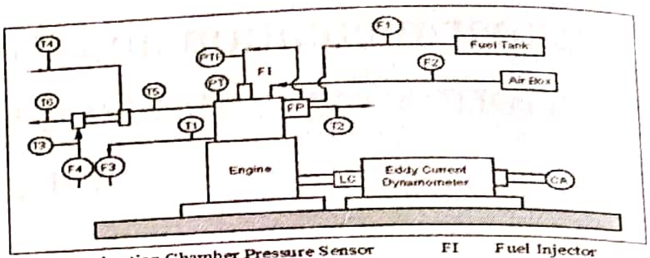
TURBULENCE

Turbulence plays a vital role in combustion phenomenon. In combustion the flame speed is very low in non-turbulent mixtures. A turbulent motion of the mixture intensifies the processes of heat transfer and mixing of the burned and unburned portions in the flame front, which practically increase in proportion to the turbulence velocity. This turbulence can be increased at the end of the compression by suitable design of combustion chamber, which involves the geometry of cylinder head and piston crown. The degree of turbulence increases directly with the piston speed. However, excessive turbulence is also undesirable. The effects of turbulence can be summarized as, turbulence accelerates chemical action so that the combustion time is reduced and hence minimizes the tendency to detonate. Turbulence increases the heat flow to the cylinder wall and in the limit excessive turbulence may extinguish the flame.

II. EXPERIMENTAL SETUP

Table 1: Engine Specifications

Sl. No.	ENGINE PARAMETERS	SPECIFICATIONS
1	Engine type	Kirloskar, 4- stroke
2	No. of Cylinders	Single cylinder
3	Rated Power	5.2 kW (7 HP)
4	Bore	87.5mm
5	Stroke	110mm
6	Swept Volume	661cc
7	Compression Ratio	17.5:1
8	Rated Speed	1500 RPM
9	Dynamometer	Eddy Current Dynamometer
10	Type of Cooling	Water cooling
11	Fuel Injection Pressure	190 bar
12	Fuel	Diesel



PT	Combustion Chamber Pressure Sensor	FI	Fuel Injector
PTF	Fuel Injection Pressure Sensor	FP	Fuel Pump
T1	Jacket Water Inlet Temperature	F1	Liquid fuel flow rate
T2	Jacket Water Outlet Temperature	F2	Air Flow Rate
T3	Inlet Water Temperature at Calorimeter	F3	Jacket water flow rate
T4	Outlet Water Temperature at Calorimeter	F4	Calorimeter water flow rate
T5	Exhaust Gas Temperature before Calorimeter	LC	Load Cell
T6	Exhaust Gas Temperature after Calorimeter	CA	Crank Angle Encoder
EGC	Exhaust Gas Calorimeter		

Fig. 1: Schematic Diagram of the Experimental Set-up

Table 1 and Fig. 1 shows the engine specifications and schematic diagram of experimental setup for determining the effects of squish and tumble on the performance of CI engine. It is a computerized single cylinder four stroke water cooled CI engine with an eddy current dynamometer. Engine is provided with Chromium-Aluminum thermocouples to measure the jacket water inlet and out let (T1 & T2), calorimeter water inlet and outlet (T3 & T4), exhaust gas inlet and outlet (T5 & T6) temperature. This engine also provided with pressure sensors, the dynamic pressure with water cooled piezo sensor, combustion gas pressure with differential pressure transducers and fuel injection pressure with differential pressure unit. Cooling water flow with calibrated Rota meter with stainless steel float. An encoder is fixed for crank angle record. The signals from all these sensors are interfaced with a computer to display P-Θ, P-V and FIP- Θ plots. The provision is also made for the measurement of volumetric fuel flow. A built-in program in the system calculates indicated power, brake power, thermal efficiency, volumetric efficiency and heat balance. The experiments are conducted at constant speed and at four different loads levels viz., 20%, 40%, 60% and 80% of full load.

A. Engine Modification

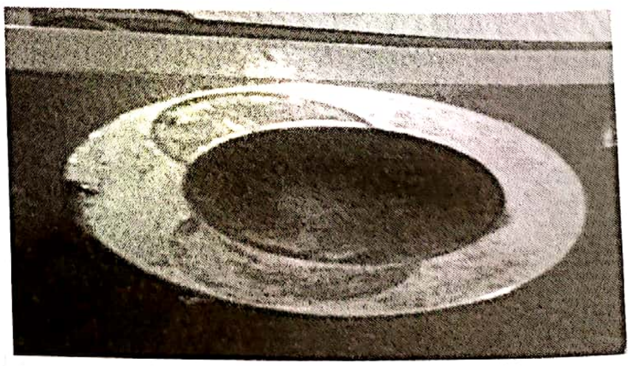


Fig. 2 Normal Piston

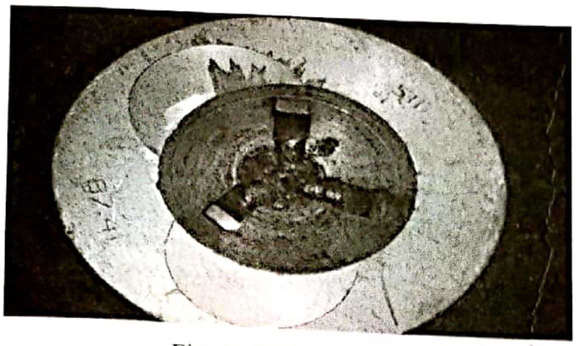


Fig. 3 Modified Piston

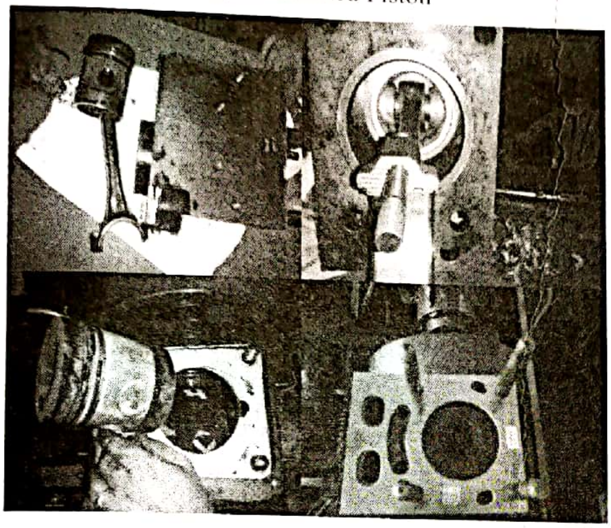


Fig. 4 Modified piston arrangement in cylinder

Fig. 2 and Fig. 3 shows the normal piston and modified piston respectively. Normal piston is having simple bowl shaped structure on the crown of it. But the modified piston (Fig. 4) is made with three blades at 120^0 to each other. Same aluminum alloy material is used in fabrication of chamber. A 2mm thick small strips are used to make the chambers.

B. METHODOLOGY

The engine was operated with compression ratios of 17.5 at constant speed of 1500 rpm. An injection pressure of 200bar, 250bar and 300bar were used. The engine is first run at steady state with pure diesel at 20%, 40%, 60% and 80% loading. At each load, performance parameters namely speed, exhaust gas temperature, brake power, peak pressure are measured. The experiments are repeated for two compression ratios, injection pressures and injection timings. With the above experimental results, the parameters such as total fuel consumption, brake specific fuel consumption, brake specific energy consumption, brake thermal efficiency are calculated and performance characteristics of the engine are determined. Emission analysis are carried out by INDUS model PEA205 as shown in Fig. 5, it is a 5-gas analyzer 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 are measured by electrochemical sensors.

Fig. 5 Five Gas Analyzer

III. RESULTS AND DISCUSSION

The experiments were conducted on a DI diesel engine for various loads with diesel.

A. Performance of Engine

The variation of brake thermal efficiency at different loads is presented and compared the performance of modified piston with normal piston. The Fig. 6 clears that the brake thermal efficiency of modified piston is more compared to normal piston at any given load. A similar trend is observed in brake thermal efficiency at all the loads considered in present work. The fuel is injected at three different rates as advanced, standard and retarded. From the above graphs it is observed that brake thermal efficiency is minimum 13.70% at CR of 17.5, injection pressure of 300bar at retard injection timing for normal piston at 20% load. And this brake thermal efficiency is maximum 35.39% at CR of 17.5, 200bar injection pressure with advanced injection rate for swirl piston at 60% load. The variation of mechanical efficiencies with respect to load for modified piston and normal piston shown in Fig. 7. It can be observed that the indicated thermal efficiency of modified piston is more compared to normal piston at given load. A similar trend in brake thermal efficiency can be observed at all the loads considered in present work. From figs. the mechanical efficiency is minimum at 17.5 CR, 300 Pressure and retard timing for normal piston at 20% load 46.61% and it is maximum at 17.5 CR, 200bar Pressure, advanced timing for swirl piston at 80% load 89.83%.so here brake thermal efficiency varies between 46.61% and 89.83%.

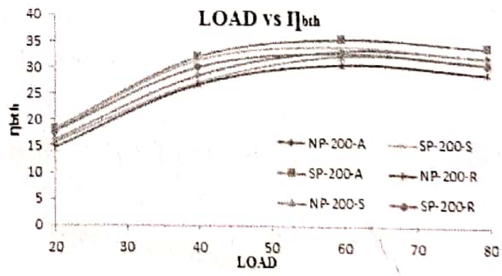


Fig. 6 (a) Load Vs Break Thermal Efficiency for 17.5 CR at 200bar

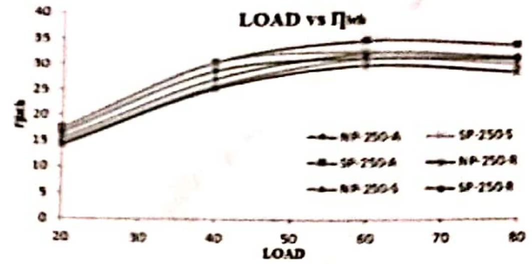


Fig. 6 (b) Load Vs Break Thermal Efficiency for 17.5 CR at 250bar

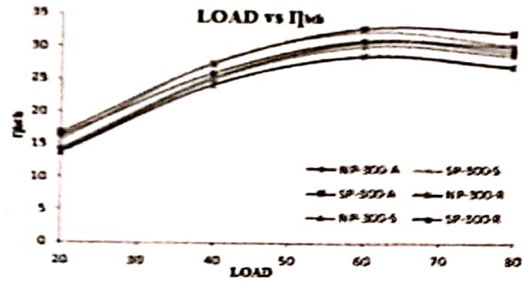


Fig. 6 (c) Load Vs Break Thermal Efficiency for 17.5 CR at 300bar

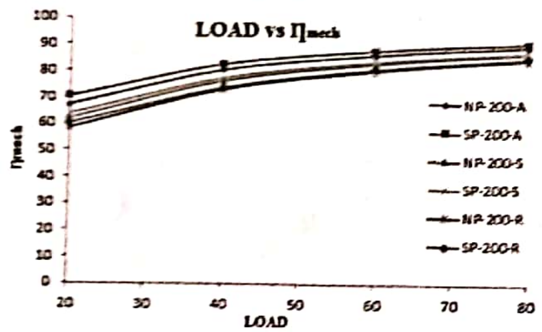


Fig. 7 (a) Load Vs Mechanical Efficiency for 17.5 CR at 200bar

Fig. 7 (b) Load Vs Mechanical Efficiency for 17.5 CR at 250bar

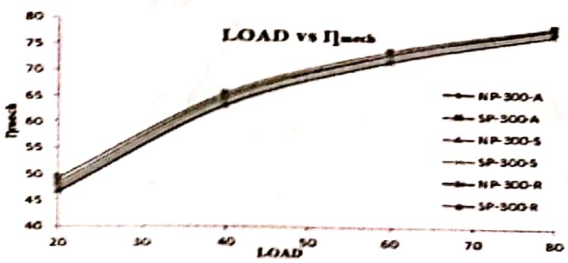


Fig. 7 (c) Load Vs Mechanical Efficiency for 17.5 CR at 300bar

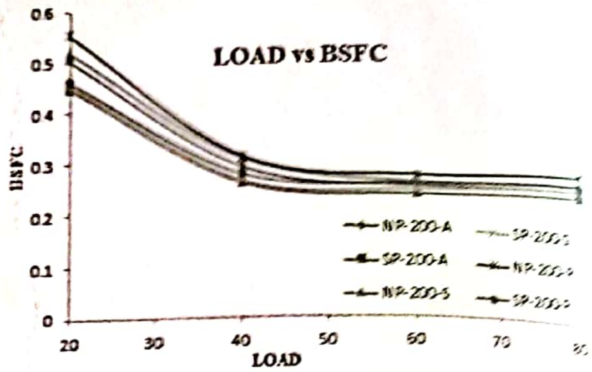


Fig. 8 (a) Load Vs Brake Specific Fuel Consumption for 17.5 CR at 200bar

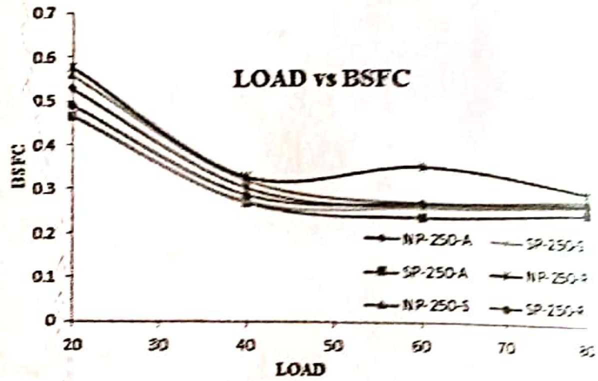


Fig. 8 (b) Load Vs Brake Specific Fuel Consumption for 17.5 CR at 250bar

The variation of brake specific fuel consumption with respect to load for modified piston and normal piston shown in Fig. 8. It can be observed that the brake specific fuel consumption of modified piston is more compared to normal piston at given load. A similar trend in brake specific fuel consumption can be observed at all the loads considered in present work. From figs. the brake specific fuel consumption is minimum at 17.5 CR, 200bar Pressure and advanced timing for normal piston at 60% load 0.231 kg/kw-hr and it is maximum at 17.5 CR, 300bar Pressure, retard timing for swirl piston at 20% load 0.597 kg/kw-hr. So here brake specific fuel consumption varies between 0.231 kg/kw-hr and 0.597 kg/kw-hr.

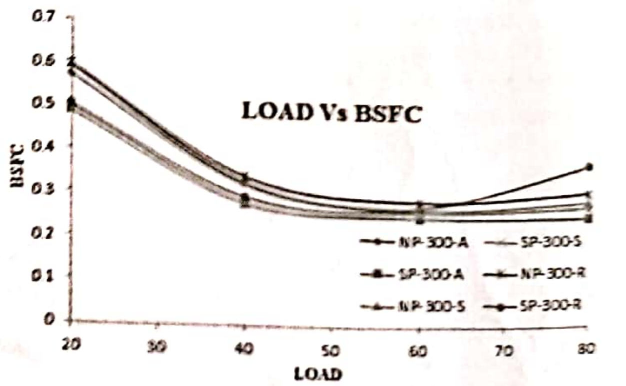


Fig. 8 (c) Load Vs Brake Specific Fuel Consumption for 17.5 CR at 300bar

B. Emissions of Engine

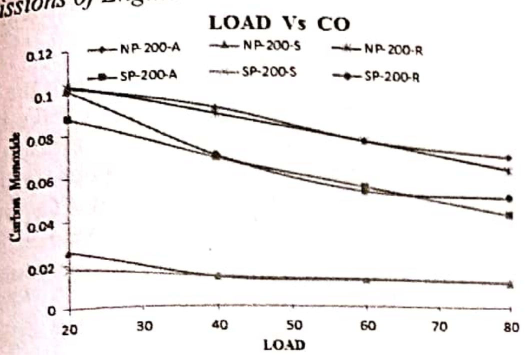


Fig. 9 (a) Load Vs Carbon Monoxide for 17.5 CR at 200bar

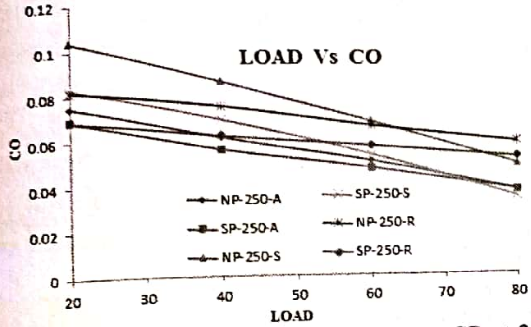


Fig. 9 (b) Load Vs Carbon Monoxide for 17.5 CR at 250bar

The variation of carbon monoxide with respect to load can be observed for normal piston and modified piston as shown in Fig. 9. The results show that Carbon monoxide is minimum at 17.5 CR, 200bar pressure, standard injection timing at 20% load 0.018%vol. and it is maximum at 17.5 CR, 250bar pressure, standard injection timing at 80% load 0.102 %vol.

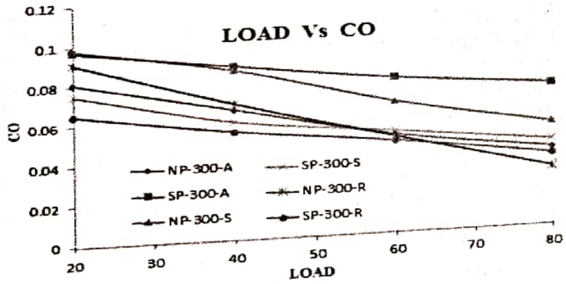


Fig. 9 (c) Load Vs Carbon Monoxide for 17.5 CR at 300bar

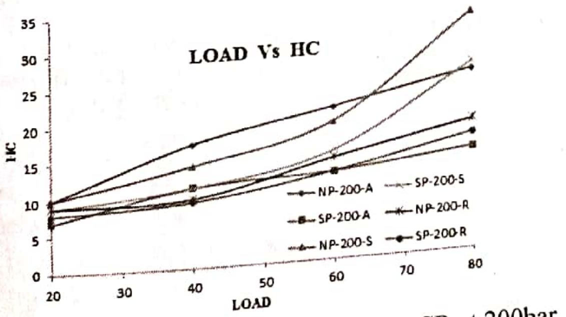


Fig. 10 (a) Load Vs Hydro Carbons for 17.5 CR at 200bar

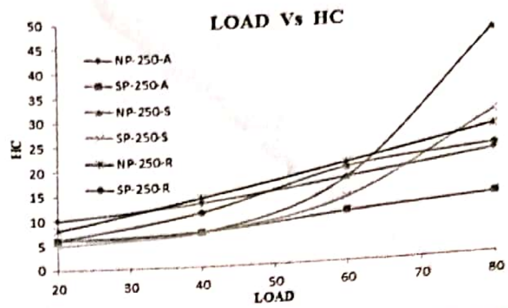


Fig. 10 (b) Load Vs Hydro Carbons for 17.5 CR at 250bar

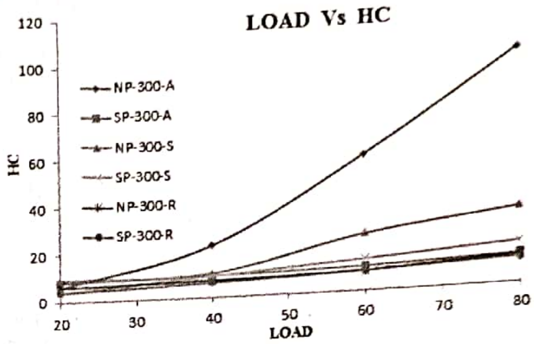


Fig. 10 (c) Load Vs Hydro Carbons for 17.5 CR at 300bar

The variation of hydrocarbons with respect to load for normal piston with modification is depicted in Fig. 10. From the results, it can be noticed that the un burnt hydrocarbon is minimum at 17.5 CR, 300bar Pressure, retard timing for swirl piston at 20% load 4ppm and it is maximum at 17.5 CR, 300bar Pressure, advanced timing for normal piston at 80% load 100ppm.

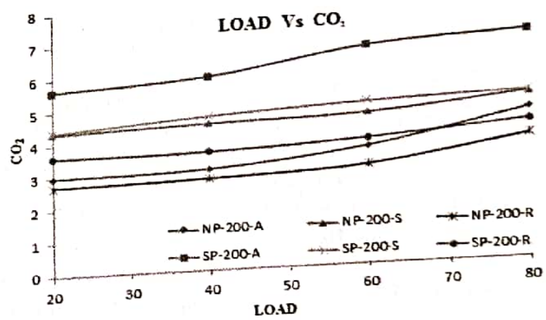


Fig. 11 (a) Load Vs Carbon Dioxide for 17.5 CR at 200bar

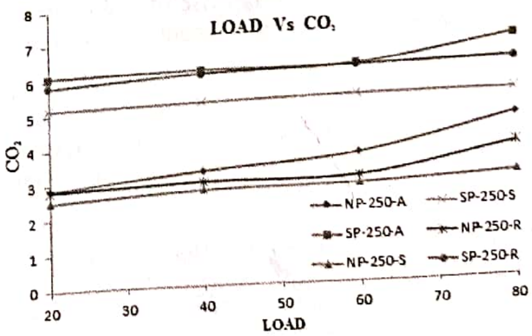


Fig. 11 (b) Load Vs Carbon Dioxide for 17.5 CR at 250bar

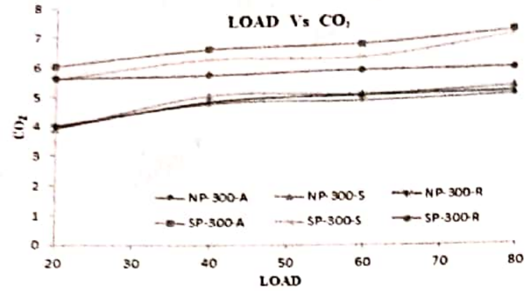


Fig. 11 (c) Load Vs Carbon Dioxide for 17.5 CR at 300bar

The variation of carbon dioxide with respect to load can be observed for normal piston and modified piston is depicted in Fig. 11. The results show that Carbon dioxide is minimum at 17.5 CR, 250bar pressure, standard injection timing for normal piston at 20% load 2.53%vol. and it is maximum at 17.5 CR, 200bar pressure, standard injection timing for swirl piston at 80% load 6.92 %vol.

CONCLUSSIONS

Fuel saving will protect the environment and related concern issues. It is necessary to do necessary modification in engine so as to minimize the exhaust emissions. In this context, the geometry of the piston is modified by accommodating rotating blades on the piston crown so as to induce turbulence by means of swirl motion of charge. From the experimental results for a diesel engine with swirl piston:

- The brake thermal efficiency at 17.5 compression ratio, 200 bar pressure and advanced injection timing at 60 % load is found to be maximum value of 35.39 % for swirl piston and it is 33.08 % for normal piston which is an approximate rise of 2-3 % at every load and injection pressures.
- The mechanical efficiency at 17.5 compression ratio, 200 bar pressure and advanced injection timing at 80 % load is found to be maximum value of 89.83% for swirl piston and it is 88.49% for normal piston. Hence there has been a rise of approximately 1-4 % at every condition of the engine for swirl piston. The volumetric efficiency at 17.5 compression ratio, 200 bar pressure and advanced injection timing at 80 % load is found to be maximum value of 82.34% for swirl piston and it is 81.49% for normal piston. Hence there has been a rise of approximately 1-2 % at every condition of the engine for swirl piston
- The brake specific fuel consumption is found to be 0.629 kg/kw-hr for normal piston and 0.543 kg/kw-hr for swirl piston. There has been a decrease of 0.086 kg/kw-hr, the brake specific fuel consumption obtained for swirl piston is less.
- The CO emissions for normal piston is 0.091 % vol and 0.065 % vol. Hence with the use of swirl piston there has been a remarkable decrease of 4% to 20% in CO emissions.
- The HC emission for normal piston is 6 ppm and 4 ppm for swirl piston. Hence with the use of swirl piston there has been a considerable decrease of 2 ppm in HC emissions.

From the above it can be concluded that with the use of swirl piston there has been an improvement in brake thermal efficiency, mechanical efficiency, volumetric efficiency and decrease of brake specific fuel consumption and have remarkable decrease in exhaust emissions of CO, HC, and increase in CO_2 emissions. Also the results obtained for 200 bar injection pressure are better 250bar injection pressure, 300 bar injection pressure.

References

[1] Somender Singh, "Design to Improve Turbulence in Combustion Chambers by Creating a Vortex" pure energy systems, 2005-10-13-9600187, 2005.

[2] Yang-Liang Jeng, Rong-Che Chen and Chao-He Chang, "Studies of Tumbling Motion Generated During Intake in a Bowl-In-Piston Engine", journal of marine science and technology, Vol. 7, No. 1, pp. 52-64, 1999.

[3] Kern Y. Kang, Je H. Baek, "Turbulence characteristics of tumble flow in a four-valve engine", experimental thermal and fluid science 18 (1998) 231-243.

[4] Henrik W. R. Dembinski Hans-Erik Angstrom, "Optical Study of Swirl during Combustion in a CI Engine with Different Injection Pressures and Swirl Ratios Compared with Calculations," SAE Technical Paper 2012-01-0682, 2012.

[5] Jeong-Eue yun, Jae-Joon Lee, "A Study on Combine Effects Between Swirl and Tumble Flow of Intake Port System in Cylinder Head", Seoul 2000 FISITA World Automotive Congress, F2000A094, June 12-15, Korea, 2000.

[6] Yoshihiro Suzuki, "Surface modifications of pistons and cylinder liners", journal of materials engineering, Vol. 10, Issue 1, PP 61-67, December 1988.

[7] S.L.V. Prasad, Prof. V. Pandurangadu, V.V. Pratibha Bharathi, V. V. Naga Deepthi, "Experimental Study of the Effect of in Cylinder Air Swirl on Diesel Engine Performance", International Journal of Engineering Science and Technology (IJEST), ISSN: 0975-5462, Vol. 3, No. 2, pp. 1571-1575, Feb 2011.

[8] Dr. S.L.V. Prasad, Prof. V. Pandurangadu, Dr. P. Manoj Kumar, Dr G. Naga Malleshwara Rao, "Reduction of Emissions by Enhancing Air Swirl in a Diesel Engine with Grooved Cylinder Head" International Journal of Innovative Research in Science, Engineering and Technology, ISSN: 2319-8753, Vol. 2, Issue 12, December 2013.

[9] Dr. S.L.V. Prasad, Prof. V. Pandurangadu, Dr. P. Manoj Kumar, Dr G. Naga Malleshwara Rao, "Enhancement of Air Swirl in a Diesel Engine with Grooved Cylinder Head" International Journal of Innovative Research in Science, Engineering and Technology, ISSN: 2319-8753, Vol. 2, Issue 8, August 2013.

[10] B.V.V.S.U. Prasad, C.S. Sharma, T.N.C. Anand, R.V. Ravikrishna, "High swirl-inducing piston bowls in small diesel engines for emission reduction", Applied Energy 88, 2355–2367, 2011.