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Research Article

INFLUENCE OF IN CYLINDER AIR SWIRL ON DIESEL ENGINE PERFORMANCE AND EMISSION

***V.V. Prathibha Bharathi[1] and G. Prasanthi[2]**
[1] *Department of Mechanical Engineering, JNTUCEA, JNT University, Anantapur, A.P., India*
[2] *Department of Mechanical Engineering, JNTUCEA, JNT University, Anantapur, A.P., India*
**Author for Correspondence*

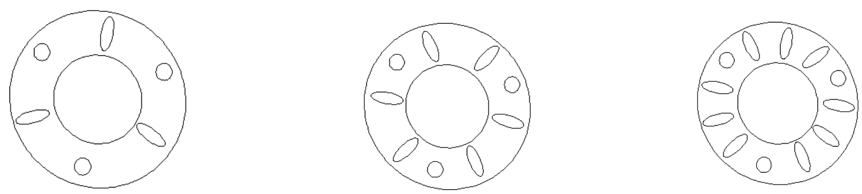
ABSTRACT
In this present work a study about influence of the air swirl in the cylinder upon the performance and emission of a single cylinder diesel direct injection engine is presented. In order to achieve the different swirl intensities in the cylinder, three design parameters have been changed: the 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. In this work three different configurations of piston i.e. in the order of number of grooves 3, 6, 9 were used to intensify the swirl for better mixing of fuel and air and their effects on the performance and emission are recorded

Key Words: *Diesel Engine, Air Swirl, Cylinder, Efficiency, Emissions*

INTRODUCTION
Swirl, considered as a two-dimensional solid body rotation, persists through the compression and combustion processes. Many researchers have demonstrated that the decay of swirl in an engine cylinder during the compression process is relatively small so that the overall angular momentum of the swirl vortex is almost conserved (Dyer, 1979; Arcoumanis et al., 1981; Hamamoto et al., 1985; Hall and Bracco, 1987; Heywood, 1988). Since the flow pattern is strain-free, swirl is expected to have negligible effect on turbulence enhancement in the bulk of the flow. However, the turbulence generated in the wall boundary can be transported throughout the bulk of the flow by diffusion and swirl driven secondary flow (Hill and Zhang, 1994). Protruding objects, such as the spark plug and valve head, also generated turbulence due to surface shear stress and vortex shedding with global swirl motion (Stephenson et al., 1996). Thus, nearly solid body rotations could result in considerable enhancement of turbulence throughout the combustion chamber, particularly near TDC.A number of engine results illustrated that turbulence was enhanced at TDC of combustion and tended to become homogeneous and isotropic for swirl motion (Liou and Santavicca, 1983; Ikegami et al., 1985; Saxena and Rask, 1987; Heywood, 1988; Li et al., 2001). For example, Liou and Santavicca (1983) found that turbulence was nearly homogeneous and isotropic near TDC in their engine experiments. They also showed that turbulence intensity near TDC at a given speed was 25-50% greater with swirl than without and then declined continuously with crank angle.
Swirl flow has been used in many different kinds of internal combustion engine because of their effects in increasing efficiency, reducing of noise and other emission pollutants, and improving combustion instability (Gupta, A.K.,et al, 1989). Heywood gives an excellent review of pollutant formation and control in SI engine (Heywood, J.B., 1976). He presented, in his paper, a graph for the effect of equivalence ratio on emission pollutants that show the complexities of emission control. Swirl motion may used before or inside the combustion chamber and, also, before or after combustion In Switzerland, May M., (1977) patented a high compression, high turbulence chamber.
The idea of generating swirl motion in the cylinder to enhance turbulence dates back to 1960s. Different mechanisms have been employed in engines to produce rotating flow during air induction process. Once created, the rotating gas motion can be intensified in the engine, depending on the piston and cylinder head geometry (Hill and Zhang, 1994).
According to payri(1990) the Swirl can be intensified in the diesel engine by modifying Three design parameters in the engine: the cylinder head, the piston (i.e modification of combustion chamber)and the inlet manifold.

MATERIALS AND METHODS

In the present work the effects of air swirl in combustion chamber are experimentally studied on performance of single cylinder light duty direct injection diesel engine. The experiments were conducted on a single cylinder Kirloskar make direct injection four stroke cycle diesel engine. The general specifications of the engine are given in Table-1. Water cooled eddy current dynamometer was used for the tests. The engine is equipped with electro-magnetic pick up , piezo-type cylinder pressure sensor, thermocouples to measure the temperature of water, air and gas, rotameter to measure the water flow rate and manometer to measure air flow and fuel flow rates, The smoke density is measured with a Bosch Smoke meter. To intensify the air swirl numbers of elliptical grooves were made on the piston crown .The different types of piston configurations which were tested in diesel engine are shown in fig. 1



GP 3 GP 6 GP 9

Figure 1: Different types of configurations of piston crowns

N P -	Normal Piston;	GP 3 -	Piston with 3 grooves
GP 6 -	Piston with 6 grooves;	GP 9 -	Piston with 9 grooves

Specifications of Diesel Engine Used for Experimentation:

Item	Specification
Engine power	3.68 kW
Cylinder bore	80 mm
Stroke length	110 mm
Engine speed	1500 rpm
Compression ratio	16.5:1
Swept volume	553 cc

RESULTS AND DISCUSSION

Brake Thermal Efficiency

The brake thermal efficiency with brake power for different configurations are compared with the normal engine configuration and is shown in Fig. 2. The brake thermal efficiency for normal piston at full load is 29.8%. It can be observed that the engine with G P 9 and G P 6 give thermal efficiencies of 31.8 % and 30.8%, respectively, at full load. It is observed that there is a gain of 6 % with G P 9 compared to normal engine. The thermal efficiencies of G P 6 and G P 3 are lower compared to that G P 9. From Fig.1, it was inferred that the brake thermal efficiencies were increasing with an increase in brake power for configurations that were under consideration. These configurations were found to offer better thermal efficiencies than the normal engine. This might be due to the enhanced mixing rate in the case of G P 9 carried by turbulence in the combustion chamber.

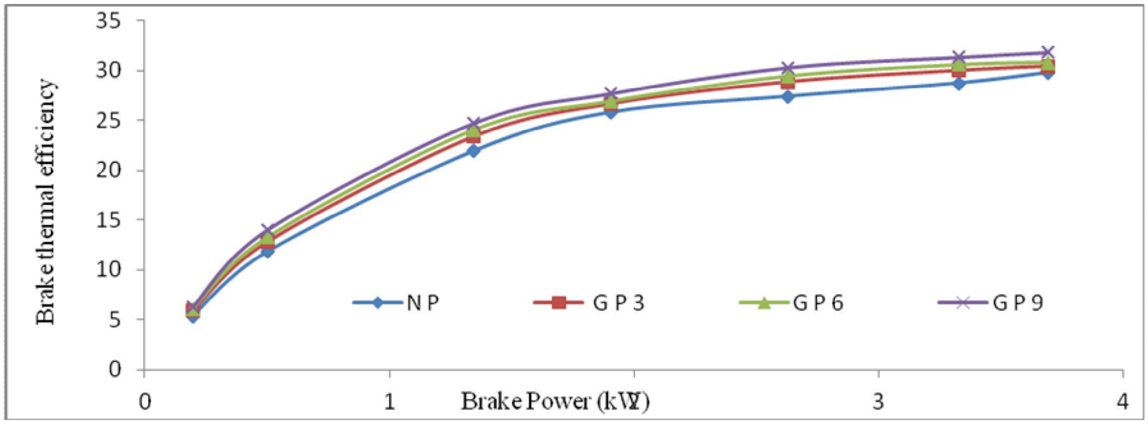


Figure 2: Comparison of Brake Thermal Efficiency with Different configurations of Piston

Exhaust Gas Temperature

Figure 3 shows the comparison of exhaust gas temperature with brake power. The exhaust gas temperatures are lower for GP 9 and GP 6 compared to that of normal engine. The exhaust gas temperature for GP 9 varies from 120°C at no load to 380°C at full load. For G P 6, the exhaust gas temperature varies from 131°C at no load to 400°C at full load whereas for G P 3 it varies from 135°C at no load to 410°C at full load. Lower exhaust gas temperature for G P 9 due to higher range turbulence created in the combustion chamber by the modified piston. It was observed that there is a decrease of 11% for G P 9 compared with normal piston configuration

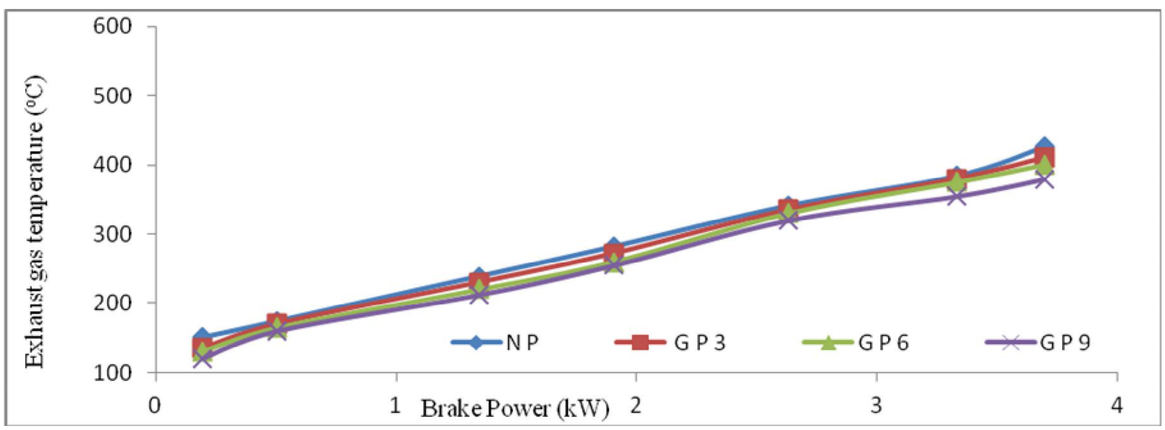


Figure 3: Comparison of Exhaust Gas Temperatures with Different configurations of Piston

Ignition Delay

The variation of ignition delay with brake power for different modes of combustion was shown in Fig. 4. It was inferred that ignition delay is decreased with an increase in brake power for almost all modes of combustion. With an increase in brake power, the amount of fuel being burnt inside the cylinder gets increased and subsequently the temperature of in- cylinder gases gets increased. This may lead to reduced ignition delay in all modes of combustion. However, the ignition delay for diesel fuel was lower under G P 9 than the normal piston configuration. It was observed that the ignition delay of G P 9 with brake power varies from 12.5 ^{o}C A at no load to about 8.8^{o}CA at full load. The reduction in the ignition delay of G P 9 is about 2.2% at max power compared to normal engine.

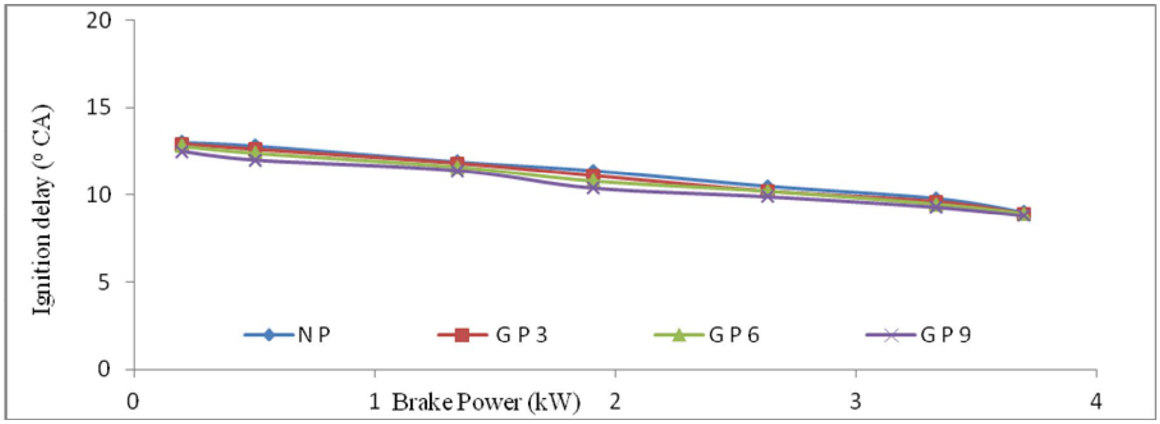


Figure 4: Comparison of Ignition Delay with Different configurations of Piston

Hydrocarbon Emissions

The comparison of Hydrocarbon emission in the exhaust is shown in Fig 5. Unburnt hydrocarbon emission is the direct result of incomplete combustion. It is apparent that the hydrocarbon emission is decreasing with the increase in the turbulence which results in complete combustion. At the rated load with GP9 a maximum reduction of hydrocarbon emission level is observed and is about 10.9% compared to normal engine. It is also observed that with GP3 and GP6 the reduction in hydrocarbon levels by about 3% and 7.8% compared to normal engine.

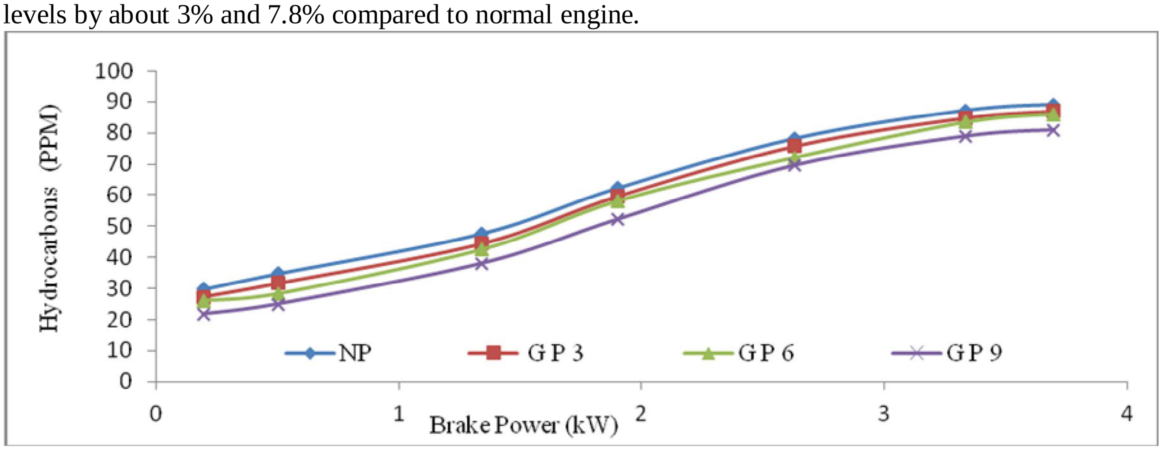


Figure 5: Comparison of hydrocarbons with Different configurations of Piston

Smoke Density

Smoke is nothing but solid soot particles suspended in exhaust gas. Fig.6 shows the comparison of smoke level with brake power. It can be observed that smoke increases with increase in brake power. The higher prevailing operating temperatures due to higher turbulence in the combustion chamber result better combustion and oxidation of the soot particles which further reduce the smoke emissions. Due to the complete combustion of diesel with excess air the smoke emissions are marginal. At the rated load, the smoke emissions of GP 9 are reduced by about 17% compared to normal engine..

Carbon Monoxide Emissions

Fig. 7 shows the comparison of Carbon monoxide emission with brake power. Generally, C.I engines operate with lean mixtures and hence the CO emission would be low. With the higher turbulence and

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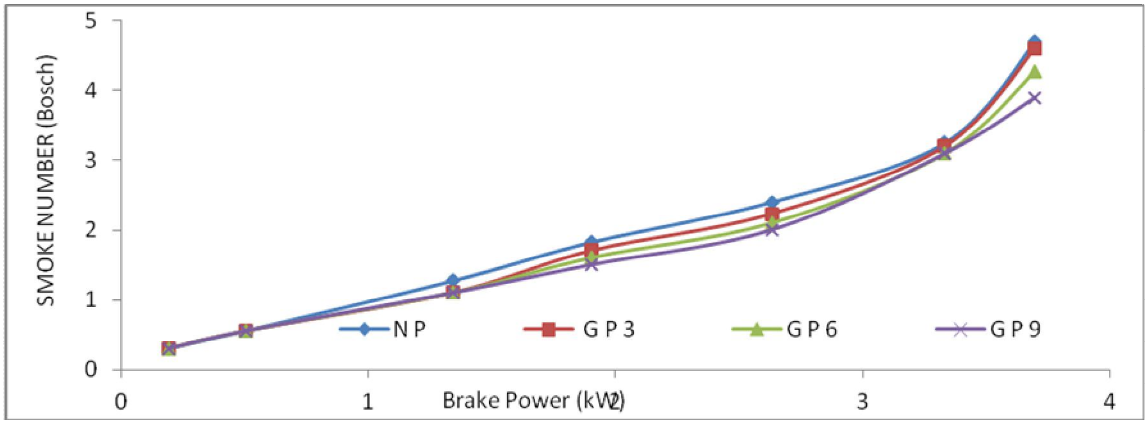

Figure 6: Comparison of Smoke Densities with Different configurations of Piston

temperatures in the combustion chamber the oxidation of carbon monoxide is improved and which reduces the CO emissions. The lowest carbon monoxide emission is with GP9 configuration compared to normal engine configuration and is about 30.6% by volume at rated load. It is also observed that with GP 6 and GP 3 the reduction in CO levels is about 29.4% and 17.7% compared with normal engine.

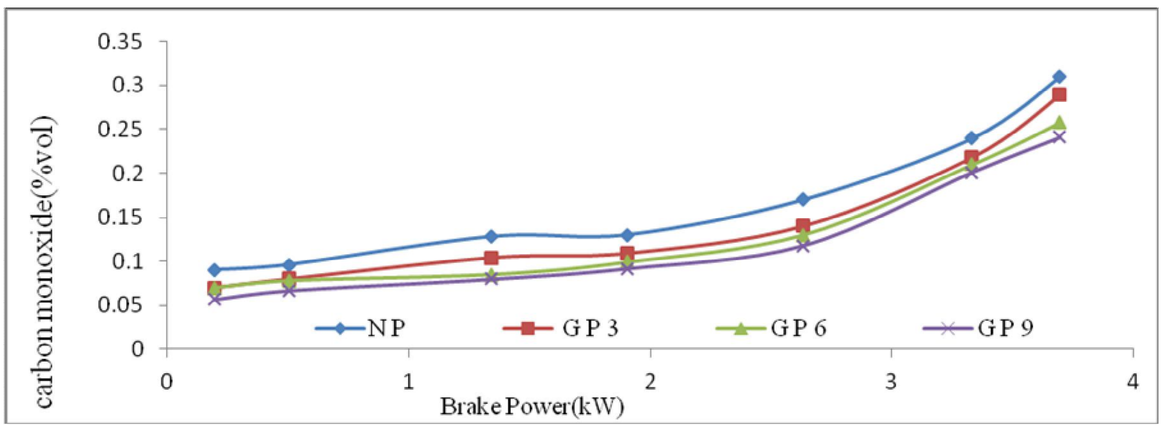

Figure 6: Comparison of Carbon monoxide with Different configurations of Piston

Conclusions

The Configuration GP 9 enhances the turbulence and hence results in better air-fuel mixing process among all three configurations of diesel engine. As a result, the thermal efficiency is increased and SFC and soot emission are reduced, although the Nox emission is increased owing to better mixing and a faster combustion process. Globally, since the reduction of soot is larger than the increase of NOx. It can be concluded that GP 9 is the best trade-off between performance and emissions.
Based on this investigation, the following conclusions are drawn:

- More power output is derived from the same given charge operating on the same compression ratio.
- Lesser smoke due to complete complete combustion..
- Lesser carbon deposits in the combustion chamber, piston crown and exhaust system occur due to controlled complete combustion.
- Exhaust gas temperatures is lower due to quicker and even flame propagation.
- There is better fuel economy due to improved and complete combustion.
- Raise in the cylinder pressure due to the effective combustion

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