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Experimental and Computational Investigations for Combustion, Performance and Emission Parameters of a Diesel Engine Fueled with Soybean Biodiesel-Diesel Blends

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Abstract

In this work experimental and theoretical investigations were carried out on a single cylinder, direct injection diesel engine operating on different blends of a soybean methyl ester (SME) with diesel fuel. The effect of blending on the cylinder pressure, heat release rate, carbon monoxide (CO), unburned hydrocarbon (UHC), nitrogen oxides (NOx), and smoke opacity were measured. The results indicate that the use of biodiesel produces lower smoke opacity up to 48.23% with 14.65% higher brake specific fuel consumption (BSFC) compared to diesel fuel. The measured CO emissions of B20% SME and B100% SME were found to be 11.36% and 41.7% lower than that of diesel fuel respectively. All blends of SME were found to emit significantly lower UHC concentration compared to that of diesel over the entire load. NOx emissions are observed to be higher for all blends of SME. The experimental results are compared with the results of Diesel-rk software and a good agreement between them is noticed.

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Keywords: Biodiesel; Combustion ; Emissions; Diesel-rk Software.

1. Introduction

A significant portion of modern energy demands are met by the use of petroleum based fuels especially in the transportation sector. On the other hand the fossil fuel combustion continually accumulating greenhouse gases into the atmosphere is responsible for the global warming. Furthermore, the regulations for particulate matter (PM) and NOx emissions from diesel engines have strengthened, and reductions in carbon dioxide (CO_2), which is a greenhouse gas, emission, also raised important issues.

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These reasons have pushed the countries towards searching for the alternative energy sources with particular emphasis on those renewable in nature. Future projections indicate that the only feasible option is the production of alternative fuels derived from non-petroleum sources [3]. For substituting the petroleum fuels used in internal combustion engines, fuels of bio-origin provide a feasible solution to the twin crises of 'fossil fuel depletion' and 'environmental degradation'. The fuels of bio-origin may be alcohols, biodiesel, and biogas. Some of these fuels can be used directly while others need to be formulated to bring the relevant properties close to conventional fuels [4]. Vegetable oil esters are receiving increasing attention as a non–toxic, biodegradable, and renewable alternative diesel fuel. These esters have become known as "biodiesel." Many studies have shown that the properties of biodiesel are very close to those of diesel fuel. Therefore, biodiesel can be used in diesel engines with few or no modifications. It has higher cetane number than petroleum diesel fuel, no aromatics, and contains 10% to 11% oxygen by weight. These characteristics of biodiesel reduce the emissions of CO, UHC, PM in the exhaust gas compared with diesel fuel [5-6]. Considerable research has been conducted to investigate the properties of biodiesel and its performance in engines [7-15]. Since the majority of modern diesel engines have direct injection (DI) fuel systems, these engines are more sensitive to fuel spray quality than indirect injection engines. Therefore, a fuel with properties that are closer to No. 2 diesel fuel is needed. The current study was to investigate the combustion characteristics, performance and emissions of a diesel engine operating on soybean oil-based biodiesel and compared them to the performance and emissions when the engine was operated on petroleum-based diesel fuel.

2. Biodiesel Preparation and Specifications

The direct use of vegetable oils for a long time in diesel engine without pre treatment or engine modification causes serious engine problems. Neat oil is converted into methyl ester of oil (biodiesel) by using one of the four well known techniques, proposed to reduce the viscosity levels of vegetable oil namely dilution, pyrolysis, micro emulsion and transesterification. The biodiesel used in this study was provided from Intech energy systems Pvt. Ltd [16]. It uses a transesterification process together with methanol, which was catalyzed by potassium hydroxide. The process of transesterification removes glycerol from triglycerides and replaces it with radicals from the alcohol used for the conversion process. The process of transesterification is affected by the mode of reaction condition, molar ratio of alcohol to oil, type of alcohol, type and amount of catalyst, reaction time and purity of reactants [17].The properties of No.2 diesel fuel and SME biodiesel is shown Table 1.

Table 1 Physical and chemical properties of No. 2 diesel fuel and soybean methyl ester [16]

Property	No.2 Diesel fuel	SME
Chemical formula	$C_{13.77}H_{23.44}$	$C_{19}H_{35}O_2$
C/H ratio	6.90	6.51
Density at 15 °C (g/cm^3)	860	876
Viscosity at 40 °C (cst)	3.0	4.25
Molecular weight (kg/kmol)	190	292.2
Surface tension factor (N/m)	0.028	0.0433
Calorific value (MJ/kg)	42.5	36.22
Flash point (°C)	76	130
Cetane number	48	51.3
Total glycerine (%)	-	0.028
Free glycerine (%)	-	0.00

3. Experimental Setup and Procedure

The engine used in the present study is a Kirloskar TAF-1, single cylinder, air cooled, vertical and DI diesel engine with the specification given in Table 2. The schematic diagram of the experimental setup is shown in Fig.1. The engine was coupled to an eddy current dynamometer. The inlet side of the engine consists of anti pulsating drum and air temperature measuring device. The exhaust side of the engine consists of exhaust gas temperature indicator, exhaust gas analyzer and smoke meter. The test rig was installed with AVL software for obtaining various curves and results during operation. The SME was tested as pure fuel, as 20% blend and as 40% blend by volume with No.2 diesel fuel. The tests were conducted at five different engine loads in terms of brake power (0, 1.1, 2.2, 3.3 and 4.4 kW) at constant engine speed of 1500 rpm with diesel as fuel was carried out to generate baseline data at steady state conditions. Engine was run with each fuel for 2 hours continuously and then the test results were obtained and compared with the baseline data. Three sets of observations were taken and the average values are considered for this study.

Table 2 Specifications of Kirloskar TAF-1 diesel engine

Engine Make	Kirloskar TAF-1
Engine type.	4-Stroke, Diesel engine
Number of cylinder	1
Bore × stroke	87.5×110 mm
Cylinder capacity	0.66 L
Compression ratio	17.5
Rated power	4.4 kW , 1500 rpm
Maximum torque	28 N.m ,1500 rpm
Orifice diameter	0.15 mm
Injection pressure	220 bar

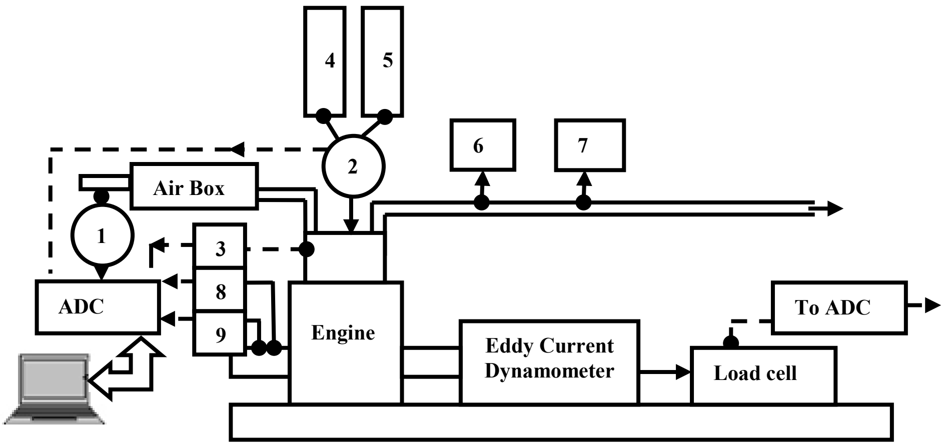

Fig. 1 Schematic diagram of the experimental set-up: 1.Air flow sensor; 2. Fuel flow sensor; 3. Pressure sensor; 4. Diesel tank; 5. Biodiesel tank ; 6. Five gas analyzer; 7. Smoke meter; 8. Speed meter; 9.Crank angle encoder

4. Theoretical Analysis

The software Diesel-rk is intended for the calculation and optimization of internal combustion engines. It has advanced RK-model of mixture formation and combustion in a diesel engine, and also the tool for multiparameter optimization [18]. In the multizone combustion model, the spray is split into seven characteristic zones, as shown in Fig. 2. In each zone specific evaporation and burning conditions are specified in the model. The spray evolution passes through three stages: (1) Initial formation of dense axial flow. (2) Main stage of cumulative spray evolution. (3) Period of spray interaction with the combustion chamber walls and fuel distribution on the walls. The border between the initial and main stages of spray evolution corresponds to the moment when the axial flow close to the spray tip starts to deform and break up, forming a condensed mushroom-shaped forward front.

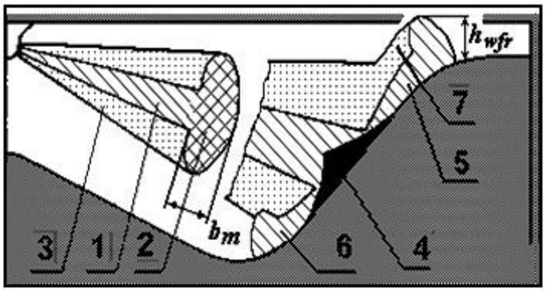

Fig. 2 Characteristic zones of the diesel spray

As the spray moves on, constant breakup of the spray forward part takes place and the front is renewed by new flying fuel portions. The delayed droplets move from the breaking front to the environment. The moving spray carries the surrounding gas with it. The gas velocity in the environment is low, but gas in the axial core is rapidly accelerated to the velocity close to that of droplets. The core diameter in the cross section is about 0.3 of the spray outside diameter. The current position and the velocity of an elementary fuel mass (EFM) injected during small time step and moving from the injector to the spray tip are related as:

$$\left(\frac{U}{U_o}\right)^{3/2} = 1 - \frac{l}{l_m} \qquad (1)$$

U: is the velocity of the control portion of the fuel (m/s), U_o: Initial velocity of the spray at the nozzle (m/s), U_m: velocity of the spray front (m/s), l: current length of the spray (m) and l_m: penetration distance of the control portion of the fuel (m). As an illustration, Fig. 3 presents the variation of spray evolution parameters as functions of time. The general principle evaporating equations are mentioned in [19], [20]. Same operating conditions and fuel properties with engine specifications were used as input data to the software.

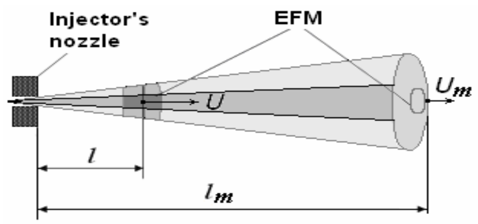

Fig. 3 Variation of spray evolution with time.

5. Results and Discussion

The combustion, performance and exhaust emissions parameters are compared for SME, 20% SME blend, 40% SME blend and No.2 diesel fuel. The blends were tested at different load conditions. However they are presented only for 100% load. This condition was chosen because it is the point of minimum air/fuel ratio and maximum smoke. This provides the best conditions for discerning any differences between the fuels.

5.1 Combustion analysis

Cylinder pressure: In a diesel engine the cylinder pressure depends on the fuel burning rate during the premixed burning phase and higher cylinder pressure ensures better combustion and heat release. Fig. 4 shows the variation of cylinder pressure with crank angle at full load for diesel and SME blends. It can be seen that cylinder pressure for soybean biodiesel is lower than that of No.2 diesel by 2.98% due to the reduction in the heat supply for the blended fuel. It is noted that the maximum pressure obtained for biodiesel is closer to top dead centre (TDC) than No.2 diesel fuel.

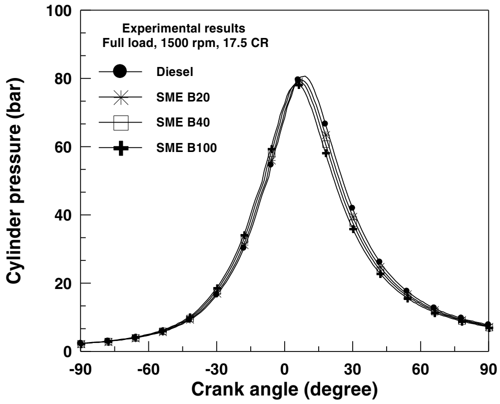

Fig. 4. Variation of cylinder pressure with crank angle

Heat release*:* Fig. 5 shows the integral heat release rate with crank angle at full load. It can be observed that the value of heat release rate decreases with increase in SME blends. It is evident from this figure that biodiesel blend had an earlier start of combustion, but slower combustion rate. The early start of combustion was caused by the advancement in the injection timing and shorter ignition delay. The slower premixed combustion rate due to less energy released in premixed phase and also probably due to the lower volatility of biodiesel. In the diffusion combustion phase, the SME biodiesel fuel had rapid combustion because at this phase most of fuel gets vaporized [21].

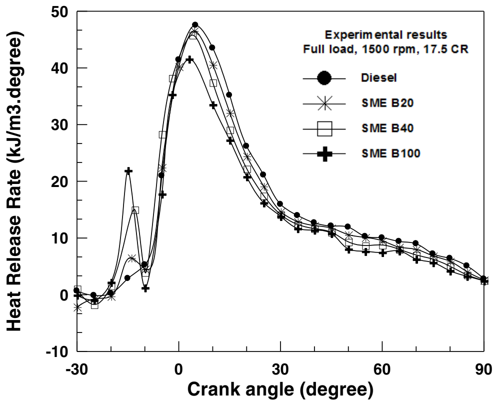

Fig. 5 Comparison of integral heat release rate with crank angle.

5.2 Performance analysis

In order to understand the effect of biodiesel on engine performance, the brake specific fuel consumption (BSFC) and brake thermal efficiency (BTE) were measure at full load and constant engine speed of 1500 rpm. The BSFC and the percentage in the BSFC are listed in Table 3. As seen in the Table 3 the soybean biodiesel blends have higher BSFCs and lower BTE than the No.2 diesel fuel. This is due to the lower heating values of the methyl esters that are about 12.4% less than for No.2 diesel fuel. Therefore, it is necessary to increase the fuel quantity to be injected into the combustion chamber in order to produce the same power. These results are similar to those predicted by Diesel-rk simulation software. Also they are in good agreement with the results obtained by other researchers [5], [9].

Table 3 Average values and % changes in BSFC and BTE

Fuel type	BSFC (kg/kW.h)	% change in BSFC	BTE (%)	% change in BTE
No.2 diesel	0.264	-	32.09	-
20 % SME	0.275	4.2	31.25	-2.61
40 % SME	0.287	8.7	30.5	-4.95
SME	0.302	14.65	29.5	-8.07

5.3 Emission analysis

The exhaust emissions were compared for SME, 20% SME blend, 40% SME blend and No.2 diesel fuel at full load condition (4.4 kW brake power). Exhaust emissions measured were UHC, CO, NOx, and the smoke opacity.

Unburned HC emissions: the UHC exhaust emissions are shown in Fig. 6. For the methyl ester and its blends, the UHC emissions were less than for the No. 2 diesel fuel because of the better combustion of the biodiesel inside the combustion chamber due to the availability of excess content of oxygen in the SME blends as compared to pure diesel fuel. The highest UHC reduction was found for SME. It is found that 15%, 27% and 38.4% reduction in the UHC emission obtained with 20% SME blend, 40% SME blend and SME biodiesel respectively, compared to No.2 diesel fuel. Same observation noticed with the results obtained by [15].

Carbon monoxide the CO emissions occur due to the incomplete combustion of fuel. The comparative analysis is shown in Fig. 7. All blends of SME are found to emit significantly lower CO concentration compared with that of No.2 diesel fuel over the entire load. When the percentage of blend of biodiesel increases, CO emission decreases. The excess amount of oxygen content of biodiesel results in complete combustion of the fuel and supplies the necessary oxygen to convert CO to CO_2. It is obtained that 11.36%, 29% and 41.7% reduction in the CO emission with 20% SME blend, 40% SME blend and SME respectively.

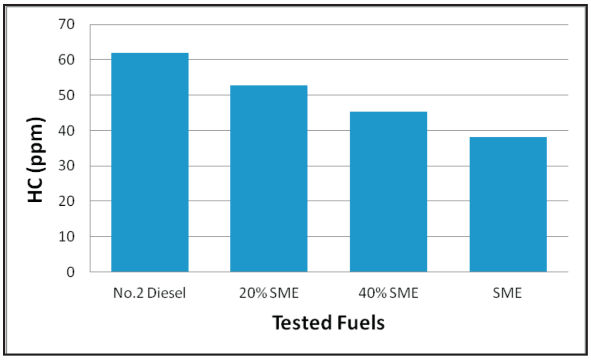

Fig. 6. Comparison of hydrocarbon emissions

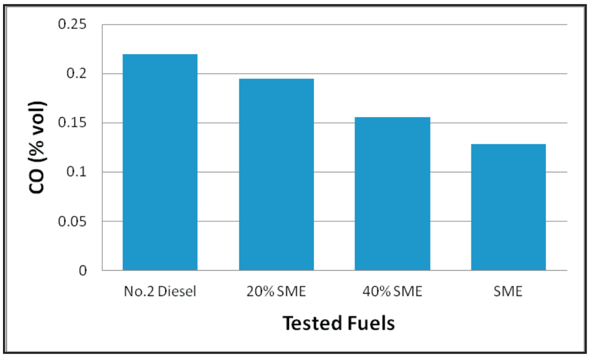

Fig. 7. Comparison of carbon monoxide emissions.

NOx emissions: Three conditions which favour NOx formation are: higher combustion temperature, more oxygen content and faster reaction rate [22]. The above conditions are attained in biodiesel combustion very rapidly as compared to diesel fuel. Hence, NOx formations for biodiesel blends are always greater than diesel fuel. The increase in the NOx emissions may be associated with the oxygen content of the methyl ester, since the fuel oxygen may provide additional oxygen for NOx formation and also the difference in the compressibility of the tested fuels can cause early injection timing and produce higher NOx emissions [23]. As shown in Fig.8 the NOx emissions of the 20% SME blend increases by 7.5% compared to No.2 diesel fuel.

Smoke opacity*:* Smoke opacity means the degree to which the smoke reduces the passage of light. It means more smoke in the exhaust will have high smoke opacity and vice-versa. As shown in Fig. 9 for the methyl ester and the blends smoke opacity were less than for the No.2 diesel fuel. It is observed that smoke opacity of diesel and SME were lower at low load, but increased at higher engine loads because more fuel is injected at higher load so less oxygen will be available for the reaction. The average smoke opacity for 20% SME blend, B40% SME blend, and SME were less than that of No.2 diesel fuel by 20.5%, 33.41%, and 48.23% respectively. This is because smoke decreases with high oxygen content in the biodiesel that contributes to complete fuel oxidation even in locally rich zones, so the oxygen within the fuel decreases the tendency of a fuel to produce soot [24]**.** Another reason for smoke reduction when using biodiesel is the lower carbon to hydrogen (C/H) ratio as compared to pure diesel fuel. This is also indicated in the study of [25].

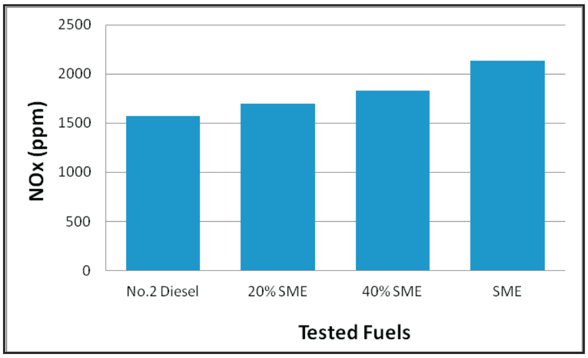

Fig.8. Comparison of NOx emissions

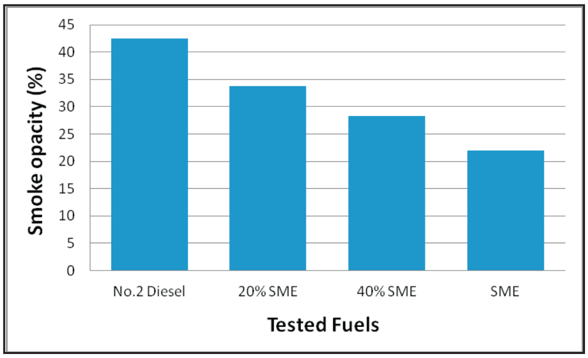

Fig.9. Comparison of smoke emissions

6. Comparison of Results

In this section the point of discussion is to compare some of the results obtained from experimental investigation with the simulation of Diesel-rk software. A slight difference between the two is recorded. In order to predict the values very near to the experimental results, a separate code has to be incorporated to put all the experimental constraints and loses by using the user defined functions in the software. The results are compared at full load with standard compression ratio of 17.5. The difference between these results is reported and arranged in Table 4.

7. Conclusions

The peak cylinder pressure was observed to be closer to TDC when blending of SME increased. All blends of SME had earlier start of combustion as compared to No.2 diesel fuel due to the shorter ignition delay which is affected by cetane number. The blends of SME are found to give nearly same brake thermal efficiency compared to No.2 diesel fuel. The BSFCs for SME and its blends were higher than for diesel fuel. The increase in the BSFC was 14.65% for neat SME. Significant reductions in the UHC, CO and smoke opacity compared with No.2 diesel fuel. The measured NOx emissions are higher than that of diesel fuel for all blends of SME. The best blending ratio is 20% SME which gives the same performance compared to No.2 diesel fuel and less increase in the NOx emissions as compared with other SME blends.

Table 4 Comparison of experimental results with Diesel-rk software results for B100 SME Biodiesel

% change in parameter	Range of SME substitution %	Experimental results (%)	Diesel-rk software results (%)	Δ%
Peak pressure	From 0 to 100	-2.98	-2.5	0.59
BTE	From 0 to 100	-8.07	-7	1.07
BSFC	From 0 to 100	14.65	17.31	2.66
Smoke	From 0 to 100	-48.31	-52.96	4.64
NOx	From 0 to 100	35.10	38.43	3.33

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