Performance of Renewable Fuel Based CI engine

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ABSTRACT

A multi cylinder naturally aspirated diesel genset (DG) was operated successfully with renewable fuels (bio-diesel of non-edible plant oil such as Jatropha oil, karanja oil, rice bran oil & producer gas) and its performance was verified through extensive, short and long duration trials. Study reveals that mixture of bio-diesel and producer gas offer better break thermal efficiency compared to mixture of fossil-diesel and producer gas. Maximum replacement of bio-diesel by producer gas was 86% with minor losses in engine output compared to fossil-diesel.

In general, exhaust gas temperature and specific energy consumption increased with renewable fuel compared to fossil-diesel. It was due to lower calorific value of bio-diesel and producer gas. In compression ignition (CI) engine having 18.4:1 simulated compression ratio, at 84% engine load and with renewable fuel concentration of pollutants like carbon monoxide (CO), hydrocarbon (HC), nitric oxide (NO), nitrogen dioxide (NO_2) were reduced, in general compared to fossil-diesel. However concentration of pollutants were more while compared to fossil-diesel – producer gas mixture.

Keywords: D G set, renewable fuels (bio-diesel and producer gas), engine performance, emission characteristics

1. INTRODUCTION

Compression ignition (CI) engine could be operated with following fuels either alone or in the form of mixture (Figure1). Use of fossil-diesel in CI engine is a well-proven technology.

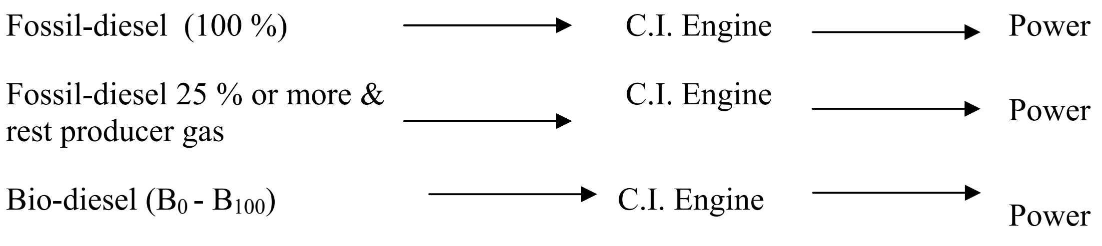

Figure 1. Fuels and their mixture for CI engine so far

In India, a large variety of biomass feedstock is available in huge amounts. As these are available locally, biomass gasifier – based power generation may be an appropriate option for decentralized power generation in many parts of the country. Biomass gasification is one such process where producer gas could be obtained from biomass feed stocks and in turn use the producer gas for power generation purposes. The cumulative installed capacity of biomass based power generation in the country as of 2003-2004 was only 631 MW (including cogeneration and biomass gasifiers) as against an estimated potential of 19500 MW (Pathak, 2004). Biomass gasifier – based system capable of producing power from a few kilowatts up to several hundred kilowatts have been successfully developed indigenously. The utilization of producer gas in the diesel engine in dual fuel operation is an established technology for conservation of fossil-diesel. Producer gas could be used in C I engine, without any modification in the engine. However, it cannot replace the fossil-diesel completely. Fossil-diesel replacements up to 70 – 90 % have been achieved in the dual fuel mode. Because of its poor ignition / delay ignition characteristics some minimum amount of fossil-diesel is required to start the ignition (Mohad et al 2003, Parikh and Arikkat, 1985).

On the other hand, plant oil esters (bio-diesel) can be blended with fossil-diesel in any proportion and could be used successfully in CI engine without any problems. However, emission of NOx increased with use of bio-diesel. Although bio-diesel has several advantages over fossil-diesel, in the present scenario, the use of plant oils and its derivative (bio-diesel) are restricted due to its high cost compared to commercially available fossil-diesel. Since combustion temperature of biomass-based producer gas is lower than that of fossil-diesel / bio-diesel combustion, it is anticipated that there will be lower NOx emission in the former case.

Hence a study was undertaken at Sardar Patel Renewable Energy Research Institute (SPRERI) Vallabh Vidyanagar with the main objective of developing a CI engine, which could be run using renewable fuels.

2. MATERIAL AND METHODS

2.1 Characteristics of Fuels

Bio-diesel of non-edible plant oils was prepared as per the procedure recommended by Gupta (Gupta 1984, Sangha et al., 2004). Characterization of fossil-diesel and methyl ester of plant oils was done as per the ASTM standards (ASTM 19103, 1983). Various characteristics studied were kinematic viscosity, density, gross heating value, flash point, cloud point and carbon residues.

2.2 Experimental Setup and Measuring Devices Used

A multi cylinder naturally aspirated DG set (Table 1) with matching alternator was used for the studies. An electrical heated loading resistance was used for loading the engine. The liquid fuel flow rate was measured on volumetric basis. Bio-diesel from three non- edible plant oil named as Jatropha oil, karanja oil and rice bran oil were prepared and tested in similar conditions. A microprocessor based flue gas analyzer was used for the measurement of emissions concentration. A gas flow rate recorder (manufactured by Star Scientific, Mumbai, India) was used to measure the flow rate of producer gas. Producer gas supply rate was controlled through an inlet valve. There was not control on aspirated air. To know the maximum intake of producer gas by engine, producer gas control valve was slowly opened till the engine started faltering. Later producer gas control valve was turned back slightly and the intake of producer gas at that point was taken as the maximum replacement in the engine for each operating condition. For proper mixing of air and producer gas before entering to the compression ignition (CI) engine intake, an air producer gas mixing chamber of 300 mm diameter and length was designed and fabricated with 2 mm thick mild steel plate. For achieving higher simulated compression ratio, the mixture of producer gas and air was supplied under pressure. Experiments were carried out on the engine at three loads (63 %, 84 % and 98 %) using fossil-diesel, bio-diesel and fossil-diesel + producer gas respectively to provide base data line. In addition to emissions data, parameters related to thermal performance of engine such as fuel consumption, crank oil temperature, noise level, rpm of engine, tar and soot particulate matter (SPM) in producer gas, gas composition, flue gas temperature were also measured and recorded. The calorific value of producer gas was calculated from its composition. A sampling port was provided in the exhaust pipe for measuring flue gas temperature and to collect flue gas samples.

The producer gas was derived from wood fuel in a down draft gasifier. The gasifier was a choke plate type with central top air nozzle manufactured by M/s Cosmo Products, Raipur, Chhatisgarh (India). Tar and dust contents in raw producer gas obtained from this gasifier ranged from 210 to 250 mg N^{-1} m^{-3}. The gas was further cooled & cleaned using cooling tower, organic filter and fabric filter developed at SPRERI. The tar and particulates matter load of clean producer gas was in the range of 40 -52 mg N^{-1} m^{-3}. Cleaned gas was used to drive the CI engine in dual fuel mode along with bio-diesel. The schematic of the experimental set up is shown in Figure 2.

Table 1: Engine details

Name and Model of engine	:	Kirlosker, KCD 2K
General Details	:	Constant speed, three cylinder, naturally aspirated, four stroke, direct injection
Bore x Stroke	:	100 x 120 mm
Compression ratio	:	17: 1
Rated output	:	25 KVA at 1500 rpm
Fuel injection opening pressure	:	175-180 kg cm^{-2}
Injection timing	:	26° BTDC
Engine conditions	:	New engine operated few hours at fossil-diesel only.

For conducting the experiments the engine was always started and closed in bio-diesel mode. After stabilization of engine (after 30 minutes of engine start up) it was changed into dual fuel (bio-diesel + producer gas) mode. The gasifier was also started during the engine start up and producer gas was flared at producer gas burner. Small pieces of *Prosopis Juliflora* (gandabaval), 20 - 30 mm in diameter and length were used as feedstock for gasification. The test was conducted as per BIS Code No.13018, 1990.

3. RESULTS AND DISCUSSION

3.1 Characterization of Fuel

Fossil-diesel and bio-diesel of non-edible plant oils were characterized for viscosity, density, flash point, cloud point, and calorific value. The results obtained are given in Table 2, which

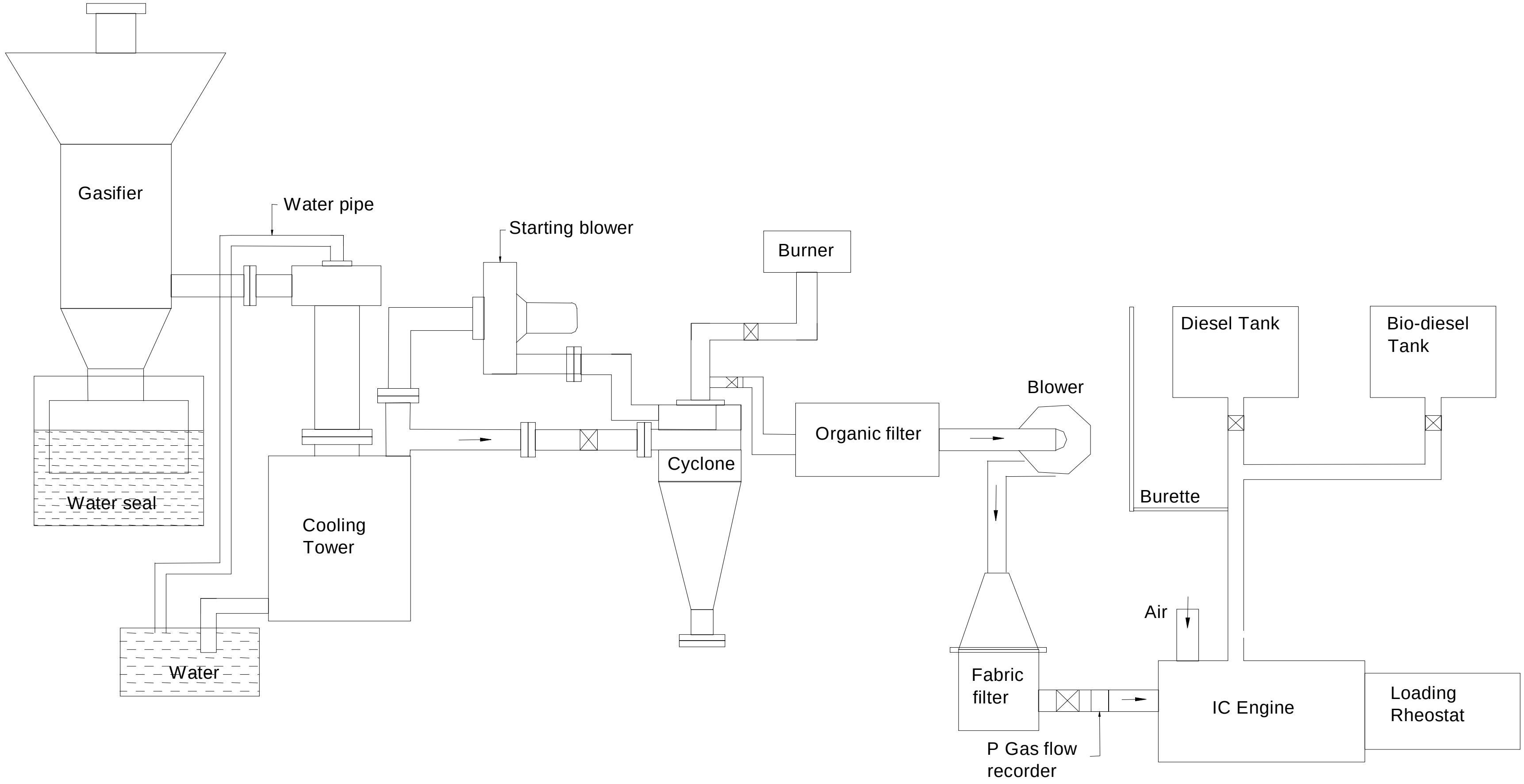

Figure 2. Schematic of the experimental set up for bio-diesel and producer gas

reveal that bio-diesel has slightly higher viscosity than fossil-diesel. Higher viscosity of bio-diesel has an added advantage over fossil-diesel as it also acts as a lubricant to CI engine (Sabeena et al, 2004)

3.2 Performance of CI Engine in Dual Fuels Mode (Bio-Diesel and Producer Gas)

While operating the gasifier CI engine system, liquid fuel economy is one of the major factors. It is reported that, the liquid fuel replacement increased with the increasing load on engine (up to 85 %) and liquid fuel replacement started decreeing if further load increased (Bhattacharya et al., 2001; Pathak, 2004; Sappani et al., 1991; Sridhar et al.2001). It was also reported that exponential increase in HC and CO at low loads where the premixed mixture is likely to be very lean.

Table 2: Characteristic of fuels

Fuels	Viscosity at 38 ºC, cS	Density at 38 ºC, kg/m^3	Flash point, ^{o}C	Cloud point, ^{o}C	Carbon residues, %	CV, kcal/kg
FD	4.438	788.98	66	13	0.03	10404
MEJO	8.818	857.3	175	05	0.024	8076
MEKO	7.02	867.18	162	8.5	--	8300
MERBO	7.231	844.40	180	12	--	8546

FD—Fossil-diesel, MEJO - Methyl ester of Jatropha oil, MEKO - Methyl ester of karanja oil, MERBO - Methyl ester of rice bran oil, CV – Calorific value

At about 80–85 % load these emissions are minimum. Considering the above fact, the performance of the DG set was tested at 2 compression ratios—17:1(normal) and 18.4:1 (Simulated). First DG set was run in dual fuel mode (fossil-diesel + producer gas) at 42, 63, 84 and 98 % engine load at normal compression. It was found that at 63 % engine load replacement of fossil-diesel was higher (76.6 %) compared to others load, but emissions were also high. At 84 % engine load emission of CO and HC reduced 6.44 & 60.34 % respectively, however fossil-diesel replacement was also reduced from 76.6 to 63.21%. Later compression ratio of engine was simulated (18.4:1) using a high-pressure (1200 mm of water column for 100 m^3 h^{-1} flow discharge) centrifugal blower. By increasing the air and producer gas mixture pressure of CI engine, CR could be simulated. The performance of DG set in terms of fossil-diesel replacement and emission concentration, improved. Finally at simulated higher compression ratio (18.4:1), bio-diesel of non-edible plant oil were used with producer gas to run CI engine at 63, 84 and 98 % engine load.

3.2.1 Specific Energy Consumption and Brake Thermal Efficiency

Since the brake specific fuel consumption is not a very reliable parameter to compare the two fuels having different calorific values and density, brake specific energy consumption was preferred to compare the performance of CI engine at different load for mixture of bio-diesel + producer gas. Specific energy consumption in dual fuel mode (bio-diesel and producer gas) was calculated from the fuel consumption and calorific value of bio-diesel and producer gas. Specific energy consumption and brake thermal efficiency for dual fuel was shown in Table 3, which indicate that the brake thermal efficiency goes down when the engine is operated on bio-diesel and producer gas mixture compare to bio-diesel. The decrease in brake thermal efficiency is dependent on the share of the producer gas in the bio-diesel – producer gas mixture. The lowering of the brake thermal efficiency can be at least partially explained by the slow progress of the combustion reported in the case of producer gas (Pathak, 2004). Also producer gas contains lowest percentage (40% maximum) of combustible gas and large fraction of inert gases like CO_2 and N_2 accounting to 12 – 15% and 48 – 50 % respectively compared to natural gas (80 % combustible gas). In case of dual fuel operation of CI engine with natural gas thermal efficiency increase with increase in the percentage of natural gas. With producer gas it decreases due to increasing percentage of inert gases (N_2 and CO_2) in the air-fuel mixture. According to Asokan, 1990; Bhattacharya et al., 2001; Uma et al., 2004 the efficiency of engine – generator operating on dual fuel (diesel + producer gas) was always lower than pure diesel operation. Some times these differences are as high as up to 8 %. It could be due to reduced heating value and lower combustion temperature of producer gas air mixture, drop in the pressure of the gas entering the air inlet and lower flame velocity. De-rating of the engine did not take place because of the use of blower in the system.

Comparing bio-diesel and producer gas mixture with fossil-diesel and producer gas, brake thermal efficiency was higher. The increase in brake thermal efficiency in the case of bio-diesel & producer gas mixture as compared to the mixture of fossil-diesel and producer gas may be attributed to better combustion due to increased percentage of oxygen in bio-diesel as well as additional lubricity of bio-diesel, leading to declination in the frictional horsepower losses in the engine (Sabeena et al., 2004).

3.2.2 Replacement of Liquid Fuel in Dual Fuel Mode

Liquid fuel replacement rates under different load conditions have been calculated from the liquid fuel consumption in bio-diesel mode and liquid fuel consumption in dual fuel mode and are shown in Table 3, which shows that the replacement of liquid fuel depends upon the load and producer gas quality and quantity at the time of measurement. It was observed that liquid fuel replacement was highest (78.34%) at 63% engine load in the case of mixture of methyl ester of karanja oil and producer gas. It may be partially explained by the inability of the engine to take sufficient producer gas to maintain high levels of liquid fuel replacement if the load was more than 63% of the rated capacity. At higher loads the engine started using more liquid fuel in order to maintain a constant speed. It can be inferred that a higher calorific value producer gas would have given high liquid fuel replacement at engine loads exceeding 63%.

3.2.3 Effect on Noise Level at Dual Fuel Mode

The noise level of the engine at different loads was measured with the engine operating on the bio-diesel and on dual fuel (mixtures of bio-diesel and producer gas) mode. Minor difference was noted in the noise level, which was found to be in the range of 96.2 to 102.15 db for bio-diesel and 96.0 to 101.00 db for dual fuel (mixtures of methyl esters of rice bran oil and producer gas) respectively at the level of the operator’s ear standing about a meter from the engine fly wheel (Table 3). There was, however, a noticeable difference in the quality of sound when the engine was shifted from bio-diesel operation to bio-diesel – producer gas mixture operation.

Table 3: Performance of CI engine operating on fossil-diesel and bio-diesel of different plant oils and mixture of bio-diesel and producer gas

Engine load, %	Mode of operation	RPM of engine	Engine output, kW	LFCR, Kg/h	LFR, %	SEC MJ/kWh	η_{bth}, %	Sound pressure level, db
63 %	**FD**	**1487**	**14.084**	**3.586**	**--**	**11.07**	**32.53**	**100.5**
63 %	MEJO	1490	13.821	4.183	--	10.23	35.19	97.2
63 %	MEKO	1489	13.964	4.117	--	09.89	36.40	97.23
63 %	MERBO	1484	14.14	4.169	--	10.50	34.30	97.87
63 %	FD & P gas	1490	14.00	1.336	62.74	23.93	15.05	96.6
63 %	MEJO & P gas)	1488	13.922	0.901	78.44	19.65	18.33	98
63 %	MEKO & P gas)	1491	13.932	0.558	86.45	18.45	19.54	96.6
63 %	MERBO & P gas)	1492	13.970	0.903	78.34	22.00	16.37	97.63
84 %	**FD**	**1493**	**19.336**	**4.517**	**--**	**10.15**	**35.45**	**101.5**
84 %	MEJO	1490	19.114	5.282	--	9.34	38.54	98.53
84 %	MEKO	1494	19.33	5.281	--	9.16	39.30	98.5
84 %	MERBO	1498	18.94	5.368	--	10.09	35.69	98.77
84 %	FD & P gas	1483	18.93	1.445	68	19.26	18.70	102.15
84 %	MEJO & P gas)	1483	18.23	2.226	57.86	16.69	21.58	99.6
84 %	MEKO & P gas)	1487	18.49	2.434	54	16.45	21.89	1002
84 %	MERBO & P gas)	1491	18.94	2.105	60.79	18.47	19.49	100.7
98 %	**FD**	**1486**	**22.59**	**5.38**	**--**	**10.35**	**34.77**	**99.5**
98 %	MEJO	1485	22.52	6.139	--	9.22	39.06	100.5
98 %	MEKO	1488	22.53	6.21	--	9.15	39.33	99.77
98 %	MERBO	1486	22.21	6.103	--	9.78	36.82	102.0
98 %	FD & P gas	1489	22.00	4.183	22.24	15.08	23.87	100.4
98 %	MEJO & P gas)	1484	22.23	4.976	18.94	15.47	23.27	100.8
98 %	MEKO & P gas)	1487	22.22	4.887	21.30	14.94	24.09	100.5
98 %	MERBO & P gas)	1485	22.61	5.074	16.86	11.81	30.49	101

LFCR— Liquid fuel consumption rate, LFR — Liquid fuel replacement, SEC — Specific energy consumption, η_{bth} — Brake thermal efficiency, P gas- producer gas

3.2.4 Emission Characteristics of CI Engine in Dual Fuel Mode

The exhaust gas temperature was found to rise up to 75 ^{0}C when the engine was operated in dual fuel (bio-diesel + producer gas) mode for maximum replacement of bio-diesel by producer gas. The carbon monoxide (CO) content of the exhaust gas increased substantially when the engine was operated in dual fuel mode, while the CO content of the exhaust gas obtained during bio-diesel mode was nominal.

At 84% load of CI engine and simulated compression ratio with dual fuel (bio-diesel & producer gas) mode the concentrations of the pollutant, except CO were less compared to the fossil – diesel. A close looks of Table 4 also reveals that at 84 % engine load having 18.4:1 compression ratio with dual fuel (bio-diesel and producer gas) the concentration of CO was also less compared to other tested loads. The concentration of hydrocarbon (HC) reduced with increasing load and it was lowest (0.21%) at 84% engine load, later it increased.

Table 4: Emission characteristics of CI engine operating on fossil-diesel and bio-diesel of different plant oils and mixture of bio-diesel and producer gas

Mode of operation	Engine load, %	Ambient Temp. °C	Exhaust gas analysis						
			Engine exhaust temp. °C	CO_2 %	CO %	O_2 %	NO ppm	NO_2 ppm	HC %
FD	**63 %**	**35**	**357**	**6.46**	**1.66**	**12.20**	**1099**	**54**	**0.57**
MEJO	63 %	28	341	5.71	0.94	12.20	1657	60	0.01
MEKO	63 %	28	340	5.98	0.62	12.86	1489	68	0.02
MERBO	63 %	33	333	6.62	1.10	11.97	1466	37	0.03
FD & P gas	**63 %**	**32**	**365**	**7.39**	**3.85**	**10.90**	**693**	**75**	**0.29**
MEJO & P gas	63 %	26	384	8.42	3.99	9.55	274	35	0.39
MEKO & P gas	63 %	28	384	9.09	3.99	8.65	128	18	0.35
MERBO & P gas	63 %	33	408	10.16	3.85	6.53	245	6	0.38
FD	**84 %**	**35**	**467**	**8.13**	**2.33**	**9.95**	**1505**	**41**	**0.58**
MEJO	84 %	28	430	7.52	1.19	10.77	1971	65	0.02
MEKO	84 %	28	439	7.70	1.13	10.50	1879	66	0.04
MERBO	84 %	41	513	8.68	1.30	9.20	1947	31	0.05
FD & P gas	**84 %**	**32**	**528**	**6.53**	**1.05**	**12.1**	**269**	**7**	**0.97**
MEJO & P gas	84 %	26	476	10.80	2.76	6.33	510	14	0.23
MEKO & P gas	84 %	28	493	10.38	3.21	6.85	771	22	0.26
MERBO & P gas	84 %	34	516	11.14	3.10	5.95	576	05	0.21
FD	**98 %**	**35**	**540**	**9.34**	**2.43**	**8.30**	**1694**	**32**	**0.62**
MEJO	98 %	28	521	8.77	1.64	9.13	2210	49	0.07
MEKO	98 %	26	537	9.45	1.70	8.20	1973	49	0.10
MERBO	98 %	34	516	9.64	2.02	7.90	2053	42	0.12
FD & P gas	***98 %***	***35***	***573***	***9.57***	***3.26***	***8.00***	***1208***	***23***	***0.27***
MEJO & P gas	98 %	26	669	10.94	3.99	6.20	1023	17	0.39
MEKO & P gas	98 %	28	588	11.57	3.99	5.37	1394	18	0.34
MERBO & P gas	98 %	34	576	11.20	3.99	5.80	1520	19	0.31

FD—Fossil-diesel, MEJO - Methyl ester of Jatropha oil, MEKO - Methyl ester of karanja oil, MERBO - Methyl ester of rice bran oil, P gas – Producer gas

With the application of producer gas in CI engine, NOx concentration decreased drastically. It is more effective at lower engine load compared to higher load. Since the NO emission concentration dependent on combustion chamber temperature which in turn was dependent on the load. Lower load led to less supply of fuel in the combustion chamber, thus, producing lower flame temperature, ultimately lower concentration of NO appeared in the exhaust gas. Carbon dioxide emission in case of dual fuel (bio-diesel + producer gas) operation for C I engine is not considered since the carbon in the bio-diesel/producer gas is a part of global carbon cycle and hence, does not contribute to global warming (Table 4).

3.3 Comparison of DG Set Performances with Mixture of Bio-Diesel and Producer Gas

Computed value of engine output, specific energy consumption, brake thermal efficiency, producer gas flow rate, liquid fuel replacement and sound pressure level at different fuels and engine loads having simulated compression ratio 18.4:1, are summarized in Table 3, which reveals that there was no significant different in engine output with use of bio-diesel made from different oil. Brake thermal efficiency and engine output was found higher with methyl ester of rice bran oil in case of single as well as dual fuel mode, however liquid fuel replacement was found maximum with methyl ester of karanja oil at 63% load. It may be due to difference in oxygen content o bio-diesel.

3.4 Comparison of Emission Characteristics of DG Set with Mixture of Bio-Diesel and Producer Gas

Computed value of exhaust temperature and exhaust composition at different fuels and engine loads having simulated compression ratio 18.4:1, are summarized in Table 4, which indicate that with the application of producer gas in CI engine at dual fuel mode al the pollutant, except CO, decreased considerable. The reduction in NOx is more effective at 63% load compared to higher loads for all the mixture. Table also indicates that with application of producer gas in CI engine, exhaust temperature increase due to late burning characteristics of producer gas. But in general it is more effective with mixture of methyl ester of rice bran oil and producer gas.

4. CONCLUSIONS

The Following conclusions have been drawn based on experimental results.

1. Compression ignition engine could be operated successfully without conventional fuel.
2. The use of producer gas to fuel a CI engine along with bio-diesel gives higher brake thermal efficiency than fossil-diesel producer gas mixture.
3. Maximum replacement of bio-diesel by producer gas was 86% at 63% engine load and simulated compression ratio of 18.4:1.
4. Addition of producer gas to bio-diesel has significantly reduced NOx emission but it increases the emission concentration of other pollutants.

5. ACKNOWLEDGEMENT

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