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Experimental Investigation on the Effect of Air Swirl on Performance and Emissions Characteristics of a Diesel Engine Fueled with Karanja Biodiesel

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Abstract—In the 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 by using karanja biodiesel 'K20'(20% Karanja oil blended with 80% diesel on volume basis) is 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 grooves 3,6,9 are used to intensify the swirl for better mixing of fuel and air and their effects on the performance and emission with 'K20' are recorded.

Key words—Diesel engine, bio diesel, air swirl, cylinder, efficiency, emissions.

I. 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]. This higher swirl number reduces the soot emission at the cost of higher NO_x level.

On the face of the upcoming energy crisis, vegetable oils have come up as a promising source of fuel. They are being studied widely because of their abundant availability, renewable nature and better performance when used in engines. Many vegetable oils have been investigated in compression ignition engine by fuel modification or engine modification.

The use of vegetable oil in a diesel engine is not a new concept. In fact early engines were demonstrated with straight vegetable oils (SVO). Vegetable oils were proved to be very costlier during those days. However due to limited reserves of fossil fuels, escalation nature of diesel fuel prices and increase in environmental pollution, created a renewed interest of research in vegetable oil as substitute fuel for diesel engines. Vegetable oil is easily available, renewable and environment friendly. However major disadvantage of vegetable oil is its viscosity, which is much higher than that of diesel[9].

Bajpai *et al.* [10] tested performance and emission characteristics of karanja SVO blending with diesel and concluded that without major engine modifications blending up to 10 percentages can be utilized in the existing engine. In order to study the effect of injector opening pressure (IOP) on performance, emission and combustion characteristics of diesel engine running on blends of karanja SVO Venkanna *et al.* [11] estimated the ignition delay period (IDP) and heat release rate at 75% load and found that IDP increases with increase in blend percentage from 20% to 30% and noticed decrease in net heat release rate. Avinash Agarwal *et al.* [12] carried out experiments on diesel engine running on karanja oil and their blends. Even without preheating engine was running successfully, they concluded that performance and emission characteristics are well comparable with that of diesel and also up to 50% with and without preheating can be used in diesel engines.

Karnja is a forest based tree-borne non-edible oil with a, production potential of 135 000 million tones [13]. Karanja tree grows all over the country. In parts of India, this tree is also known as pongamia, belongs to the family of Leguminaceae. It is a medium sized tree that attains a height of about 1.8 m and a trunk diameter greater than 50 cm. The fresh extracted oil is yellowish orange to brown and rapidly darkens on storage [14].

II. EXPERIMENTAL SETUP

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.

An attempt is made in this work with variable number of grooves on the crown of the brass piston along with knurling on the face of the piston crown. Elliptical grooves of size 16 mm X 6 mm X 2 mm are prepared on the piston crown (fig.1). The experiments are conducted on a diesel engine by varying number of grooves on the crown of brass piston with Karanja bio diesel 'K20'(Karanja oil 20% + Diesel 80%). The results of three different configurations of brass piston attempted with K20, are compared with the normal engine.

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

GP1 GP2 GP3

Fig.1 Different types of configurations of piston crowns

N P - Normal Piston ; GP1 - Piston with 3 grooves GP2 - Piston with 6 grooves; GP3 - Piston with 9 grooves

III. RESULTS & DISCUSSIONS

1. Brake Thermal Efficiency

The variations of brake thermal efficiency with power output for the piston with different configurations are shown in Figure 1. The brake thermal efficiency for normal engine at 3/4 of rated load is 26.1%. It can be observed that the engine with GP9 and GP12 give thermal efficiencies of 27.9% and 27%, respectively, at 3/4 of rated load. From Figure, it is inferred that the brake thermal efficiencies are increasing with an increase in brake power for configurations that are under consideration. It is also observed that there is a gain of 6.9% with GP2 compared to normal engine. This may be due to the enhanced air swirl in the combustion chamber which resulted in better mixing of fuel and air and as well as complete combustion of the charge in the combustion chamber.

2. Brake Specific Fuel Consumption

The variations of brake specific fuel consumption with brake power for different configurations are shown in Figure 2. The brake specific fuel consumption for normal engine at 3/4 of rated load is 0.34 kg/kW-hr. It can be observed that the engine with GP1, GP2 and GP3 give brake specific fuel consumption of 0.31 kg/kW-hr and 0.32 kg/kW-hr and 0.33 kg/kW-hr respectively, at 3/4 of rated load. From Figure 4.3, it is inferred that the brake specific fuel consumption is increasing with an increase in brake power for configurations that were under consideration. It is also observed that the GP2 has the lowest fuel consumption of 8.8% when compared with normal engine. This is because of the complete combustion of charge in the combustion chamber by liberating maximum energy due to the inducement of enhanced air swirl in the combustion chamber.

3. Ignition Delay

The variation of ignition delay with brake power for different configurations is shown in Figure 3. It is inferred that ignition delay, decreases with an increase in brake power for almost all configurations. With an increase in brake power, the amount of fuel being burnt inside the cylinder is increased and subsequently the temperature of in-cylinder gases is increased. This may lead to reduced ignition delay in all configurations. However, the ignition delay for diesel fuel was lower under GP1, GP2 and GP3 configurations than the normal engine. It is observed that the ignition delay of GP1, GP2 and GP3are 10.2^o CA, 10.7^o CA and 10.8^o CA at 3/4 of rated load respectively. The reduction in the ignition delay of GP2 is about 7.3% at 3/4 of rated load when compared to normal engine. This is due to the fast and complete burn of the charge because of the flow of high velocity flames through the grooves in the piston of the combustion chamber.

4. Smoke Density

Smoke is solid soot particles suspended in exhaust gas. The comparison of smoke level with brake power is shown in Figure 4. It can be observed that smoke increases with increase in brake power. The smoke number for GP1, GP2 and GP3 are 2.32 BSU, 2.36 BSU and 2.4 BSU respectively, whereas for normal engine it is 2.46 BSU. This is because of higher turbulence in the combustion chamber which results in the better combustion and oxidation of the soot particles, which further reduces the smoke emissions. Due to the complete combustion of diesel with excess air, the smoke emissions are marginal. At 3/4 of the rated load, the smoke emissions for GP2 are reduced by about 5.7 % when compared to normal engine.

5.Nitrogen Oxide Emissions

The comparison of NO_x emission with brake power for different configurations is shown in Figure 5. It can be observed from the figure that NO_x emission increases with increase in turbulence in the cylinder because of high temperature. The NO_x emissions for GP1, GP2 and GP3 are 552 ppm, 555 ppm and 559 ppm respectively, whereas for normal engine it is 562 ppm. The NO_x emissions are lower by 1.8 % for GP2 when compared to normal engine at 3/4 of rated load. This may be due to decrease in combustion duration; the residence time for the gas in the combustion chamber is too short to form the NO_x at normal level.

6. Hydrocarbon Emissions

The comparison of Hydrocarbon emission with brake power is shown in Figure 6. The HC emissions for GP1, GP2 and GP3 are 76 ppm, 77 ppm and 77.8 ppm respectively, whereas for normal engine it is 78.2 ppm. The HC emissions are lower by 2.8% for GP2 when compared to normal engine at 3/4 of rated load. The Un-burnt 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.

7. Carbon monoxide Emissions

The comparison of Carbon monoxide emission with brake power is shown in Figure 7. The CO emissions for GP1, GP2 and GP3 are 0.15, 0.16 and 0.165 % by volume respectively, whereas for normal engine it is 0.17 % by volume. The CO emissions are lower by 11.76% for GP2 when compared to normal engine at 3/4 of rated load. Generally, C.I engines operate with lean mixtures and hence the CO emission would be low. With the higher turbulence and temperatures in the combustion chamber, the oxidation of carbon monoxide is improved and which reduces the CO emissions.

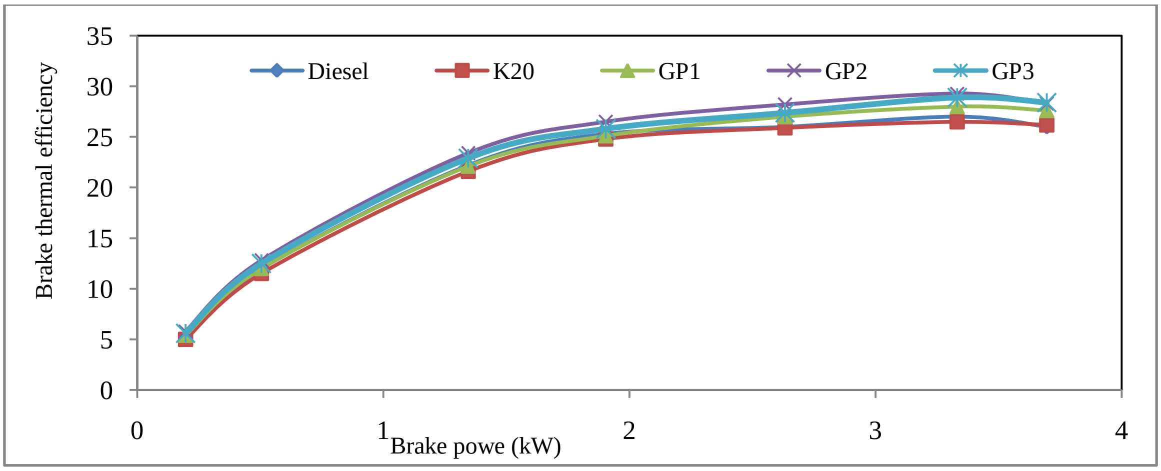

Fig. 1 **Comparison of Brake thermal Efficiency with different configurations of grooved piston.**

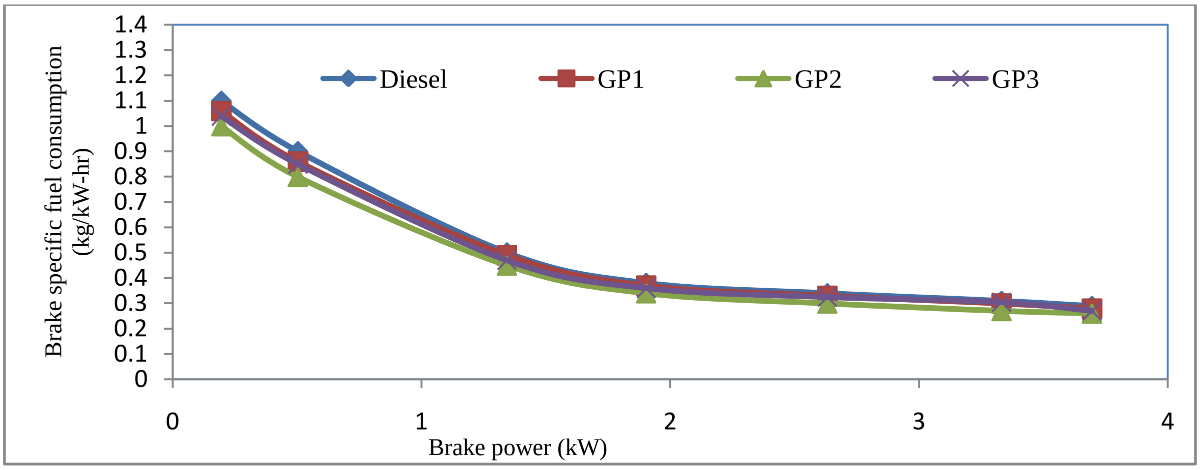

Fig: 2 **Comparison of Brake specific fuel consumption with different configurations of grooved piston.**

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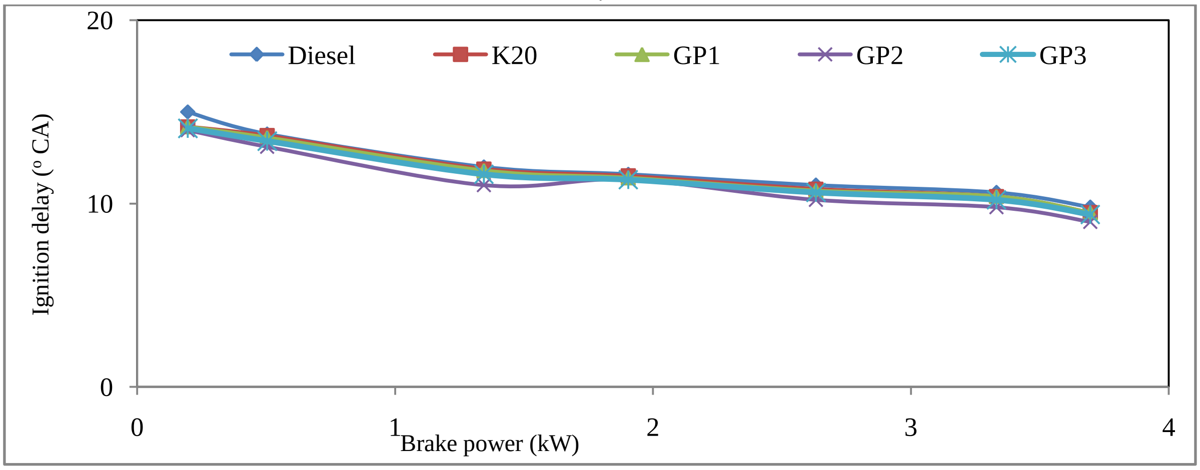

Fig. 3 **Comparison of Ignition delay with different configurations of grooved piston.**

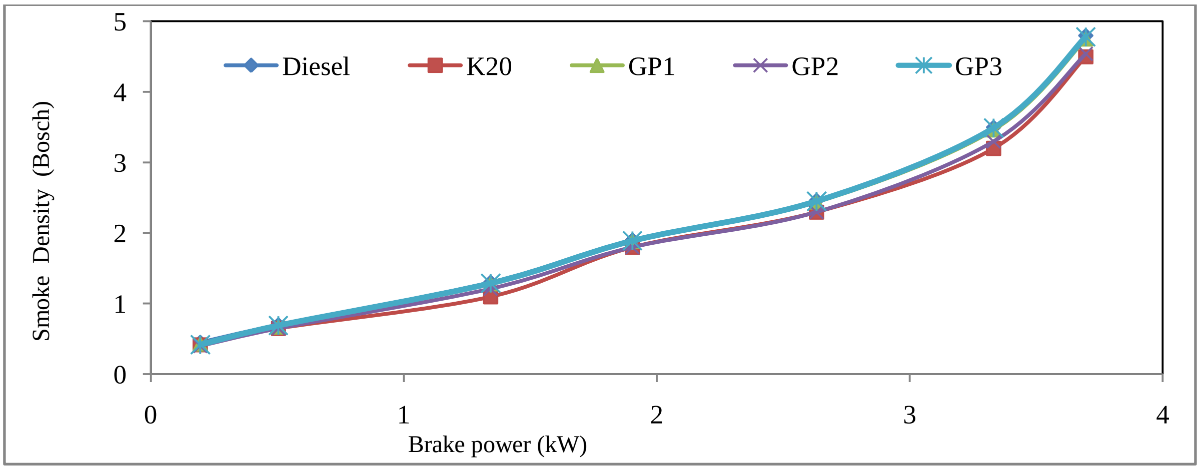

Fig: 4 **Comparison of Smoke Density with different configurations of grooved piston.**

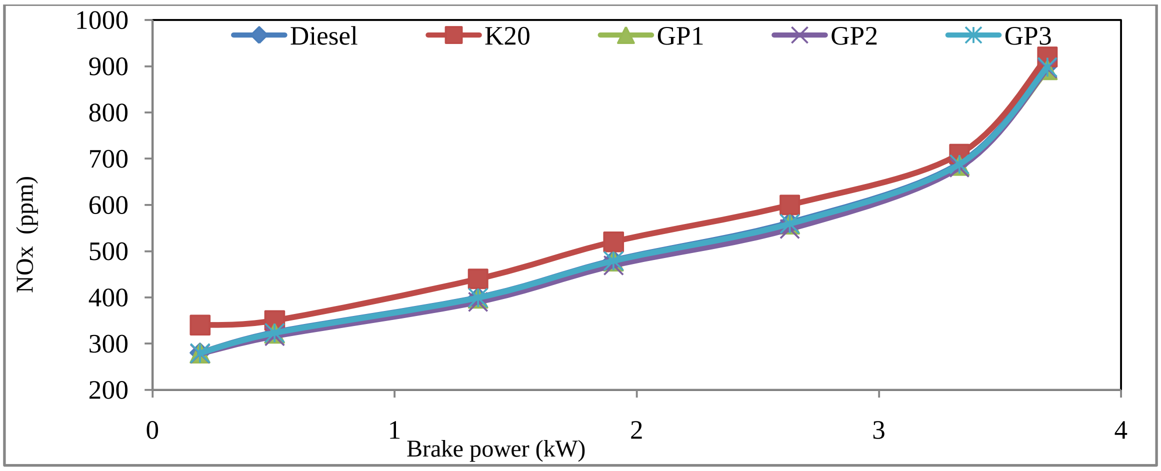

Fig. 5 **Comparison of NO_x with different configurations of grooved piston.**

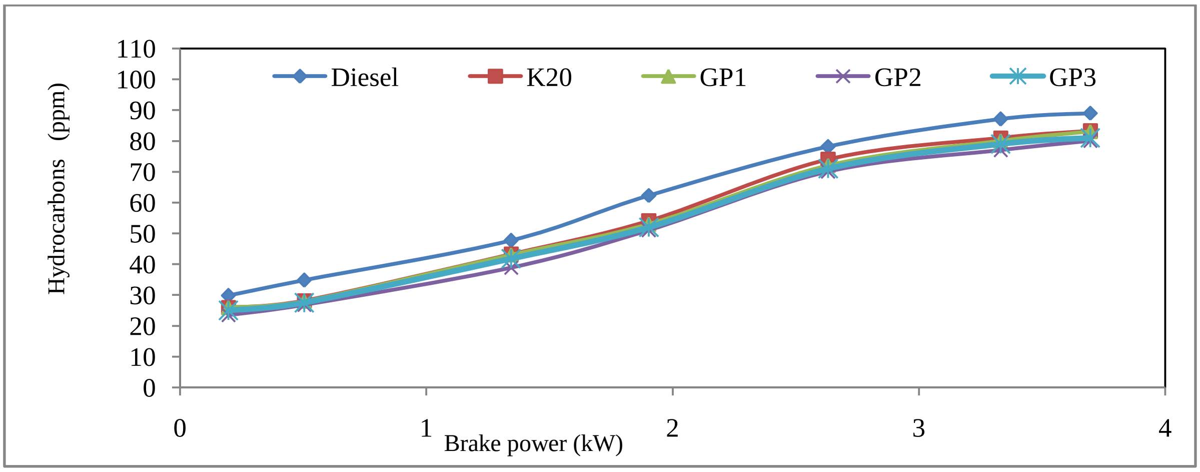

Fig .6 **Comparison of HC with different configurations of grooved piston.**

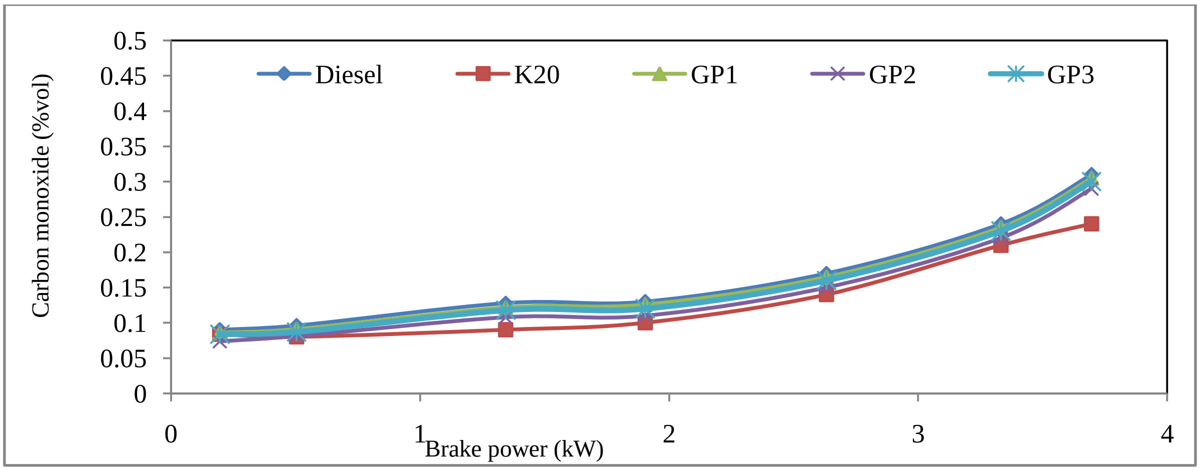

Fig. 7 **Comparison of CO with different configurations of grooved piston.**

IV. CONCLUSIONS

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 about 8.8%.
- 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 about 5.9%.
- The maximum reduction in NO_x 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%.

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 with karanja biodiesel K20 gives better performance in all the aspects.

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