EXPERIMENTAL INVESTIGATION OF A SINGLE CYLINDER 4- STROKE DI DIESEL ENGINE BY SWIRL INDUCTION WITH TWO DIFFERENT CONFIGURATION PISTONS

B.MADHUBABU[1], Prof. K. GOVINDARAJULU[2] and Dr.S.L.V.PRASAD[3]

[1]Lecturer in Mechanical Engineering, JNTUACEA, Ananthapuram.

[2]Principal&Professor, Department of Mechanical Engineering,JNTUA College of Engineering, Pulivendula.

[3]Professor, Department of Mechanical Engineering, SVIT, Ananthapuram.

ABSTRACT

The in-cylinder air 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. In this present work the experimental investigation of air swirl in the cylinder upon the performance and emission of a single cylinder diesel direct injection is presented. This intensification of the swirl is done by the cutting grooves on the crown of the piston, by two different configurations of RGP 3, RGP 6and normal pistons are investigatating performance and emission characteristics.Experiments are carried out on a diesel engine using Modified different configuration piston which is a four stroke single cylinder air cooled and constant speed engine. Performance parameters such as brake power, specific fuel consumption and Thermal efficiency are calculated based on experimental analysis of the engine. Emissions such as carbon monoxide, carbon dioxide and unburnt hydrocarbons are measured.

1.INTRODUCTION

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. 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 [1,2]. 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 [3]. 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 entire space [4]. This requires matching of the fuel sprays with combustion chamber geometry to effectively make use of the gas flows. In other words, matching the combustion chamber geometry, fuel injection and gas flows is the most crucial factor for attaining a better combustion [5]. In DI diesel engines, swirl can increase the rate of fuel-air mixing [6], reducing the combustion duration for re-entrant chambers at retarded injection timings. Swirl interaction [7] with compression induced squish flow increases turbulence levels in the combustion bowl, promoting 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, thus the fuel-air mixing, combustion and pollutant formation processes. 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. The re-entrant chamber without central projection and with sharp edges provides higher swirl number than all other chambers [8].

2.INFLUENCE OF AIR MOTION IN COMBUSTION CHAMBER

To enhance the efficiency of an engine it is important to optimize thermal efficiency, which is obtained at the highest possible compression ratio. However, if the compression ratio is too high, there is a chance to have knock, which should be avoided at all cost. A solution for this problem is to promote rapid combustion, to reduce the time available for the self-ignition to occur [9]. To promote rapid combustion, sufficient large-scale turbulence (kinetic energy) is needed at the end of the compression stroke because it will result in a better mixing process of air and fuel and it will also enhance flame development. However, too much turbulence leads to excessive heat transfer from the gases to the cylinder walls, and may create problems of flame propagation [10] [11] [12]. The key to efficient combustion is to have enough swirl in the combustion chamber prior to ignition. In order to provide complete combustion at a constant rate, there is common design objective of bringing sufficient air in contact with the injected fuel particles. For this purpose, the piston crown and the cylinder head are shaped to induce a swirling motion to air, while during compression piston is moving towards TDC. The production of turbulence i.e. swirl by different means, however, is considered necessary for better fuel-air mixing. The complexities of production and the higher costs of these methods of creating turbulence are the limiting factors in their wider use. An increase in air swirl level is noted to increase the air mass of all zones. Thus at the moment when the mixture first ignites in one zone, all other zones approaching their self–ignition temperature contain more air. Increased swirl results in an increase in the initial combustion rate and hence a higher rate of pressure rise is expected [13]. The Swirl can be generated in the diesel engine by modifying three parameters in the engine, they are the cylinder head, the piston i.e. modification of combustion chamber and the inlet manifold [14]. Lin.et.al [15] has invented a multi impingement wall head is located at the center of the cylinder head to enhance the swirl and squish. Somendersingh [16] has identified a method to improve turbulence in combustion chamber by making grooves on the cylinder head, to reduce the heat losses; the burn time needs to be as quick as possible. According to Al-Rousan[17] swirl is generated in the inlet manifold by inserting a loop inside the intake manifolds to increase the swirling in the air during induction. Rasul and Glasgow [18] prepared a convergent-divergent induction nozzle and is tested in order to increase the airflow into the engine, which may increase the overall performance. S.L.V.PRASAD, et al**.** [19 20 21]experimentally investigation on influence of the air swirl in the cylinder upon the performance and emission of a single cylinder diesel direct injection engine is presented. In order to achieve the different swirl intensities in the cylinder, three design parameters have been changed the cylinder head, piston crown, and inlet duct. In this way, the piston crown is modified i.e. alteration of combustion chamber to enhance the turbulence in the cylinder. This intensification of the swirl is done by cutting grooves on the crown of the piston. Performed experimentally different configurations of piston i.e. in the order grooves intensify the swirl for better mixing of fuel and air and their effects on the performance and emissions.

3.EXPERIMENTAL SETUP & PROCEDURE

The experimental set up consists of engine, an alternator, top load system, fuel tank along with immersion heater, exhaust gas measuring digital device and manometer.The engine which is supplied by M/s Anil Company. The engine is single cylinder vertical type four stroke, Water-coo1ed, compression ignition engine. The engine is self governed type whose specifications are given in Appendix 1.is used in the present work.The above engine is one of the extensively used engines in industrial sector in India. This engine can with stand the peak pressures encountered because of its original high compression ratio. Further, the necessary modifications on the cylinder head and piston crown can be easily carried out in this type of engine. Hence this engine is selected for the present project work. As first said, allowed to run the engine for about 20 min with normal piston, so that it gets warmed up and steady running conditions are attained. The experiments were conducted in diesel engine with grooved pistons of 3and 6 to known the performance and emissions.

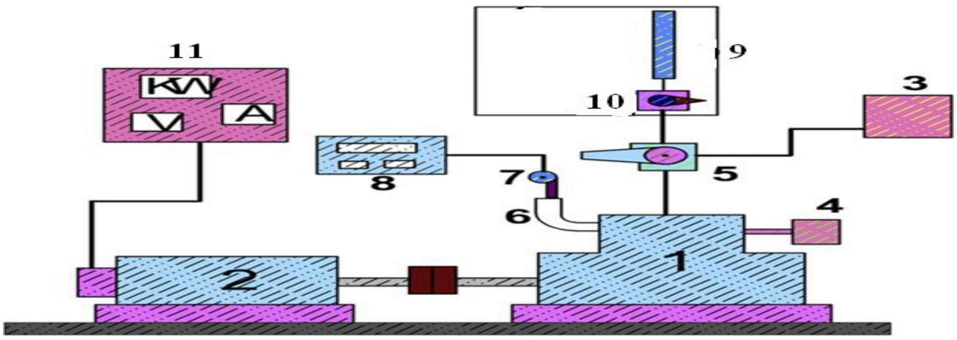

Fig 1 Experimental Setup of the Test Engine

The test is carried on the ANIL Engine for the following

Fig. 2 Different types of configurations of piston crowns

- **RGP3Rhombus 3 grooves with piston**
- **RGP 6 Rhombus 6 grooves with piston**

TABLE: 1 Specifications of Diesel engine used for Experimentation

Item	Specifications
Engine power	**3.75 kW**
Cylinder bore	**80 mm**
Stroke length	**110 mm**
Engine speed	**1500 rpm**
Compression ratio	**16.5:1**
Swept volume	**553**

4.RESULTS AND DISCUSSIONS

Experiments are carried out on 4- stroke single cylinder ANIL diesel engine using existing normal piston and two different configurations of grooved pistons which are generally known as RGP 3 and RGP 6 respectively. The performance of the engine is evaluated in terms of brake specific fuel consumption, brake thermal efficiency, exhaust gas temperature and volumetric efficiency. The emission characteristics of the engine are studied in terms, concentration of HC, CO and CO_2. The results obtained by the piston with grooves are compared with that of the normal piston. Due to experimental constraints, the present investigation has restricted the experimentation up to 2000 W only and could not conduct up to peak load of 3750 W.

4.1 LOAD VsBRAKE SPECIFIC FUEL CONSUMPTION

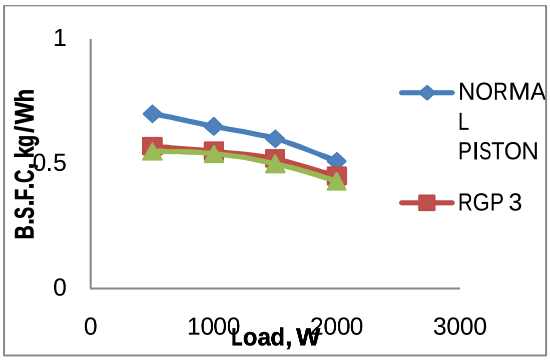

Fig.2 LoadVs B.S.F.C

The variations in the break specific fuel consumption (BSFC) with load is presented in the fig 2 for the normal piston. It is found that the BSFC falls with increasing load up to 2000W. Due to experimental constraints, the present investigation has restricted the experimentation up to 2000 W only and could not conduct up to peak load of 3750 W. Probably the BSFC may increase with load after 2000 W. From fig 2 it can be observed that, BSFC is maximum for normal piston, minimum for RGP 6, and in between these two for normal piston and RGP3at a given load. A similar trend in BSFC can be observed at all the loads considered in present work. The BSFC values of 0.51kg/kWh for normal piston, 0.45kg/kWh for RGP 3, 0.43kg/kWh for RGP 6 at a load of 2000 W is obtained. It is also observed that the RGP 6 has the lowest fuel consumption which is 15.68% lower than that of normal piston, because of the complete combustion of the charge in the combustion chamber by liberating maximum energy, due to the inducement of enhanced air swirl in the combustion chamber.

4.2 LOAD Vs BRAKE THERMAL EFFICIENCY

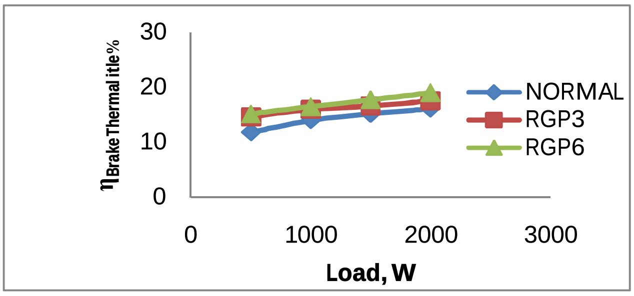

Fig 3 Load Vs Brake Thermal Efficiency

The variation of brake thermal efficiency with respect to load for piston with grooves and normal piston shown in Fig.3. Due to experimental constraints, the present investigation has restricted the representation up to 2000 W only and could not conduct up to peak load of 3750 W. Probably the brake thermal efficiency may increase with load after 2000 W. From fig 3. It can be observed of brake thermal efficiency is minimum for normal piston and maximum for RGP 3 and RGP 6 at a given load. A similar trend in brake thermal efficiency can be observed at all the loads considered in present work. The brake thermal efficiency values are of 16.13% for normal piston and 18.55% for RPG 3 at a load of 2000 W. It is also observed that RGP 6 is a gain which is 19.05% higher than that of normal piston, due to the enhanced air swirl in the combustion chamber which resulted in better mixing of air and fuel (A/F) as well as complete combustion of the charge in combustion chamber.

4.3LOADVsVOLUMETRIC EFFICIENCY

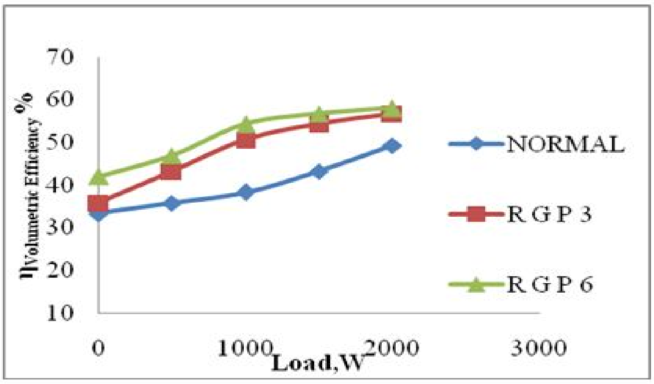

Fig 4 Load Vs Volumetric Efficiency

The variation of volumetric efficiency for piston with grooves and normal piston shown in Fig.4. Due to experimental constraints, the present investigation has restricted the experimentation up to 2000 W only and could not conduct up to peak load of 3750 W. It is observed that, volumetric efficiency is minimum for normal piston and maximum for piston with grooves RGP 3 and RGP 6 at a given load. A similar in volumetric efficiency can be observed at all the loads considered in present work. The volumetric efficiency values are 49.32% for normal piston, 56.71% for RGP 3 and 57.95% for RGP 6 at load of 2000 W. It is also observed that the RGP 6 is a gain which is 17.49% higher than that of normal piston.

4.4LOADVsEXHAUST GAS TEMPERATURE:

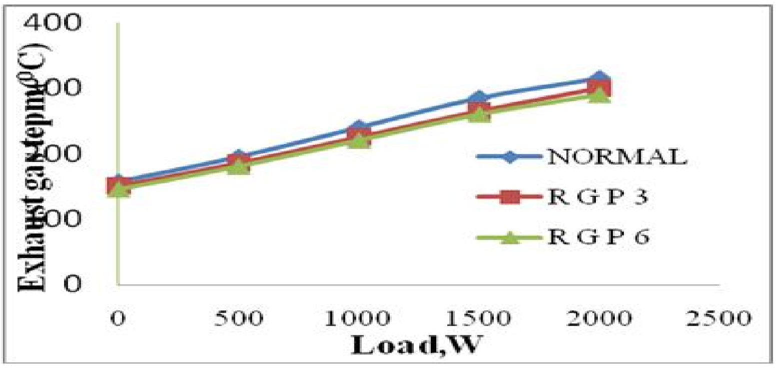

Fig 5 Load Vs Exhaust Gas Temperature

The variation of exhaust gas temperature presented in normal piston and piston with grooves are shown in fig 5. Due to experimental constraints, the present investigation has restricted the experimentation up to 2000 W only and could not conduct up to peak load of 3750 W. The exhaust gas temperature of piston RGP 6 is lower when compared to remaining piston grooves (RGP 3) and normal piston. The exhaust gas temperature values are 315°C for normal piston, 300°C for RGP 3 and 290°C for RGP 6 at a load of 2000 W. It is observed that the RGP 6 has the lowest exhaust gas temperature which is 7.93%lower than that of normal piston.

4.5LOADVsHC EMISSION:

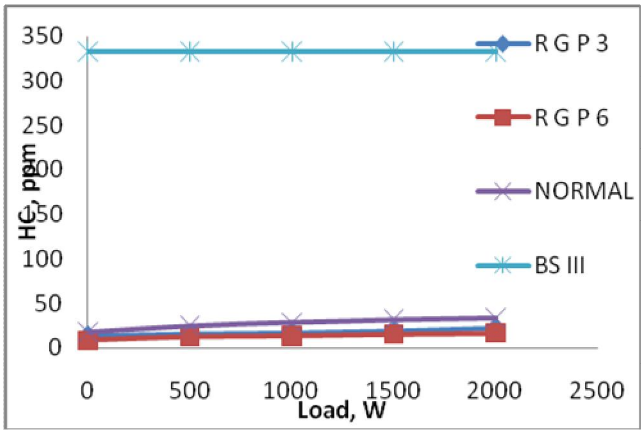

Fig. 6 Load Vs Hydro carbons

The variations of hydrocarbon emission in the exhaust are shown in fig 6. Unburnt hydrocarbon emission is because of incomplete combustion. It is apparent that the hydrocarbon emission is decreasing with increase in the turbulence, which results in complete combustion at the load 2000 W. Maximum reduction of hydrocarbon emissions levels is about 50.00%, 35.29% for RGP 6, RGP 3 compared with the normal piston. It is clear that the emissions concentrations for the piston with grooves are lower than Bharath Stage III norms.

4.6 LOADVsCO EMISSION:

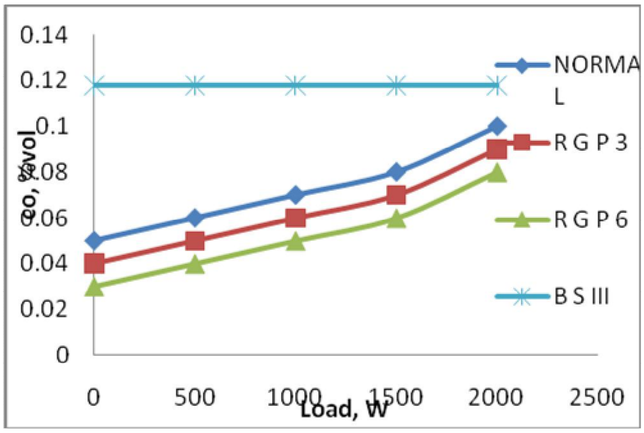

Fig. 7 Load Vs Carbon monoxide

From Fig7. The variations of carbon monoxide with respect to load are observed for normal piston and piston with grooves. The results show that CO emission of the piston with grooves is slightly lower than with the normal piston. Carbon monoxide from the exhaust gas for the normal piston is 0.10% by vol., and for RGP 3and RGP 6 are 0.09% by vol., 0.08% by vol., respectively. It is also observed that the RGP 6 reduction in CO level is about 20.00% compared with the normal piston. It is clear that the emissions concentrations are lower than Bharath Stage III up to a load of 2000 W.

5.CONCLUSIONS

The following conclusions are drawn based on the present investigation.

- The maximum increase in brake thermal efficiency for RGP 6, RGP 3 compared to normal piston was found to be 18.10%, 8.80% respectively.
- The reduction in the brake specific fuel consumption for RGP 3, RGP6 compared to normal piston was found to be 11.76%, 15.68% respectively.
- The maximum increase in Volumetric efficiency for RGP 6, RGP 3 compared to normal piston was found to be 17.49%, 14.98% respectively.

- The exhaust gas temperatures are minimized for RGP 6, RGP 3 as compared to normal piston was found to be 7.93%, 4.76% respectively.
- The carbon monoxide emissions for RGP 6, RGP 3 are found to be reduced by 20%, 10% respectively.

From the above conclusions, the RGP 6 piston configuration can be suggested on diesel engine compared with the other piston configurations.

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