2018-01-0915

Presented in WCX SAE World Congress, Detroit, USA held on 10-12 April 2018

PERFORMANCE AND EMISSION CHARACTERISTICS OF A DIESEL ENGINE FUELED WITH DIESEL AND ORANGE OIL BLENDS USING DIFFERENT BOWL-IN PISTON GEOMETRIES

Niklesh Reddy P, Santosh Kumar Paruvada, and Abhijeet Killol NIT Rourkela
Geetha Murugan B.S. Abdur Rahman Crescent Institute of Technology, Chennai, India
Khayum Naseem and Murugan Sivalingam NIT Rourkela

Abstract

Biomass derived fuels have a potential to replace fossil fuels that are used in transportation. Orange oil is considered as one of the possible alternative biofuels for compression ignition (CI) engines, as it is renewable and available in a significant quantity throughout the world. It is a biomass derived fuel obtained from orange skin, which has 90% D-limonene. In this study an orange oil diesel blend is used as a test fuel in a single cylinder, four stroke, direct injection (DI) diesel engine with a power output of 4.40kW at a rated speed of 1500 rpm to assess the performance and emission parameters of the engine, when the engine is run with three different piston geometries. The experimental results of the performance and emission parameters of the engine for the orange oil operation were compared with those of the diesel operation of the same engine and presented in this paper.

Introduction

Achieving higher engine efficiency and utilizing the renewable fuels to a maximum possible extent are the two methods that are mainly focused in many countries for mitigating greenhouse gas (GHG) emissions. Although, biodiesel and alcohols are currently utilized as alternative transportation fuels in many developed and some of the developing countries, exploration of different sources for production of these two biofuels and utilizing them as transportation fuels is still continuing. Apart from utilization of these biofuels, other biofuels such as orange oil and turpentine oil have recently been explored for their suitability as substitute fuels for compression ignition (CI) engines. Orange oil was tested for its suitability as an alternative fuel for CI engines by Purushothaman and Nagarajan **[1-2].** A series of tests in the form of sole orange oil, different percentages of orange oil diesel fuel blends, orange oil diethyl ether dual fuel mode were performed in a single cylinder, four stroke, air cooled, direct injection (DI), diesel engine developing power of 4.40kW at a rated speed of 1500rpm. It was reported that the efficiency increased with increase in percentage of orange oil in the blend. CO and HC emissions reduced for orange oil compared that of diesel operation at the entire operation. Smoke emission reduced marginally for orange oil than the diesel fuel, while NO emission increased at full load. When DEE was used as an ignition improver and inducted in suction at low quantities with orange oil as a pilot fuel, HC and CO emissions increased, while NO emission decreased at full load. It is suggested that there must be a compromise between the fuel efficiency, NO and smoke emissions. There are more research works to be carried out to establish the possibility of using orange oil in CI engines.

Engine modifications affect the engine performance **[3-4].** Combustion and emissions were greatly influenced by the swirl ratio, and which makes the distribution of fuel/air equivalence ratio and the flow fields also accordingly change in the combustion chamber. Before spray, the squish primarily controls the turbulent kinetic energy distribution, while after spray; the combustion reverse squish and the swirl have a great impact on the in-cylinder temperature. Effect of swirl with turbulence induced on the piston surface showed better results and reduced emissions. A simulation work was carried out to study the effects of the piston bowl geometry on the combustion and emission characteristics of a diesel engine fueled with biodiesel which was produced from waste cooking oil with a major composition of palm oil under medium load conditions **[5].** Three different bowl geometries namely: the baseline omega combustion chamber (OCC), hemispherical combustion chamber (HCC) and shallow depth combustion chamber (SCC) were considered with same compression ratio of 18.50. The simulation results indicated that in terms of performance, the SCC was found to be favorable at low engine speed as it had high indicated power compared to that of OCC and HCC, while for this geometry the NO emission was found to be higher due to high temperature and pressure during the combustion process. At high engine speeds, OCC was preferred as its superiority in combustion chamber shape in forming strong squish in a short time which was better than HCC and SCC. The results also indicated that concentration of CO was found to be low due to a well-mixed mixture. A typical study was carried out to **[6]** optimize the combustion bowl geometry of a single cylinder stationary diesel engine for the effective operation of diesel-KME (kapok methyl ester) blends. For the study, in addition to the ~~**convention**~~ conventional design of HCC (hemispherical combustion chamber) two different combustion chamber geometries such as TCC (toroidal combustion chamber) and TRCC (trapezoidal combustion chamber) were chosen. In the experimental investigation, as mentioned above suitable blends such as B25 (25% KME + 75% diesel), B50 (50% KME + 50% diesel), B75 (75% KME + 25% diesel) and B100 (100% KME) were tested in a diesel engine with various combustion chamber geometries. From the experimental investigation, TCC (toroidal combustion chamber) was found to exhibit better emission, combustion and performance than other combustion chamber geometries for all test blends.

Some researchers have modified the piston geometry for improving turbulence. For instance, three different researchers have assessed the engine performance and emissions of a single cylinder, four stroke, air cooled, constant speed DI diesel engine when the engine was run on tested Jatropha methyl ester (JME) **[7]**, tyre derived fuel diesel dimethyl carbonate blend **[8]**, and carbon black water diesel emulsion **[9]**. Two micro drilled holes were oppositely provided to induce more turbulence to the engine. By providing internal jets, the engine performance was significantly improved with reduced UHC and carbon monoxide (CO) emissions in the entire engine operation.
This investigation is aimed to study the effect of operating a single cylinder, four stroke, air cooled DI diesel engine with three different nature of pistons; (a) bowl-in piston, (b) piston with internal jets (micro holes made), and (c) spiral groove made on internal jet bowl-in piston, when engine was run on an orange oil diesel blend comprising. The results are presented in this paper.

Materials and methods

Orange oil

Orange oil is produced by cells within the peel of an orange fruit. D-Limonene ($C_{10}H_{16}$) is the major component of orange oil. Unlike plant oils, orange oil is extracted as a by-product of orange juice production by centrifugation producing cold-pressed oil and even it can be extracted from the rind of citrus fruits at normal temperature by steam distillation process ~~**orange oil can be extracted**~~. The molecular structure of D-Limonene is shown in Fig. 1. The chemical composition of orange oil is given in Table 1. Some of the properties of orange oil are compared with diesel and shown in Table 2.

Table 1 Chemical composition of orange oil

Components	**Percentage of component**
Moisture	Nil
Mineral water	Nil
Carbon	82.97
Hydrogen	12.63
Oxygen	0.21
Sulphur	0.006
Nitrogen	3.11

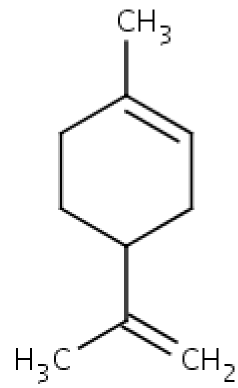


Fig. 1 Chemical structure of orange oil

Table 2 Comparison between diesel and orange oil

Property	**Diesel**	**Orange oil**
Density (Dt) at 30ºC, kg/l	0.83	0.82
Calorific value (CV), kJ/kg	43000	34650
Viscosity (Vs) at 40ºC, cSt	2.70	3.52
Fire point (FrP), ºC by PMCC method	65	82
Flashpoint (FlP), ºC by PMCC method	52	74
Cetane number (CN)	49	47
Latent heat of vaporization (LHV) at 15ºC, kJ/kg	232.60	325.64

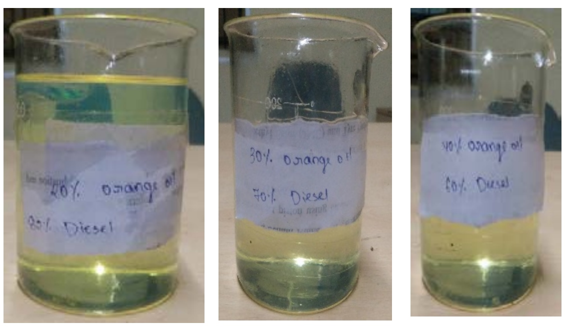
In this study initially experiments were conducted in the test engine with five different orange oil diesel (O-D) blends. The designation and the composition of the test fuels used in this study are given in Table 3.

Table 3 Composition of the test fuels

Test Fuel	**Diesel (%)**	**Orange Oil (%)**
Diesel	100	0
10% O-D	90	10
20% O-D	80	20
30% O-D	70	30
40% O-D	60	40
50% O-D	50	50
Orange oil	0	100

The different orange oil blends, orange oil and diesel fuel are shown in Fig. 2(a) to Fig. 2(f). All the orange oil diesel blends were kept in an observation for 15 days to ensure miscibility. There was no separation took place in the blends. The important physicochemical properties of all the test fuels used in this study are given in Table 4.

(a) Diesel (b) Orange oil (c) 10% O-D



(d) 20% O-D (e) 30% O-D (f) 40% O-D

Fig. 2 Test fuels used in this investigation

Table 4 Important physicochemical properties

FP	Dl	O-O	10% O-D	20% O-D	30% O-D	40% O-D	50% O-D
Dt	0.828	0.817	0.827	0.826	0.825	0.824	0.823
CV	43000	34650	42165	41330	40495	39660	38825
Vs	2.70	3.52	2.78	2.86	2.95	3.03	3.11
FrP	65	82	66.70	68.40	70.10	71.80	73.50
FlP	52	74	54.20	56.40	58.60	60.80	63
CN	49	47	48.80	48.60	48.40	48.20	48
LHV	232.60	325.64	241.90	251.20	260.50	269.80	279.10

Engine experimental setup

The schematic diagram of the experimental procedure of the investigation engine is shown in Fig 3. Experiments were carried out in a single cylinder, air cooled, four stroke direct injection (DI) compression ignition (CI) engine with a power of 4.40kW at 1500 rpm. The technical specifications of the test engine are given in Table 5. Two different fuel tanks A and B were used in this study to fill diesel fuel and orange oil based fuel. A fuel level indicator was used to measure the total fuel consumption, and a K-type thermo couple was installed for the measurement of exhaust gas temperature. An electrical resistance load bank was used to provide engine loading through an alternator.

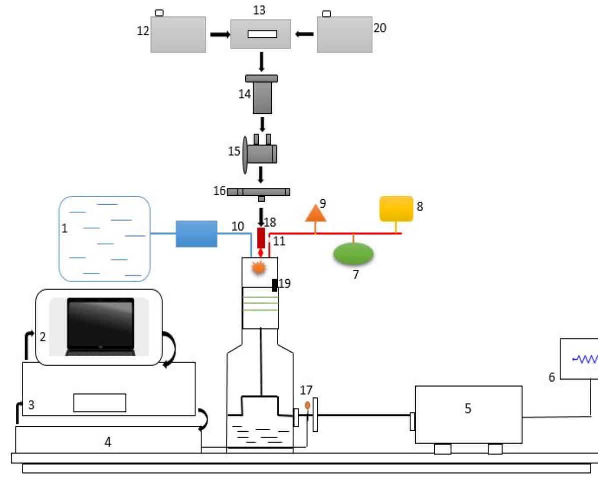


Fig. 3 Schematic Diagram of Experimental setup

1. Intake air box	2.Computer	3. DAS
4.Control panel	5.Dynamometer	6.Load bank
7.Smoke meter	8.Gas analyzer	9.Exhaust gas temperature sensor
10.Intake manifold	11.Exhaust manifold	12.Fuel tank A
13.Fuel meter	14.Fuel filter	15.Fuel pump
16.Fuel line	17.Crank angle encoder	18.Fuel injector
19.Pressure transducer	20. Fuel tank B	

Table 5 Engine Specifications

Make/model	Kirloskar TAF1
Brake power (kW)	4.40
Rated speed (rpm)	1500
Stroke (mm)	110
Bore (mm)	87.50
Compression ratio	17.50:1
Piston type	Bowl in piston
Injection timing (CA)	23 BTDC
Nozzle opening pressure (bar)	200
Injection type	Pump line nozzle injection system
No. of holes	3
Nozzle type	Multi hole

A data acquisition system (DAS), in **~~combined~~** combination with a piezo-electric pressure transducer and crank angle encoder was used for the measurement of cylinder pressure. In all the cases pressure-crank angle diagrams were recorded and processed to get the required combustion parameters by the data given system. The exhaust emissions coming from the engine was measured by an AVL DiGas444 exhaust gas analyzer. Initially, experiments were conducted in the test engine using diesel fuel to obtain the reference data. Further, the test engine was tested with four different orange oil diesel blends.

Turbulence in combustion chamber

This experimental investigation is focused on increasing the air swirl and turbulence in combustion chamber. There have been many methods employed to induce the swirl ratio or turbulence in the combustion chamber namely, mounting different headed pistons, providing different combustion chambers, increasing nozzle pressure [9]. Internal jet bowl-in (IB) piston and spiral grooved internal jet bowl-in (SGIB) piston are the examples of variable head of the piston. The change in geometry of the piston may help to form complete mixing of fuel air mixture during combustion process. Further SGIB piston is made to increase the swirl motion further in a uniform way by making grooves on the surface of the bowl which helps to increase the combustion further. The kinetic energy of air increased further, by the grooves made on the piston bowl rotates the air in a helical motion within the combustion chamber which helps in further increase in turbulence of air, this result in the complete mixing of air fuel mixture followed by complete combustion of charge.

Pistons used in this study

In this experimental study, three different pistons were used. The first one is bowl-in piston which is already fitted in the engine. Fig. 4(a) and 4(b) show two views of bowl-in piston used in this study. The diameter of the bowl in the piston is 52mm and depth is 26mm. The second one is a bowl-in piston provided with four micro drilled holes (internal jets). The third one is a bowl-in piston provided with a spiral grooved internal jet bowl-in piston.

Fig. 4(a) Front view of the bowl in piston

Fig. 4(b) Top view of the bowl in piston

The developments of the second and third type pistons are described below which were made from bowl-in piston geometry;

Internal jet bowl-in piston

Fig. 5(a) shows the arrangement of internal jets provided in the bowl-in piston and Fig. 5(b) illustrates the photographic view of the internal jet bowl-in (IB) piston used in this study.

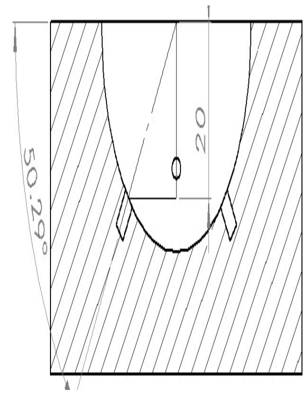


Fig. 5(a) Line diagram of internal jets

Fig. 5(b) Photographic view of internal jets

Four micro drilled holes were made at the bottom of the bowl. By making holes the turbulence of the air may get increased thereby turbulence kinematic energy of the air increases which helps for the complete combustion of air-fuel mixture **[9]**. They were symmetrically located as shown in the figure. The diameter of the hole is 3mm and depth is 5mm in each hole. The holes are at a depth of 20mm and at an angle of 50.29^{o} from the top of the piston surface.

Spiral grooved internal jet bowl-in piston

Fig. 6(a) shows the line diagram of spiral grooved internal jet bowl-in piston **~~made in the piston~~** and Fig. 6(b) illustrates the photographic view of the spiral grooved internal jet bowl-in (SGIB) piston used in this study.

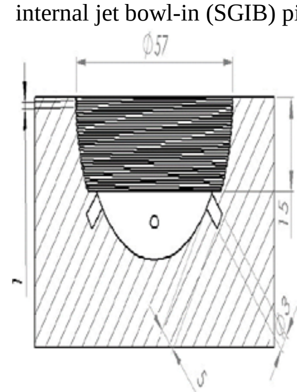


Fig. 6(a) Line diagram of SGIB

Fig. 6(b) Photographic view of SGIB

The spiral groove was made with the help of a CNC machine with the help of end-mill tool. In this type, diameter of the bowl in the piston was increased up to 57mm. The grooves made on the piston were 0.50mm deep into the piston and gradient between each groove is 1mm. The grooves were made up to a depth of 15mm from the top surface of the piston.

Error analysis

The details of instruments used in the present investigation are provided in Table 6. The comprehensive uncertainty of the experiment was calculated by the inclusion of the uncertainties of the respective apparatus, it is given below.

Total uncertainty of the experiment = square root of [uncertainty of $\{(TFC)^2 + (\text{brake power})^2 + (\text{specific fuel consumption})^2 + (CO)^2 + (HC)^2 + (NO)^2 + (\text{pressure transducer})^2 + (EGT)^2\}$]

= square root of $\{(1.50)^2 + (0.20)^2 + (1)^2 + (1)^2 + (0.20)^2 + (0.20)^2 + (0.20)^2 + (0.05)^2 + (1)^2 + (0.15)^2\}$

= ±2.33

Using the calculation procedure, the total uncertainty for the entire experimentation is found to be ±2.33

Table 6 Ranges, accuracy and uncertainty of the instruments

Sl.no.	Instrument	Range	Accuracy	Uncertainty
1	Temperature indicator	0-900	±0.1ºC	±0.15
2	Load indicator	250-6000W	±10W	±0.20
3	Burette	1-30cc	±0.20cc	±1
4	Exhaust gas analyzer	NO 0-5000ppm HC 0-20000-ppm CO 0-10%	±12ppm ±12ppm 0.06%	±0.20 ±0.20 ±0.20
5	Pressure transducer	0-110bar	±0.10bar	±0.05
6	Speed sensor	0-10000rpm	±10 rpm	±0.10
7	Crank angle encoder	0-720	±1º	±0.20

~~**A data acquisition system (DAS), in combined with a piezo-electric pressure transducer and crank angle encoder was used for the measurement of cylinder pressure. In all the cases pressure-crank angle diagrams were recorded and processed to get the required combustion parameters by the data given system. The exhaust emissions coming from the engine was measured by an AVL DiGas444 exhaust gas analyzer. Initially, experiments were conducted in the test engine using diesel fuel to obtain the reference data. Further, the test engine was tested with four different orange oil diesel blends.**~~

Results and Discussion

Orange oil diesel blends Performance and emissions

Brake thermal efficiency (BTE)

Fig.7 portrays the variation in brake thermal efficiency (BTE) with brake power for five different O-D blends. The BTE of an engine depends on fuel air mixture formation, heating value of fuel, and combustion nature. It can be observed from the figure that, the diesel operation exhibits the lowest values among all the fuels tested in this study. The brake thermal efficiency of the orange oil diesel (O-D) blends increases as the orange oil content in the blend increases compared to that of diesel **[1-2]**. The brake thermal efficiency (BTE) is the highest for orange oil compared to that of other blends and diesel at full load condition. As per ASTM standards, the prescribed viscosity value for diesel ranges between 2-4 cSt at 40ºC. The viscosity of orange oil lies within this range. The oxygen content in orange oil results in more complete combustion. Hence, the thermal efficiency of orange oil and its diesel blends are higher than that of diesel operation at full load.

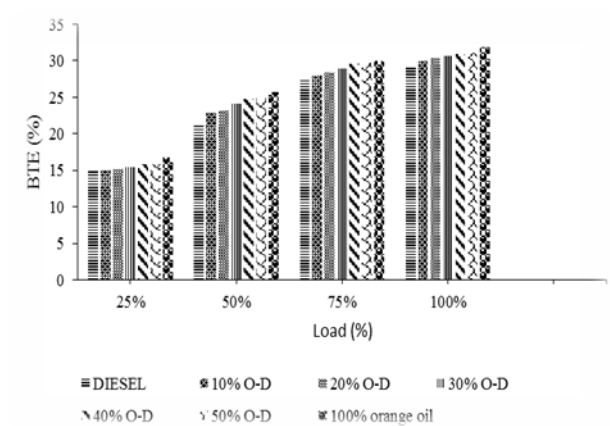


Fig. 7 Brake thermal efficiency with load

For orange oil, the brake thermal efficiency value varies from 16.75% to 31.89% from 25% load to 100% load respectively. Similarly, BTE for diesel and other blends like 10% O-D, 20% O-D, 30% O-D, 40% O-D, and 50% O-D BTE varies from 14.92% to 29.20%, 15.01% to 29.93%, 15.23% to 30.35%, 15.40% to 30.61%, 15.79% to 30.95%, and 15.86% to 31.12% from 25% load to 100% load respectively.

Brake specific energy consumption (BSEC)

The variation of BSEC for bowl-in piston with brake power for different fuels tested in this study is depicted in Fig. 8. The brake specific fuel consumption is not a reliable parameter, as the density and calorific value of different blends varies marginally, so brake specific energy consumption (BSEC) is considered. It can be observed from the figure that the BSEC decreases with increasing the brake power for diesel and other blends. This is owing to higher heat energy prevailing inside the combustion chamber that reduces fuel consumption. The BSEC is higher for diesel which varies from 24.21 MJ/kW-hr to 12.81 MJ/kW-hr from 25% load to 100% load condition.

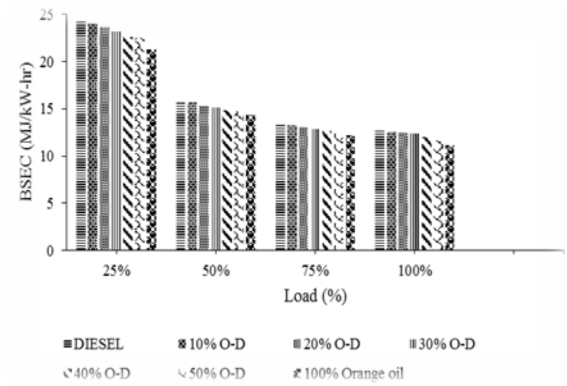


Fig.8 BSEC of bowl-in piston

Similarly, BSEC for orange oil, 50% O-D, 40% O-D, 30% O-D, 20% O-D and 10% O-D varies from 21.26 MJ/kW-hr to 11.12 MJ/kW-hr, 22.49 MJ/kW-hr to 11.63 MJ/kW-hr, 22.61 MJ/kW-hr to 11.96 MJ/kW-hr, 23.18 MJ/kW-hr to 12.37 MJ/kW-hr, 23.69 MJ/kW-hr to 12.51 MJ/kW-hr and 24.06 MJ/kW-hr to 12.60 MJ/kW-hr from 25% load to 100% load condition. It can be observed that BSEC decreases by increasing the brake power for diesel and O-D blends. This may be due to more fuel injected for producing the same power output BSEC decreases when the percentage of orange oil increases. This is due to better combustion of orange oil when compared to that of diesel.

Exhaust gas temperature (EGT)

The exhaust gas temperature (EGT) is a parameter to determine the efficiency of the combustion. The exhaust temperature gives an idea about the amount of energy going waste.

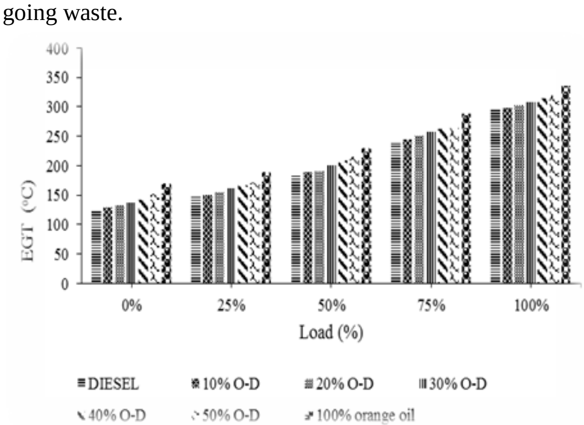


Fig. 9 EGT of bowl-in piston

Fig. 9 portrays the variation of exhaust temperature with brake power for the fuels tested in this study. It can be observed that the EGT increases with increase in brake power as a result of increase in fuel consumption that takes part in combustion. It can be observed from the figure that diesel has the lowest EGT value of 295^{o}C compared to all O-D blends at full load condition. This is may be due to lower viscosity and higher cetane number of diesel relatively to that of all O-D blends and orange oil. The EGT of 10% O-D, 20% O-D, 30% O-D, 40% O-D, 50% O-D, orange oil is found to be 298^{o}C, 304^{o}C, 308^{o}C, 315^{o}C, 319^{o}C, 335^{o}C at full load condition. It can also be observed that EGT is increases with increase in orange oil percentage. ~~**This may be due to longer ignition delay of orange oil as a result of lower cetane number and higher viscosity.**~~ Since the orange oil diesel blends have lower cetane numbers, the ignition delay of them would be longer. As a result, relatively more fuel air mixture might burn in the later part of the expansion stroke. This may be the reason for higher exhaust gas temperatures found with the orange oil diesel blends. Advancing the injection timing may reduce the delay period and exhaust gas temperature.

CO emission

The variation of CO emission for diesel and orange oil blends for different engine loads is depicted in Fig. 10. Generally CI engines operated with lean mixture. Therefore, the CO emission is found to be ~~**lesser**~~ lower than the SI engine. Carbon combines with oxygen to form CO_2 during complete combustion, in case of incomplete combustion due to lack of oxygen results in CO formation **[9]**. The CO emission is found to be highest for diesel compared to the other blends with concentration of 0.396 g/kW-hr at full load condition. The concentrations of CO emission for the blends 10% O-D, 20% O-D, 30% O-D, 40% O-D, 50% O-D and orange oil were found to be 0.366 g/kW-hr, 0.348 g/kW-hr, 0.306 g/kW-hr, 0.284 g/kW-hr, 0.272 g/kW-hr, 0.258 g/kW-hr at full load condition respectively. Orange oil contains about 3.2% of oxygen that may result in more complete combustion of carbon, results in the lower CO emission values **[10].**

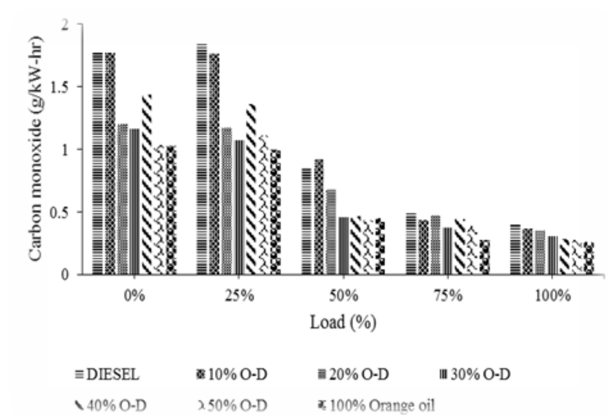


Fig.10 CO emission of bowl-in piston

The decrease in CO emission is due to increase in the orange oil percentage in the blend, thereby percentage of oxygen in the O-D blends increases which results in more complete combustion of air fuel mixture.

HC emission

The variation of HC emission for orange oil diesel blends and diesel is depicted in the Fig 11. ~~**The increase in HC emission**~~ HC emission increases due to incomplete combustion because of improper mixing of charge **[9].** The HC emission concentration decreases with increasing brake power **[2].** The concentration of HC emission is the highest for diesel than the other orange oil blends with an emission concentration of 0.128 g/kW-hr at full load condition. This may be due to the poor combustion of diesel compared to that of other blends used in this study.

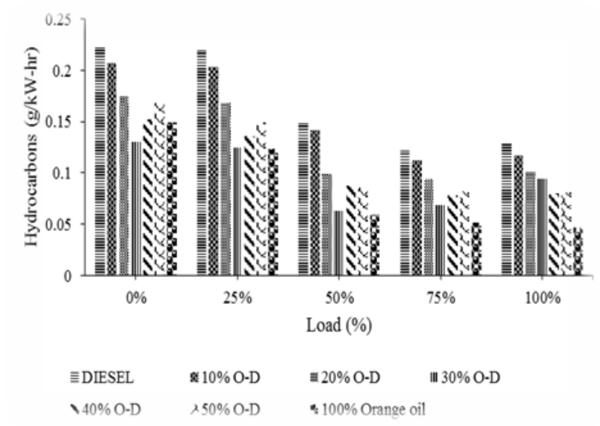


Fig. 11 HC emissions of bowl-in piston

The increase in percentage of orange oil involves in the formation of rich mixture which leads to more combustion rate resulting in the higher heat release rate. The concentration of others blends 10% O-D, 20% O-D, 30% O-D, 40% O-D, 50% O-D blends and orange oil are found to be 0.117 g/kW-hr, 0.101 g/kW-hr, 0.094 g/kW-hr, 0.080 g/kW-hr, 0.081 g/kW-hr and 0.047 g/kW-hr at full load condition respectively. The HC concentration of orange oil is less when compared to that of diesel throughout the engine load, because of complete combustion of air fuel mixture **[2].** As the ignition delay period lengthens, for example, due to a reduction in the fuel cetane number, some part of the mixture may become over-mixed with air **[11]**. Due to this, HC emission is decreased for orange oil.

Nitric oxide emission

Fig. 12 depicts the concentration of oxides of nitrogen (NO) from exhaust at different loads for bowl-in piston.

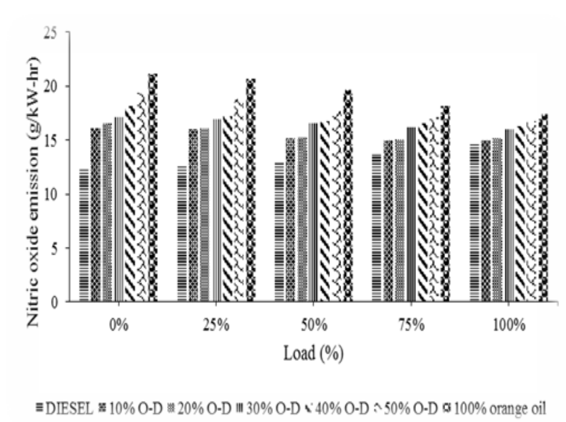


Fig. 12 NO emissions of bowl-in piston

The oxides of nitrogen in a CI engine is mainly formed due to oxygen available for combustion and higher peak cylinder temperature **[12].** Diesel exhibits lowest NO emission trend throughout the engine operation. Due to its higher cetane number and oxygen free molecules, the NO emission is lower for diesel operation than the other fuels in entire engine load. The NO is found to be higher for 100% orange oil compared to the other blends and diesel with an emission of 17.36 g/kW-hr at full load. As the orange oil contains more amount of oxygen, the NO is higher when there is an increase of orange oil percentage in the blend **[13].** This may be due to the higher intensity of heat release rate in the premixed combustion phase and quick burning for more percentage of orange oil in the blend **[2].** The emissions of 10% O-D, 20% O-D, 30% O-D, 40% O-D, 50% O-D and diesel are found to be 14.95 g/kW-hr, 14.70 g/kW-hr, 15.96 g/kW-hr, 16.25 g/kW-hr, 16.76 g/kW-hr and 14.70 g/kW-hr respectively at full load condition.

Smoke

Smoke forms due to incomplete combustion of air fuel mixture (incomplete combustion of hydrocarbon and partial combustion of carbons in the liquid fuel), and it formed in rich mixture zone of combustion chamber **[14]**. Fig.13 portrays the variation of smoke percentage with brake power for the fuels tested in this study. Smoke percentage increases with increase in load as a result of increase in fuel consumption. It can be observed that up-to 40% orange oil blend operation, the smoke values are ~~**lesser**~~ lower compared to those of diesel operation in the entire engine operation. This may also be due to low C/H ratio of the fuel. The smoke emission for 50% and 100% are higher than that of diesel values throughout the load.

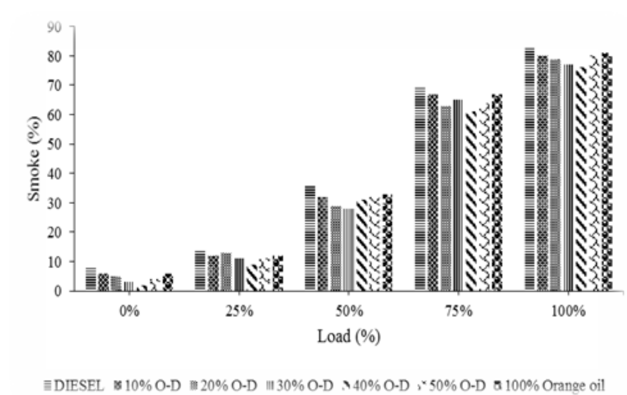


Fig. 13 Smoke percentage of bowl-in piston using diesel and O-D blends

This may be due to dominance of fuel viscosity than the oxygen present in the fuel. The smoke percentage of diesel, 10% O-D, 20% O-D, 30% O-D, 40% O-D, 50% O-D and orange oil is found to be 83%, 80%, 79%, 77%, 76%, 80% and 81% at full load condition respectively. It was observed that, increase in the percentage of orange oil in diesel results in decrease in the smoke percentage. This is due to increase in oxygen content in the O-D blends which results in complete combustion of charge **[12].**

Optimum orange oil diesel blend with different pistons

~~**From the above experimental results, it is found that 40% O-D exhibits higher thermal efficiency and lower smoke emissions, than that of diesel operation at full load. Therefore, experiments were further conducted with 40% O-D when the engine piston was provided with more turbulence.**~~

From the above experimental results, it is found that 40% O-D blend exhibits higher efficiency than 30%, 20%, 10% O-D blends and diesel operation at full load condition. CO, HC, smoke for 40% O-D blend has lower percentage compared to that of 30%, 20%, 10% O-D blends and diesel. Similarly, 40% O-D has less percentage of NO emission when compared to that of 50% O-D blend and pure orange oil. Therefore, experiments were further conducted with 40% O-D when the engine piston was provided with more turbulence.

BTE

The variation of brake thermal efficiency with brake power for different piston geometries when the engine was run on 40% O-D depicted in the Fig. 14. The brake thermal efficiency of different piston geometries with 40% O-D increases with increase in load as expected. The brake thermal efficiency of IB and SGIB pistons for 40% O-D varies from 15.96% to 31.21% and 16.87% to 31.94% from low load to full load conditions respectively. The BTE of 40% O-D of all pistons is higher than that of diesel in the entire engine operation. It can also be observed that BTE of IB piston is increased by about 0.85% at full load when compared to bowl-in piston. This is due to the increase of turbulence by introducing holes in the combustion chamber (internal jet piston). As the air swirl increases, the velocity of air may increase and improve more spray formation resulting in more complete combustion. The brake thermal efficiency is further increased by about 2.29% compared to IB and bowl-in piston by 3.09% respectively at full load when the engine was made to run with SGIB.

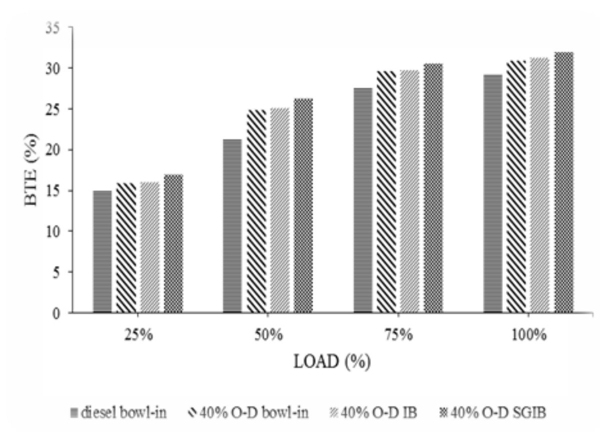


Fig. 14 BTE of different pistons with 40% O-D

This may be due to both turbulence with more swirl of air created in the combustion chamber which results in a good mixing of air and fuel in the combustion chamber leading to a higher rate of combustion.

BSEC

The variation of BSEC with brake power for different piston geometries is illustrated in Fig.15.

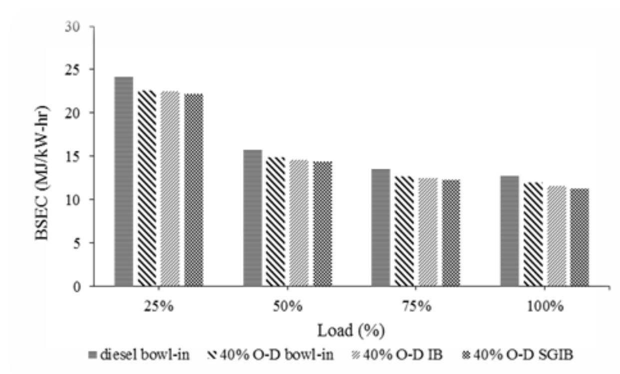


Fig. 15 BSEC of different pistons with 40% O-D

The BSEC of different piston geometries decreases with increase in brake power, due to less amount of fuel is injected to produce the required power output. The BSEC of IB and SGIB pistons for 40% O-D varies from 22.50 MJ/kW-hr to 11.57 MJ/kW-hr and 22.19 MJ/kW-hr to 11.23 MJ/kW-hr from low load to full load condition. The BSEC of all the pistons with 40% O-D is found to be ~~**lesser**~~ lower in comparison with diesel operation in the entire engine operation. At a given load, the BSEC decreases in the order of diesel, 40% O-D bowl-in, 40% O-D IB, 40% O-D SGIB. The data implies that BSEC also decreases for IB piston, when compared to bowl-in piston. This is due to increase in air turbulence which results in more mixing of air and fuel which results in better combustion to give the same power output using less amount of fuel. The BSEC is also decreased for SGIB piston by 2.93% and 6.10% when compared to IB and bowl-in pistons respectively. The decrease in BSEC for SGIB piston may be due to further increase in turbulence and swirl motion in the cylinder. As a result heat prevailing inside the cylinder may be spread uniformly by tiny droplets of fuels reducing fuel consumption.

Exhaust gas temperature

Fig. 16 depicts the variation of EGT for different piston geometries with varying load. The EGT of different piston geometries increases with increase in brake power. At a given load, EGT follows increasing trend in the order of diesel bowl-in, 40% O-D SGIB, 40% O-D IB, 40% O-D bowl-in. The value of EGT for 40% O-D of IB and SGIB pistons are 309^{o}C and 301^{o}C at full load condition. By providing internal jets more turbulence is offered and hence the EGT is marginally lower for IB operation in the entire load spectrum. Further, provision of spiral grooves, the swirl is created along with the turbulence causing increased air motion in the cylinder resulting in increased thermal efficiency. This may be the reason for reduction in EGT values for SGIB operation.

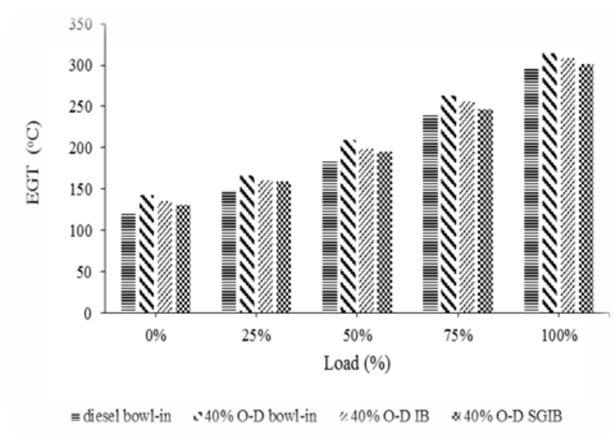


Fig. 16 EGT of different pistons with 40% O-D

However, the values of EGT are marginally higher than diesel and lower than that of 40% O-D operations.

CO emission

The CO formation depends on the oxygen availability, fuel air mixture formation and combustion duration **[15].** The variation of CO emission concentration of different piston geometries when the engine is operated at different loads is shown in Fig.17. The concentration of CO emission decreases with increase in brake power. Even at a particular load, the CO emission decreases in the order of diesel, 40% O-D bowl-in, 40% O-D IB, 40% O-D SGIB. The concentration of CO emission of IB and SGIB pistons for the 40% O-D is 0.264 g/kW-hr and 0.224 g/kW-hr at full load condition respectively. The CO is decreased further more for SGIB piston when compared to that of IB and bowl-in piston by 15.15% and 21.13% respectively at full load. With the available oxygen, turbulence and swirl motions, mixture formation may be faster resulting in lower CO emission for both IB and SGIB operations. Diesel is operated with the conventional mode; hence, higher CO emission is noticed throughout the engine operation.

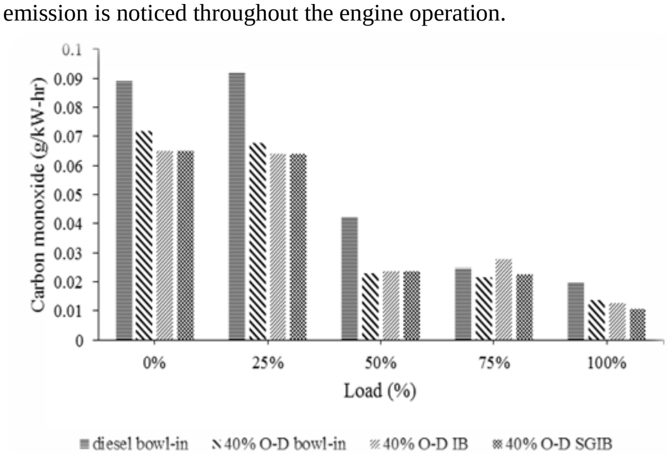


Fig. 17 CO emission of different pistons with 40% O-D

Hydrocarbon emissions

Fig. 18 depicts the concentrations of HC emission of IB and SGIB pistons at different load conditions.

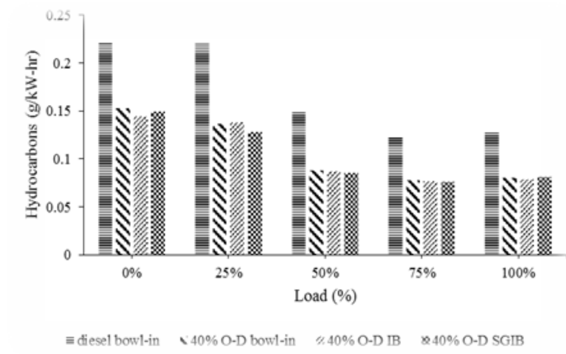


Fig. 18 HC emission of different pistons with 40% O-D

The concentration of HC emission decreases with increase in brake power. At full load, the HC emission decreases in the order of diesel, 40% O-D SGIB, 40% O-D bowl-in, 40% O-D IB. The HC concentration for IB and SGIB pistons of 40% O-D is 0.078 g/kW-hr and 0.081 g/kW-hr at full load condition respectively. The HC concentration for IB is ~~lesser~~ lower compared to that of bowl-in piston in the entire range of engine operation. Internal jets offer reduction in HC emission than that of 40% O-D operation; however provision of spiral grooves with the internal jets marginally decrease unburnt HC emission. This may be due to over mixing of fuel air mixture that takes part in combustion. The HC emission of SGIB piston is increased by 3.57% and 1.36% for IB and bowl-in respectively at full load condition.

NO emission

Fig. 19 illustrates the variation of oxides of nitrogen when the engine is made to run on 40% O-D at different loads using different piston geometries. The NO emission strongly depends upon in-cylinder temperature and amount of air supplied **[9]**. The NO emission of different piston geometries increases with increase in brake power.

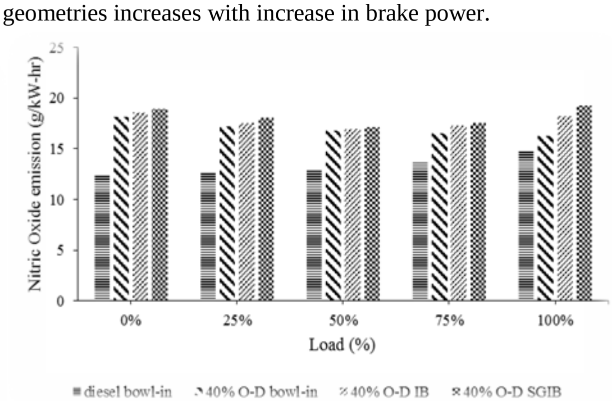


Fig. 19 NO emission of different pistons with 40% O-D

At a given load, NO emission follows increasing trend in the order of diesel, 40% O-D bowl-in, 40% O-D IB, 40% O-D SGIB. The NO concentration of IB and SGIB piston for 40% O-D is 18.19 g/kW-hr and 19.26 g/kW-hr at full load condition respectively. The NO emission of SGIB is found to be higher by 15.62% and 5.56% respectively when compared to bowl-in and IB pistons at full load condition.

As the internal jets increases the kinetic energy of the air inside the cylinder, and the collision occurs between the atoms inside the cylinder which results in increase of pressure and temperature; hence higher NO emission is notified in the entire engine operation **[16].** The increase in temperature inside the cylinder results in high formation of NO emission for SGIB piston. This increase in the NO emission may be due to increased air motion offered by the internal jets and the grooves on the piston.

Smoke emission

Fig. 20 portrays the trend of smoke percentage of IB and SGIB pistons for 40% O-D with brake power. Smoke percentage increases with increase in brake power for all pistons **~~is portrayed in Fig. 20.~~**

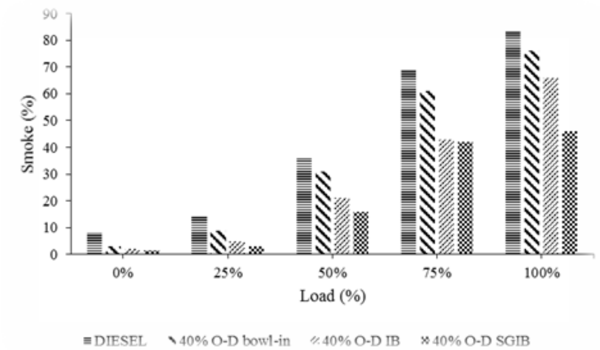


Fig. 20 Smoke percentage of different pistons for 40% O-D

At a given load, smoke percentage order followed by diesel, 40% O-D bowl-in, 40% O-D IB, 40% O-D respectively. Smoke percentage of IB and SGIB for 40% O-D is 66% and 46% at full load condition respectively. Smoke percentage of SGIB is **~~lesser~~** lower when compared to that of bowl-in piston and IB piston by 39.47% and 30.30% respectively. As a result of more complete combustion, and lower carbon molecules present in the fuel, the smoke emission is **~~lesser~~** lower for 40 % O-D which further reduced by increased air fuel mixing rates offered by the IB and SGIB operations.

Conclusion

In this experimental investigation, an analysis was made to assess the performance and emission characteristics of single cylinder, compression ignition (CI) engine using three different piston geometries namely bowl-in piston internal jet bowl-in (IB) piston and spiral grooved internal jet bowl-in (SGIB) piston fuelled with different types of orange oil-diesel blends and diesel. Using the results, the following conclusions were made.

- Among all the orange oil diesel blends and diesel used in this study, 40% O-D is the optimum blend which exhibited better BTE efficiency and **~~lesser~~** lower HC, CO and smoke emissions at full load. With increase in percentage of orange oil in the orange oil diesel blend increased the BTE with a maximum value of 31.89% at full load while for diesel and 40% O-D, the values are 29.20%and 30.61% respectively at full load.
- IB and SGIB pistons increased turbulence and swirl of air showing better results in performance and reduced emissions.
- The brake thermal efficiency of SGIB piston increased by 2.29% and 3.09% when compared to IB and bowl-in pistons when 40% O-D is used.
- Brake specific energy consumption of SGIB piston is decreased when compared to IB and bowl-in pistons by 2.93% and 6.10% when 40% O-D is used.
- The CO emission of SGIB piston decreased greatly to a maximum of 15.15% and 21.13% for IB and bowl-in piston.
- The HC emission for 40% O-D with bowl-in piston and IB are lower than that of diesel while it is marginally higher for SGIB piston at full load.
- The NO emission for all the orange oil diesel blends is higher than diesel while it is lower than orange oil in the entire load spectrum.
- SGIB piston has higher NO emission than IB and bowl-in piston by 5.56% and 15.62%
- Smoke emission decreases with increase in orange oil percentage. SGIB piston is ~~lesser~~ lower when compared to bowl-in and IB pistons by 39.47% and 30.30% respectively.

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CONTACT INFORMATION

P.NIKLESH REDDY
B-Tech Mechanical engineering,
National Institute Of Technology,
Rourkela, Odisha, India
Phone No. +91-9437720292
Email Id: niklesh95@gmail.com

Dr. Murugan Sivalingam
Associate Professor
National Institute of Technology,
Rourkela, Odisha, India
Email ID:
murugans@nitrkl.ac.in/recmurugan@yahoo.co.in

ACKNOWLEDGEMENT

Thanks to all of our parents and Dr. Murugan Sivalingam who helped us to do this project successfully.

Abbreviations

CI	Compression ignition
DI	Direct injection
GHG	Greenhouse gas
DEE	Diethyl ether
HC	Hydrocarbons
CO	Carbon monoxide
NO	Nitric oxide
OCC	Baseline omega combustion chamber
HCC	Hemispherical combustion chamber
SCC	Shallow depth combustion chamber
TCC	Toroidal combustion chamber
TRCC	Trapezoidal combustion chamber
KME	Kapok methyl ester
JME	Jatropha methyl ester
UHC	Unburnt hydrocarbons
DAS	Data acquisition system
CNC	Computer numerical control
O-D	Orange oil-diesel blend
BTE	Brake thermal efficiency
BSEC	Brake specific energy consumption
EGT	Exhaust gas temperature
IB	Internal jet bowl-in piston
SGIB	Spiral grooved internal jet bowl-in piston
ASTM	American society for testing and materials
FP	Fuel property
Dl	Diesel
O-O	Orange oil
Dt	Density at 30ºC, kg/m^3
CV	Calorific value
Vs	Viscosity at 40 ºC, cSt
FrP	Fire point, ºC by PMCC method
FlP	Flash point, ºC by PMCC method
CN	Cetane number
LHV	Latent heat of vaporization at 15ºC, kJ/kg

Alternative and Advanced Fuels
Manuscript Number: 18PFL-0078
Title: Performance and Emission Characteristics of a Diesel Engine Fueled With Diesel and Orange Oil Blends Using Different Bowl-in Piston Geometries

Author's response to the reviewer comment

The authors sincerely thank the reviewers for their valuable comments and suggestions to improve the manuscript for publication. As per their suggestions, the authors have carried out all the corrections in the revised manuscript. The sentences to be removed in the final version are shown in bold and strike through, whereas the corrections are indicated in yellow color for reviewer #207851, pink color for reviewer #207854 and green color for reviewer #217957 highlighted in the revised manuscript.

Reviewer #207851:
Comment 1: There has been quite a few of emissions studies more focus on the emissions characteristic on oxygenated fuel (DMC, DME, biodiesel, etc.) that agrees to your results on NO, CO, and HC, it would be beneficial to bring up those studies to strength your results section.
Response: As per your suggestion, necessary references are given in the revised manuscript which provide support to the results. i.e.., [10], [12], [13], [15], [16].

Comment 2: Please rewrite some of the experimental section to prevent copy and paste the same sentence in the paper, as such "A data acquisition system (DAS), in combined with a piezoelectric pressure transducer and crank angle encoder was used for the measurement of cylinder pressure. In all the cases pressure-crank angle diagrams were recorded and processed to get the required combustion parameters by the data given system. The exhaust emissions coming from the engine was measured by an AVL DiGas444 exhaust gas analyzer. Initially, experiments were conducted in the test engine using diesel fuel to obtain the reference data. Further, the test engine was tested with four different orange oil diesel blends. This Sentence has been used twice exactly the same in the paper.
Response: Authors rectified it, and the sentences repeated under Table 6 have been removed and the newly added sentence is shown in yellow color in the revised manuscript.

Comment 3: For table 4, are those value calculated from the pure diesel and orange oil properties or those are calculated based on the blend ratio.
Response: The values of pure diesel and pure orange oil were calculated in a standard laboratory and the other values were calculated according to blend ratio.

Reviewer #207854:
Comment 1: How many times have the measurements been repeated? It is not enough just listing the uncertainty of the instruments, while all other sources of uncertainty in the experiment should be included for consideration.
Response: Experiments were conducted three times for each test fuel and average of them were taken as final observation. All the uncertainties of the experiment were considered for calculations.

Comment 2: The structure of this paper is clear, however, repeated statement was found in multiple sections. For example, the paragraph below Table 6 is the same as the one after Table 5, with similar issue found in subsection "Brake specific energy consumption (BSEC)", "CO emission", and etc. Contrary statement was found in "Exhaust gas temperature (EGT)" part on the discussion of viscosity and cetane number.
Response: Authors rectified it, and the repeated sentence below Table-6 was strike through and the newly added sentence was colored. In "Exhaust gas temperature (EGT)", more appropriate answer is given now in the revised manuscript.

Comment 3: Considering the emission of CO and HC, 40% blend was not the best fuel, which is more likely to be 30%, especially when looking at the highest CO emission of 40% and the minor difference for smoke emission. Can the authors include more discussion on the choice of 40%?
Response: A biofuel can be examined for its suitability of replacing diesel fuel based on two factors, one is technical feasibility and another is availability of fuel. In this study we examined maximum replacement of diesel fuel by orange oil by conducting the experiments upto 50% orange oil and the maximum 100% orange oil. So the engine was able to run with 100% orange oil. Also considering emissions NO emission was increasing with increasing orange oil percentage in the blend while smoke was decreasing for all the blends and 100% orange oil. But however the availability of orange oil cannot completely meet the demand as it is less available in comparison with diesel fuel. In general investigation results on utilization of alternative fuels for CI engines reveal that 40% can be an optimum blend ratio. Therefore we examined with 40% Orange oil diesel blend which was tested.
Purushothaman and Nagarajan experimentally proven that 30% orange oil would be an optimum blend ratio. In our study we examined with 40% O-D blend only 2% smoke difference between 30% O-D and 40% O-D. As mentioned in the previous paragraph based on the availability orange oil it is possible to replace diesel fuel by 40% that is another reason for why we chose 40% O-D blend.

Reviewer #217957:
Comment 1: Where "lesser" is used, please replace with "lower".
Response: Corrected in the revised manuscript.

Comment 2: Figure 7: add "bowl in piston" to the title to be clear what piston was used. In the text, make it clearer that all of the data in the section is from the "bowl in piston" design.
Response: “bowl-in piston” is added to the title and it is also added to the section of piston geometries.

Comment 3: Table 6: the paragraph under the table is a duplicate of the paragraph on the previous page, 1st column, 1st paragraph. Also, change "combined" to "combination".
Response: Paragraph under Table 6 has been removed and all other uncertainties of the experiment is added in the revised manuscript. “Combined” is changed to “combination”.

Comment 4: Use a grammar check on the entire paper to locate the sentences where "and" is missing (for example: "Apart from utilization of these biofuels, other biofuels such as orange oil and turpentine oil...." The use of commas should also be reviewed.
Response: Throughout the paper “and” is given at required place and unnecessary commas are removed.

Comment 5: Page 1, column 2, paragraph 2: "convention" should be "conventional".
Response: **“**convention” is changed to “conventional”.

Comment 6: Page 2, column 1, paragraph 3: please reword "Unlike plant oils....extracted." It is confusing.
Response: Sentence is modified to a better way and unnecessary commas are removed in it.

Comment 7: Page 2, table 3: define "O-D" in the text or in the table title; Add "oil" to the 3rd column heading.
Response: “O-D” is defined in the text above Table 3 and “oil” is added in the Table 3 in the third column.

Comment 8: Table 4: define the abbreviations in the table title or text. Also, the abbreviations on the last page is missing “CV”.

Response: All the abbreviations of Table 4 are defined in Table 2 and the abbreviation of “CV” is added at the last.

Comment 9: Page 4: column 2, paragraph 3: “Fig. 6(a) shows….study.” “Made in the piston” doesn’t make sense.
Response: Sentence mentioned above is rectified and “made in the piston” is removed in the revised manuscript.

Comment 10: Page 6, column 1, paragraph 1: “It can be observed that BSEC decreases by increasing the brake power…” Do you mean to say “by increasing the orange oil concentration”?
Response: In general, BSEC decreases with increasing brake power for all the fuels used in this experiment.

Comment 11: Page 7, column 1, paragraph 1: As written, it says that the HC emissions increase, but I think you are referring to the reference [9]. A better sentence might be” HC emissions increase due to …”
Response: As per your suggestion, sentence is changed to “HC emission increase due to…”

Comment 12: Pages 9-10: there are 3 instances of “40” instead of “40%”.
Response: “40” is changed to “40%”

Comment 13: Page 9, column 2, last paragraph: Incomplete sentence and confusing. Please reword.
Response: As per your suggestion, sentence is changed accordingly.

Comment 14: Page 10, column 2, paragraph 2: “…is higher while...” should this be “is higher than diesel while…”?
Response: As per your suggestion “…is higher while…” is changed to “is higher than diesel while…”