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Experimental Investigations on the Effect of Piston Geometry on CI Engine performance with Alternate Fuels

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ABSTRACT

Bio-diesels have received a lot of attention as an alternate vehicular fuel. But the properties of bio-diesels are not the same as diesel fuels especially their higher viscosity and low volatility. Also the bio-diesels have very poor atomization characteristics due to decreased cone angle during fuel injection. Hence there is a need to customize the engine properties to adjust the limitations of these bio-diesels. Towards this end, in this paper two different piston geometries are considered along with standard one. The biofuels considered are Cuban royal seed oil, Thurayi seed oil and their blends with diesel. Parameters like Brake thermal efficiency, Specific Fuel consumption, Smoke density, CO emissions, HC emissions, NOx emissions, cylinder pressure and Heat release rate are measured and plotted.

Keywords: Emission parameters, Piston geometry, Royal palm seed oil, Thurayi seed oil

I INTRODUCTION

As engines have evolved over the years, pistons have evolved with them. They're getting shorter and lighter, and use smaller skirts – the cylindrical "body" of the piston. Newer pistons are often made of aluminum alloys comprised of more silicon than in the past. This improves resistance to heat and reduces thermal expansion. One of the biggest advancements in piston technology is the use of different piston "tops" or "crowns," the part that enters the combustion chamber and is subjected to combustion [1]. While older piston tops were mostly flat, many now feature bowls on top that have different effects on the combustion process. The piston bowl is primarily used in diesel engines. Direct Injection Diesel engines don't have an ignition phase, so the piston crown itself may form the combustion chamber [2].

These engines often use pistons with differently shaped crowns, although with direct injection becoming increasingly popular, gasoline engines are starting to use them as well. The shape of the piston bowl controls the movement of air and fuel as the piston comes up for the compression stroke (before the mix is ignited and the piston is pushed downward.) The air and fuel swirl into a vortex inside the piston bowl before combustion (or compression) takes place, creating a better mixture. By affecting the air/fuel mixture, you can achieve better and more efficient combustion, which leads to more power [3]-[6]. The bowls have a variety of different shapes — some are also designed to optimize fuel economy. With direct injection becoming the hottest new technology for gasoline engines, expect uniquely-bowled pistons to become more and more popular [7][8].

II ENGINE MODIFICATION

The Performance, exhaust emissions and combustion characteristics of diesel engines depend on various factors like the engine design, operating parameters and fuel properties [9]-[12]. The engine design, particularly the combustion chamber design in a direct injection diesel engine has to achieve a high degree of air movement inside the cylinder in terms of swirl, squish and turbulence, in order to prepare better air-fuel mixture, to promote the evaporation in a very short time and to achieve higher combustion efficiency [13]. If a good mixture can be achieved, the resulting combustion is both clean and efficient, with all the fuel burned and minimal exhaust remaining. The conventional combustion chamber has been optimized for combustion of diesel fuel, including improvement of mixing between injected diesel and in-cylinder air, but not for derived alternate fuel [14][15]. With this background, in order to achieve enhanced engine characteristics with the derived alternate fuel from biodiesel, combustion chamber modification is mandatory.

The dimensions of the pistons are chosen so as to maintain the same piston bowl volume. The fuel injection quantity was maintained the same for all the piston geometries. In respect of the design modification with the combustion chamber geometry, the conventional hemispherical combustion chamber is modified to have SCC and TCC [16][17]. The piston bowl geometry is modified without changing the compression ratio of the engine. This is realized by modelling the combustion bowl geometry using CAD initially and when the volume is found to attain the constant value, the geometric dimension are finalized. Followed by this, the combustion bowl in piston geometry is fabricated based on the obtained design and used for the experimental investigation in a diesel engine fueled with biodiesel. Two piston configurations, P1 toroidal combustion chamber (TCC) and P2 swirl blade combustion chamber (SCC) were designed in such a way that the piston bowl volumes of these modified bowls are exactly the same with that of the original engine [18]-[20].

This will ensure a similar compression ratio between models. In the design of SCC, a small baffle plate is welded in the piston. The swirl inducing blade has six holes of 2.5mm diameter were drilled. This setup was chosen to investigate, how the distribution of vortices inside the bowl enhances combustion. The piston configurations are shown below in Fig. 1.

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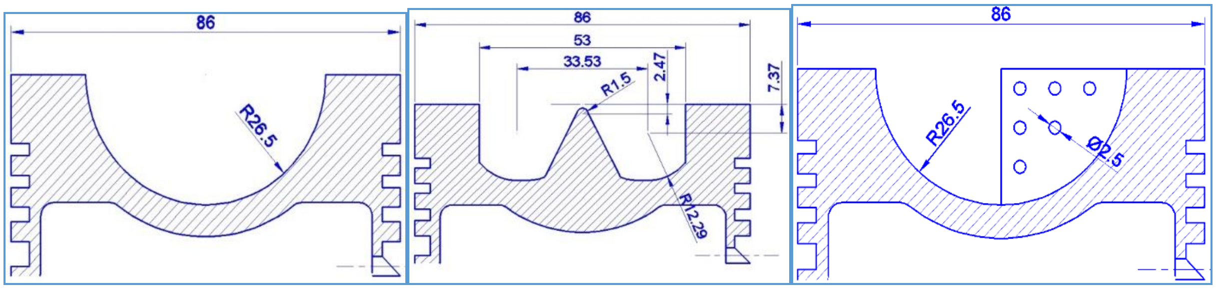

Figure 1. Dimensions for standard piston (SP) (Left), Toroidal combustion chamber (TCC) (Center) andSwirl blade combustion chamber (SCC) (Right)

In the event of optimization of combustion chamber geometry for trans-esterified DR25, RR25 blends, three different shapes of combustion chamber geometry such as SP- standard piston (hemispherical combustion chamber) P1- modified piston 1 (toroidal combustion chamber), P2- modified piston 2 (swirl blade combustion chamber) were selected. In the design aspect of every combustion chamber, fabrication is done in such a way that the volume of the combustion bowl is not altered so as to maintain the same compression ratio for all configurations. The photographic view of the modified piston bowl geometry, employed in the current work, has been depicted in Fig.2. Experiments are carried out in a diesel engine, after assembling and dis-assembling the modified combustion chambers sequentially.

Figure 2. SP: Standard piston; P1-Toroidal combustion chamber (TCC); P2-Swirl blade combustion chamber (SCC)

III EXPERIMENTAL SETUP AND ARRANGEMENT

a. Test engine

The experiment was conducted on Kirloskar TV-1, single cylinder, four-stroke, water cooled DI diesel engine with a displacement of 661cc. The rated power of the engine is 5.2 kW at 1500 rpm with constant speed. The schematic view of the experimental setup is shown in Fig. 3. The engine had a hemispherical bowl piston, 3 holes injector. The inline mechanical fuel pump was operated at a standard injection pressure of 220 kg/cm^2 and the recommended injection timing of 23° bTDC. The governor was used to control the speed of the engine. Cooling of the engine was accomplished by supplying water through the jackets in the engine block and cylinder head. A hole was made on the top of the cylinder head surface area to place the piezoelectric pressure transducer for measuring the heat release rate and cylinder pressure. The engine was stabilised for a particular operating point, fuel flow rate and exhaust gar temperatures are recorded. The engine was allowed to run for 15 to 20 minutes to attain the steady state condition to reach cooling water temperature of 70 °C.

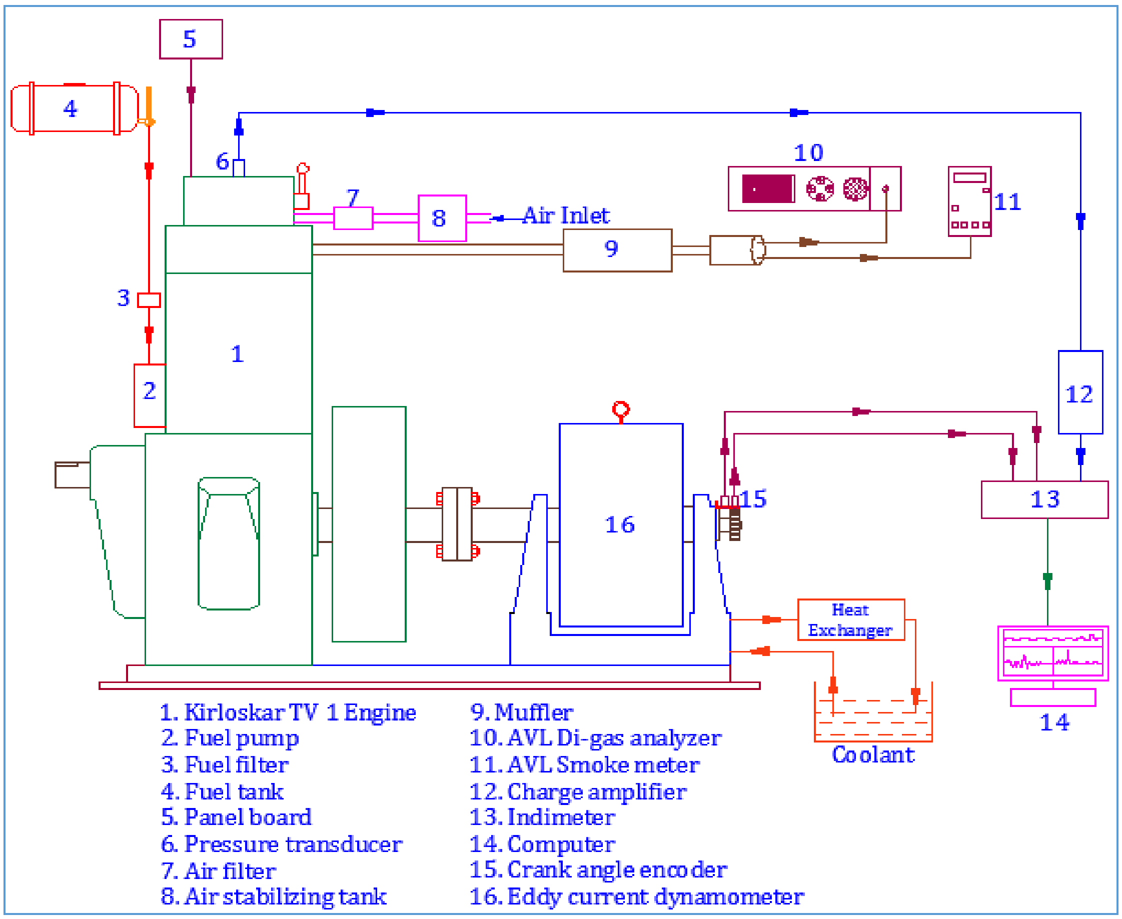

Figure 3. Schematic view of the experimental setup

b. Load and speed measurement

The test engine was coupled with an eddy current dynamometer for measuring the load with more accuracy. The dynamometer functions according to the eddy current principle and is used to determine the effective power generated by the engine. The engine was set to operate at a constant speed of 1500 rpm. The load of the engine was measured from load cell and the speed of the engine was controlled by a governor mechanism.

c. Combustion measurement

The AVL combustion analyzer was used to measure the combustion parameters such as cylinder pressure and heat release rate. The combustion analyzer consists of pressure transducer, crank angle encoder, charge amplifier and Indimeter.

d. Pressure measurement

The piezoelectric pressure transducer was used to measure the in-cylinder pressure of the test engine. An AVL-make transducer with a sensitivity of 16.11 pC/bar was used. The pressure transducer was mounted by suitable adapter. The pressure transducer produces a charge output proportional to the in-cylinder pressure. The charge output was supplied to an AVL-make charge amplifier where it was amplified for an equivalent voltage that was converted to the actual pressure.

e. Crank angle encoder

For the measurement of cylinder pressure with respect to the crank position, a crank degree marker, an electromagnetic pickup and a signal-processing unit were used. The marker was fitted concentrically to the end of the crank shaft of the engine.

f. Charge amplifier

In order to obtain sharp, flicker free and stable signal of any desired portion of the cylinder pressure trace, the charge amplifier was used. The signal derived from the pickup that was suitability positioned with respect to single pulse generator installed on the extension of the generator shaft was fed to a charge amplifier. The output of the amplifier has been sent to the computer for the post-processing of the best results.

g. Data acquisition system

The AVL 619 Indwin software is easy to operate, menu driven parameter editor for setting up the system, utilized for TDC detection, the numerical monitor for on-line display of calculated results like Indicated Mean Effective Pressure (IMEP) or mass burn fractions as well as monitor programme for curve display. This versatile software is designed by AVL, Austria, and used for online data acquisition from the pressure transducer and crank angle degree marker.

h. Measurement of smoke density

The smoke meter works on the light extinction principle. It consists of a flexible sampling hose with appropriate exhaust gas probe. The sampling probe is inserted in the exhaust pipe approximately 200 mm from the end of the exhaust pipe. A continuous exhaust sample is passed through the tube of about 46 cm length, which had a light source at one end and the other end is fitted with a photo cell. The amount of the light passing through the smoke column is sensed as an indication of the smoke level. The smoke meter consisted of a display unit giving out the smoke density in HSU.

i. Measurement of HC, CO and NO_x Emissions

The exhaust emission of hydrocarbon, carbon monoxide and oxides of Nitrogen are measured on the dry basis. AVL – DI gas analyzer was used for the test. The exhaust sample to be assessed is passed through a cold trap (moisture separator) and filter element to prevent water vapour and periodically calibrated with standard gas, in conformity with the instruction of the manufacturer. Hydrocarbon and oxides of nitrogen are measured in parts per million (ppm), while carbon monoxide emission are measured in terms of percentage by volume. The specification of the AVL Di gas analyzer is shown in appendix E. The electro-chemical sensor is used to measure NOx emission and the unit is represented by ppm. CO and HC emission are measured by the NDIR principle.

j. Experimental Procedure

The test engine was operated by diesel, Thurayi and Cuban Royal Palm seed biodiesel with different blends ratio by using both standard and modified pistons. The engine was allowed to run with neat diesel at a various loads for nearly 10 minutes to attain the steady state with constant speed conditions. Then the following observations were made.

- The water flow was started and maintained constant throughout the experiment.
- The load, speed and temperature indicators were switched ON.
- The engine was started by cranking after ensuring that there was no load.
- The engine was allowed to run at the speed rate of 1500 rpm rev/min for a period of 20 minutes to reach the steady state.
- The fuel consumption was measured by a stop watch for 10cc consumption.
- Smoke readings were measured by using the AVL smoke meter at the exhaust outlet.
- The combustion parameters were measured by AVL combustion analyzer.
- The amount of CO, HC and NO_x emission was measured by using AVL Di gas analyzer.
- The exhaust temperature was measured at the indicator by using a sensor.
- Then the load was applied by adjusting the knob, which was connected to the Eddy Current Dynamometer.

IV PERFORMANCE ANALYSIS WITH MODIFIED PISTON GEOMETRY

In this section the performance of the IC engine with different piston geometry is presented. It is observed that the performance is optimum with Thurayi seed oil blend DR25. Brake thermal efficiency, Specific Fuel consumption, Smoke emission, CO emission, HC emission, NOx emissions, cylinder pressure and Heat release rate with DR25 are shown in Figures 4 to 11.

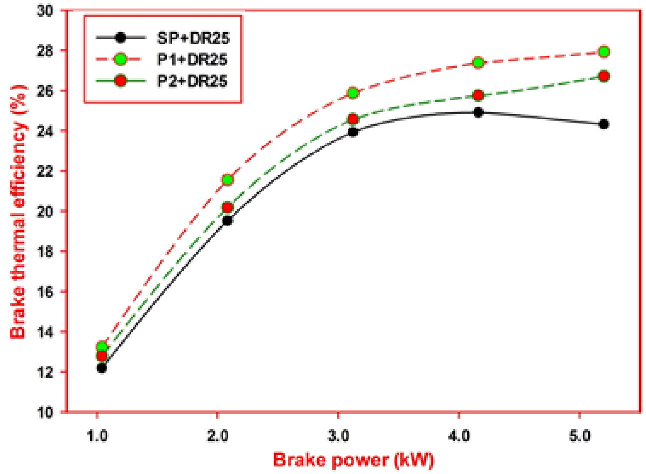

Figure 4. Brake thermal efficiency using modified pistons with DR25 biodiesel blend

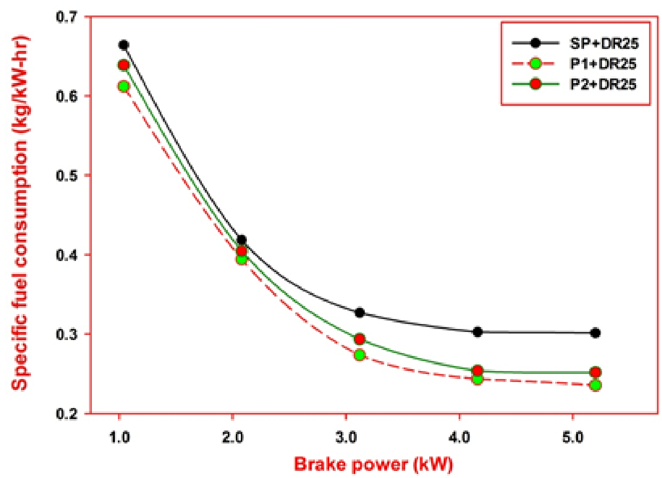

Figure 5. SFC using modified pistons with DR25 biodiesel blend

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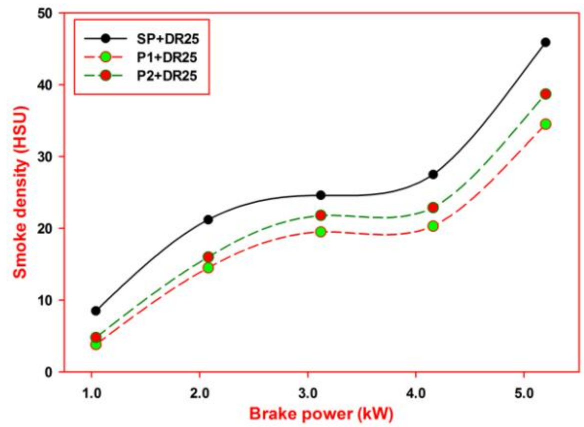

Figure 6. Smoke density using modified pistons with DR25

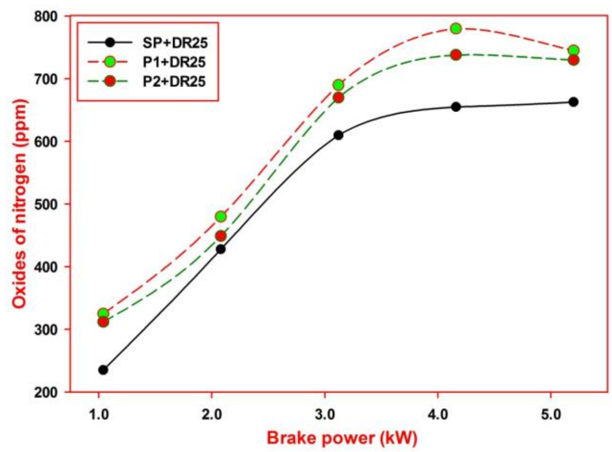

Figure 7. NOx emissions using modified pistons with DR25 biodiesel blend

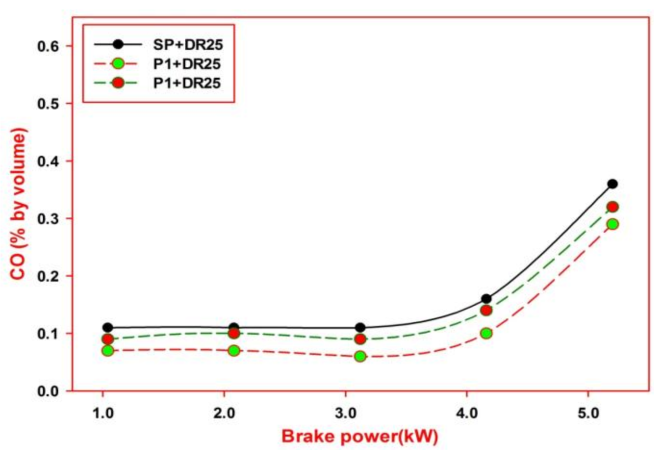

Figure 8. CO emissions using modified pistons with DR25 biodiesel blend

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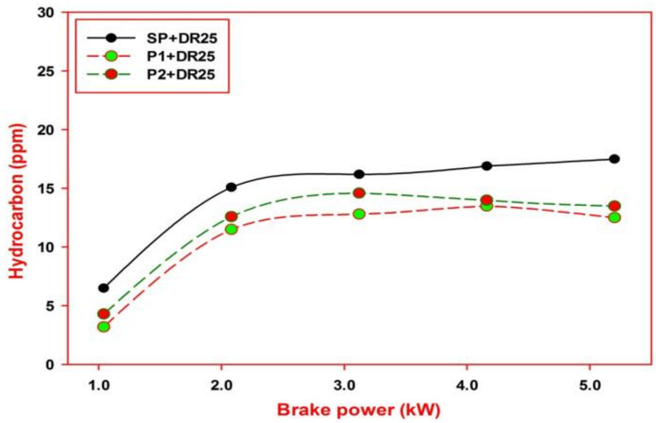

Figure 9. HC emissions using modified pistons with DR25 biodiesel blend

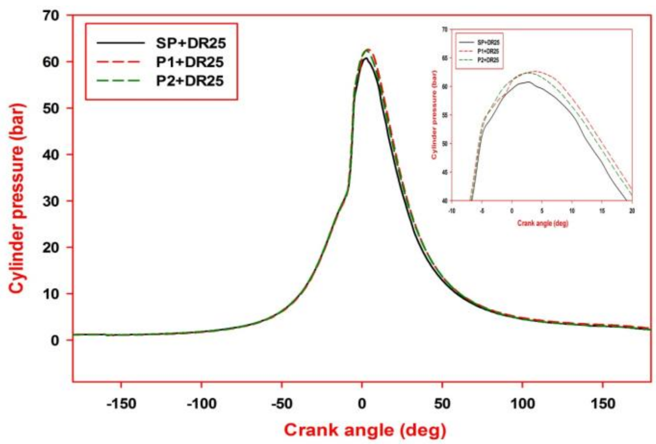

Figure 10. Cylinder pressure using modified Pistons with DR25 biodiesel blend

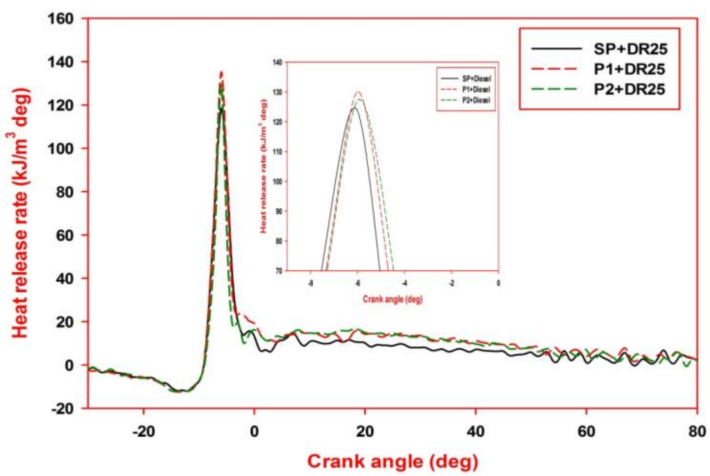

Figure 11. Heat release rate using modified pistons with DR25 biodiesel blend

V CONCLUSIONS

From the experimental investigation, P1 piston was found to evince better combustion for the reported blend DR25 than that of other piston configurations with an increased heat release rate and cylinder pressure of about 2.2% and 8.5% respectively for DR25 with P1 than that of diesel with SP. The other piston and biodiesel blend having lower heat release and cylinder pressure. The BTE of DR25 with P1 shows an increase of about 2.3% when compared to that of diesel with SP. The gaseous emission smoke, HC, and CO for DR25 biodiesel blend with piston P1 were noted to be decreased by 20%, 21.9% and 20% respectively at the penalty of increased NO_X emission by 8.5% when compared to that of diesel with SP.

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