Effect of Port Premixed Liquefied Petroleum Gas on the Engine Characteristics

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In the present work, liquefied petroleum gas (LPG) is premixed with air for combustion in a compression ignition engine, along with neat rubber seed oil as the direct injected fuel. The LPG is injected directly into the intake manifold using an electronic gas injector. The var-iation in the LPG flow rate is from zero to the maximum tolerable value. The engine load was varied from no load to full load at regular intervals of 25% of full load. Experimental results indicate a reduction in thermal efficiency at low loads, followed by a small improve-ment in the thermal efficiency at 75% and 100% loads. Premixing of LPG prolongs the delay in the ignition with a simultaneous decrease in the duration of combustion. With an increase in the LPG flow rate, the maximum in-cylinder pressure increased at high outputs, whereas it decreased at low outputs. The heat release rate shows that the combus-tion rate increases with LPG induction. Carbon monoxide (CO) and hydrocarbon (HC) levels reduced at high outputs, whereas at all loads, the oxides of nitrogen (NOx) levels increased. The NOx level at full load increased from 6.9 g/kWh at no LPG induction to 10.36 g/kWh at 47.63% LPG induction. At all loads, the smoke level decreased drastically. The smoke level at full load decreased from 6.1BSU at no LPG induction to 3.9BSU at 47.63% LPG induction.

Introduction

Almost all the developing countries are highly dependent upon the diesel engine vehicles. The torque, power, and fuel economy of diesel engines make them ideal for various applications. However, due to the shortage of fossil fuels, we may not avail its services for a long time. Hence, efforts are underway, to find alternative fuels for the diesel engine vehicles [1,2]. Vegetable oils are a promising alternative to diesel [3–6]. Rudolf diesel confidently predicted that the diesel engines will be operated with vegetable oils since they have major advantages as a substitute for diesel. It is renewable in nature and the properties are very close to diesel oil in many aspects. In rural areas, the production of vegetable oil takes place by using well-known agricultural practices. However, the high viscosity of vegetable oil leads to inefficient spray resulting in improper mixing with air and hence decreases the engine performance. Long-term use of vegetable oil causes carbon deposits, fuel filter clogging, sticking of piston rings, and thickening of lubricating oil, which necessitates the change of engine parts regularly [6–8]. Poor volatility also makes the vegetable oil difficult to evaporate and ignite. Hence, the vegetable oil has to be further processed or the engine needs to be modified to render the use of vegetable oils practicable.

Formation of a homogeneous mixture prior to ignition can largely reduce the problem of emissions and improve thermal efficiency. A gaseous fuel or a volatile liquid fuel can be inducted in the inlet manifold to form a homogeneous mixture whose combustion can be started by injecting small quantities of diesel or vegetable oil [9,10]. Zhang et al. [11] during fumigation of ethanol and methanol observed reduction in the brake thermal efficiency (BTE) at low loads for both the alcohols. With an increase in the fumigation rate, carbon monoxide (CO) and hydrocarbon (HC) emission increased with emissions being higher for methanol as compared with ethanol. However, an increase in fumigation reduces the NOx and particulate emissions. With methanol fumigation producing higher NOx emission but lower particulate emission as compared with ethanol fumigation. Edwin Geo et al. [12] used diethyl ether (DEE) as the port injected fuel in an agricultural engine. The engine's efficiency improved by varying the timing and quantity of DEE injection. Maximum BTE of 28.5% is recorded in dual-fuel mode when the DEE injection rate of 200 g/h is maintained. Except NOx, all other emissions reduced. With DEE injection, the heat release rate is high during the initial stages of combustion contributing to lower combustion duration. Lim et al. [13] modified an existing diesel engine, which could inject diesel at high pressures using common rail direct injection (CRDI) technique while compressed natural gas (CNG) was injected at the intake port for premixing. NO_X and HC emissions were higher than diesel due to rapid combustion, soon after premixing. A reduction of 30% was seen in CO_2 emission while particulate matter (PM) emissions decreased significantly as less amount of diesel was used for ignition. The authors also observed an increase in power output with dual-fuel mode as compared with single-fuel mode.

Liquefied petroleum gas (LPG) is formed when crude oil is thermally cracked. Its main constituents are propylene, butane, propane, and other light hydrocarbons. LPG has a high octane number, which makes it a good spark ignition fuel, whereas, it is difficult to use LPG in compression ignition engines due to their low cetane number [14]. However, it is possible to use LPG in diesel engines in port premixed charge mode.

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Poonia et al. [15] conducted experiments by inducting LPG with diesel as the direct injected fuel in a compression ignition (CI) engine at various intake charge temperatures and LPG flow rates. The authors also improved the engine's performance by throttling the charge along with exhaust gas recirculation (EGR). The authors at low loads observed improvement in brake thermal efficiency as the intake temperature and pilot quantity increases. The use of high pilot quantity and high intake temperature along with low pilot injection rates lead to a reduction in HC levels. Throttling of intake improves the combustion rate, which also improves the brake thermal efficiency, and along with EGR, the HC level is further reduced. Experiments on a genset engine were performed by Lata et al. [16] with hydrogen and LPG as the inducted fuel. With only LPG, the brake thermal efficiency improved by 6% with 40% energy share of the LPG. HC and CO emission increased but NOx and smoke level reduced. With 40% mixture of hydrogen and LPG (70:30 ratio), BTE increased by 27% and HC emission reduced by 68%. Lata et al. [17] also studied the combustion parameters for the same engine. The author observed the increase of 1.37 bar/°CA in the rate of pressure rise, 6.95 bar increase in peak cylinder pressure, and 5 °CA increase in combustion duration with 30% LPG as secondary fuel. The 40% mixture of hydrogen and LPG also improved the combustion. Tira et al. [18] observed the reduction in CO, HC, and soot emission for LPG—rapeseed methyl ester dual-fuelled engine in comparison to LPG—diesel engine operation. Mohanan and Suresh Kumar [19] tried improvement in engine performance by inducting LPG into the diesel-fuelled engine. The authors varied the pilot fuel quantity, intake temperature, and injection timing of LPG. At 27 °BTDC ignition timing, 50 °C LPG intake temperature, and 0.45 kg/h pilot diesel, the dual-fuel engine's efficiency is higher than single-fuel mode. Unburned hydrocarbon level reduced by using a high quantity of pilot fuel and at high LPG intake temperature. CO level also decreased with high LPG intake temperature. Cernat et al. [20] investigated the combustion characteristics for LPG port injected diesel engine. From this analysis, the author establishes the maximum dose of LPG that the engine can run on reliably. At the beginning of combustion, the high cyclic variability increases due to LPG substitution. With the cyclic variability limited to 10% for indicated mean effective pressure, the author observed that at 2000 rpm, the maximum LPG substituted is 8%, whereas at 4000 rpm, it is 25%. With a limit of 8 bar/°CA on maximum pressure rise, the LPG substitute ratio is 10% for 2000 rpm engine speed and 40% for 4000 rpm engine speed.

In the present work, a diesel engine's performance fuelled with rubber seed oil (RSO) is improved by inducting LPG in the intake manifold at various flow rates. A single cylinder diesel engine was used for the experimental work. To form the baseline, the engine's performance was also evaluated with neat diesel and RSO. Tables 1 and 2 show the properties of liquid fuel and gaseous fuel, respectively.

Experimental Setup and Procedure for Testing. The present work is carried out on an air-cooled, single cylinder, direct injection engine developing a power of 4.4 kW. The test engine specifications are shown in Table 3. An electrical dynamometer was used to load the engine. The intake manifold is connected to a surge tank through an orifice meter that measures the airflow rate. The LPG tank is connected through a flame trap to an electronically controlled gas injector, placed in the intake manifold. The LPG flow rate is manually controlled and measured with the help of a gas flow meter. The top dead center (TDC) signal is acquired using an optical shaft position encoder. The liquid fuel flow rate (RSO) is measured using a burette/stopwatch.

Table 1 Properties of liquid fuel

Property	RSO	Diesel
Specific gravity	0.922	0.83
Kinematic viscosity (cSt)	33.91	3.8
Flash point (°C)	198	50
Calorific value (MJ/kg)	37.5	42.9
Iodine value	135.3	38.3
Acid value	23.8	0.062
Cetane number	37	47

Table 2 Properties of gaseous fuel

Property	LPG
Density at 1 atm. and 15 °C (kg/m^3)	2.24
Flame speed (cm/s)	38.25
Stoichiometric A/F ratio	15.5
Flammability limits (vol% in air)	2.15–9.6
Octane number	103–105
Auto-ignition temperature (°C)	493–549
Lower calorific value (LCV) at 1 atm. and 15 °C (kJ/kg)	46,000

Table 3 Specification of the test engine

Make	Kirloskar, TAF1
Type	Single cylinder, CI, 4-stroke
Type of cooling	Air cooled
Bore * Stroke	87.5 * 110 mm
Compression ratio	17.5:1
Piston bowl	Hemispherical
Rated power	4.4 kW at 1500 rpm
Displacement	661.5 cc
Fuel injector opening pressure	200 bar
Lubrication oil	SAE 40

A pressure transducer containing piezoelectric material is placed on the cylinder head to measure the cylinder pressure for every 1 deg crank angle rotation. An acquisition system acquires the signal from the sensors and sends it to a personal computer. The pressure versus crank angle values are computed by averaging 100 consecutive cycles by a software. From the computed values, heat release rate, peak pressure, the maximum rate of pressure rise, and the occurrence of peak pressure are determined. A QROTECH, QRO-401 exhaust gas analyzer measures the unburnt HC, CO, and oxides of nitrogen (NOx) emissions. Nondispersive infrared principle is used by the analyzer to measure HC and CO emissions, whereas the electrochemical method is used to measure NOx emissions. The smoke density is measured using a Bosch-type smoke meter. The schematic of the experimental setup is shown in Fig. 1. The accuracy of various instruments used in the experimental work is tabulated in Table 4.

The baseline data were generated by operating the diesel engine with neat RSO and diesel. The readings were taken in triplicate for all the loads, and they were averaged. The engine's performance was evaluated using exhaust gas temperature (EGT), BTE, brake specific energy consumption (BSEC), and emission and combustion characteristics. LPG was then used to conduct the experiments at 25%, 50%, 75%, and 100% of full load. Initial testing showed that at low loads the maximum amount of LPG substitution is limited by unstable operation, whereas at high loads, it is limited by knock. The LPG flow rates were also varied to conduct the experiments.

To evaluate the accuracy of the engine performance, it is necessary to carry out the uncertainty analysis. The estimation of uncertainty consists of error in the instruments used, changes in climatic condition, or due to human error. Therefore, it is necessary to evaluate the overall uncertainty of the experiment. The estimation of uncertainty of some important parameters from known measured

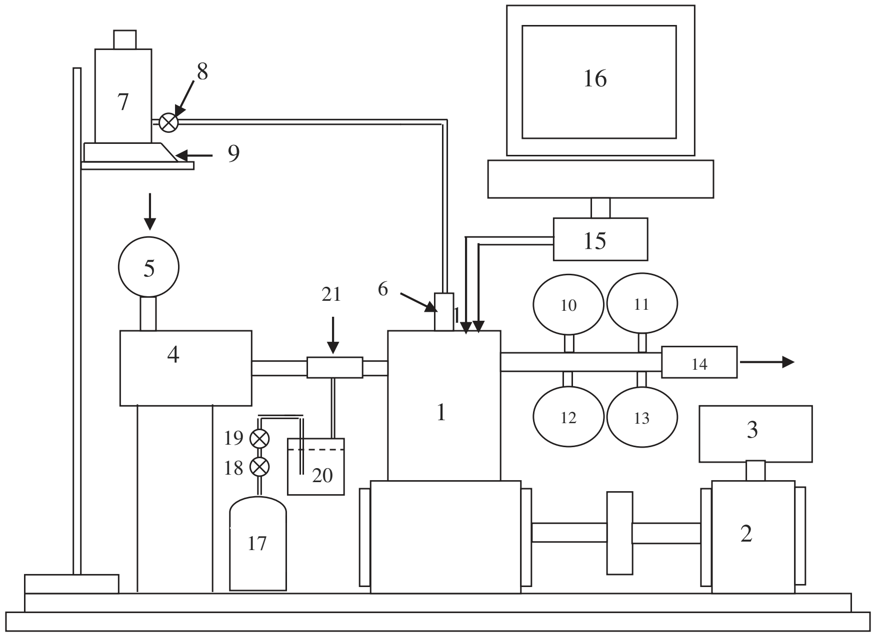

Fig. 1 Schematic of the experimental setup: 1, engine; 2, dynamometer; 3, control panel; 4, air tank; 5, air flow meter; 6, injector; 7, fuel tank; 8, fuel valve; 9, balance; 10, smoke pump; 11, FID; 12, CO analyzer; 13, NO analyzer; 14, silencer; 15, charge amplifier; 16, computer; 17, LPG cylinder; 18, pressure regulator; 19, control valve; 20, flame trap; 21, LPG injector

values is evaluated based on the square root method. The overall uncertainty of the experiment is calculated as follows:

$$\begin{aligned}\text{Overall uncertainty} &= \text{Square root of}\Big\{(\text{uncertainty of BP})^2 + (\text{uncertainty of BSFC})^2 + (\text{uncertainty of BTE})^2 \\ &\quad + (\text{uncertainty of HC})^2 + (\text{uncertainty of CO})^2 + (\text{uncertainty of NOx})^2 + (\text{uncertainty of smoke})^2\Big\} \\ &= \text{Square root of}\Big\{(0.05)^2 + (0.1)^2 + (0.498)^2 + (0.01)^2 + (0.8)^2 + (1.1)^2 + (2.0)^2\Big\} = 2.47\%\end{aligned}$$

Results and Discussion

Analysis of the Engine With Neat Rubber Seed Oil and Diesel in Single-Fuel Mode. The engine was initially operated in single-fuel mode with neat RSO and diesel to form the baseline. The engine's performance and emission characteristics with neat RSO and diesel are tabulated in Table 5.

Table 4 Accuracy of various measuring instruments

S. No.	Instruments	Range	Accuracy	Uncertainty (%)
1	Engine speed	1–1500 rpm	±10 rpm	0.2
2	Exhaust gas temperature	0–900 °C	±1 °C	0.2
3	Digital stopwatch		±0.1 s	0.3
4	Pressure pickup	0–250 bar	±0.1 bar	0.1
5	Crank angle encoder	0–360 deg	±1 deg	0.2
6	U-tube manometer	0–500 mm	±1 mm	1.0
7	Burette system	0–100 cc	±0.1 cc	1.0
8	Exhaust emission analyser	CO: 0–9.95%	±0.1 cc	0.2
		HC: 0–9999 ppm	±10 ppm	0.3
		CO_2: 0–20%	±0.02%	0.2
		NOx: 0–5000 ppm	±10 ppm	0.4
9	Bosch smoke meter	0–10 Bosch smoke number	±0.2	1.0

Performance Parameters. At all loads, RSO engine operation tends to lower the brake thermal efficiency in comparison to diesel. Whereas, the EGT and energy consumption is higher with RSO at all loads. The poor engine characteristics with RSO operation is due to its physical properties, such as viscosity and density higher than diesel, which affects the atomization and formation of a combustible mixture. Similar results were obtained by Reddy and Ramesh [21] while operating a CI engine with neat jatropha oil. Devan and Mahalakshmi [22] also observed similar results while operating a CI engine with neat poon oil. Another reason is the dominance of the diffusion combustion phase for RSO operation resulting in the release of energy until the end of the power stroke. Therefore, the energy lost to the exhaust gas is more resulting in increased EGT and lower BTE.

Emission Parameters. In comparison to diesel, RSO engine operation resulted in higher CO, HC and smoke emissions for all loads. Due to heavy molecular structure and high viscosity of RSO, the fuel atomization is not proper resulting in larger droplet size during injection and poor mixture formation. As the diffusion combustion phase is dominant with RSO engine operation, some amount of fuel combusts in the later stages of power stroke resulting in incomplete oxidation of fuel due to lack of time resulting in higher emissions. Higher HC, CO, and smoke emissions were also observed by Agarwal and Agarwal [23] with neat jatropha oil engine operation. The authors tried to reduce the viscosity of the oil by preheating it using waste heat of the exhaust gas and

Table 5 Performance, emission, and combustion values for diesel and RSO at different loads

	25% Load		50% Load		75% Load		100% Load	
	Diesel	RSO	Diesel	RSO	Diesel	RSO	Diesel	RSO
BTE (%)	16.5	13.8	24.4	20	27.9	24.3	29.9	26.5
BSEC (MJ/kWh)	21.8	26	14.8	17.9	12.8	14.8	12	13.5
EGT (°C)	180	231	237	281	303	347	364	410
Smoke (BSU)	0.6	1.6	0.9	2.4	1.9	3.8	3.4	6.1
CO (g/kWh)	7	9.2	4.2	5.9	3.2	5.3	3.1	4.7
HC (g/kWh)	0.9	1.6	0.7	1.1	0.6	0.8	0.5	0.7
NOx (g/kWh)	14.4	11	12.7	8.7	11.7	7.9	10.7	6.9
Peak pressure (bar)	62	58	67	62.1	71	67.2	75.5	72
Ignition delay (°CA)	10	13.5	8	10	7	9	6	8
Combustion duration (°CA)	33	37	35	40	39	45	41	47

found some improvement in these emissions. The authors argued that by heating the oil its atomization improves resulting in improved combustion thereby reducing emissions.

NOx emissions occur at high combustion temperatures when the nitrogen present in the incoming air is oxidized. NOx emissions are lower at low loads with RSO engine operation as compared with diesel. NOx is reduced because of less intense premixed combustion resulting in lower combustion temperature. Haldar et al. [24] also observed lower NOx emissions with karanja, jatropha, and putranjiva as compared with diesel engine operation.

Combustion Parameters. Due to lower combustion rates, RSO engine operation results in lower peak pressures. The maximum pressure, at full load, with diesel and RSO engine operation, is about 75.5 bar and 72 bar, respectively. The peak pressure in CI engines is strongly dependent upon the initial combustion rate. If large quantity of fuel takes part in the uncontrolled combustion phase, the initial combustion rate is high. During RSO engine operation less quantity of fuel is ready for burning during the initial phase of combustion, hence the initial combustion rate is low, resulting in lower peak pressure. During the operation of a CI engine with neat poon oil, Devan and Mahalakshmi [22] observed that the first peak in the heat release curve is lower than diesel, whereas the second peak is higher than diesel. Therefore, the peak pressure with neat poon oil is lower than diesel. The ignition delay with RSO engine operation at full load is 8 °CA, whereas for diesel, it is 6 °CA. Increase in power output reduces the ignition delay for both the fuels, as the engine's operating temperature at high loads is high. The viscosity of the raw vegetable oil is high, and its volatility is low resulting in slow vaporization and fuel-air mixing attributing to the increase in delay of ignition. For both the fuels, an increase in engine load increases the combustion duration as at higher loads more quantity of fuel is required to produce the necessary power [25]. RSO engine operation resulted in longer combustion duration than diesel due to the fuel burning deep in the power stroke.

Analysis of Engine With Rubber Seed Oil—LPG Injection in Dual-Fuel Mode. At each engine load, the engine was initially operated with neat RSO and then the amount of LPG injected was gradually increased until the engine either misfires or knocks. The graphs are plotted based on the percentage of the total energy obtained from LPG. Thus, 0% on the *x*-axis indicates neat rubber seed oil (RSO) mode.

Performance Parameters. The variation in brake thermal efficiency with the percentage of LPG energy at different outputs is depicted in Fig. 2. A marginal rise in the BTE is observed at high loads, namely 75% and 100% with LPG injection, due to fast burning of the LPG. At low outputs, BTE reduces with increase in the LPG substitution. Since the quantity of RSO injected into the cylinder is less, when LPG is inducted, the amount of RSO injected reduces further leading to poor ignition of the secondary fuel. In addition, due to lean LPG air mixture, the burning rate is low and the combustion is incomplete resulting in poor BTE at low loads [26]. The reduction in BTE due to this effect is a notable drawback of premixed gaseous fuel engine operation. With very high LPG admission rate, BTE falls due to knock at high outputs. Knocking occurs due to faster combustion of the gaseous fuel. At full load, the BTE increases from 26.6% with neat RSO to a maximum of 28% at an energy share of 42% for LPG (maximum brake thermal efficiency point), whereas for neat diesel operation, the brake thermal efficiency is 29.9%. The knock limited LPG substitutions are 59% with diesel and 49% with RSO as the primary fuel at 100% load.

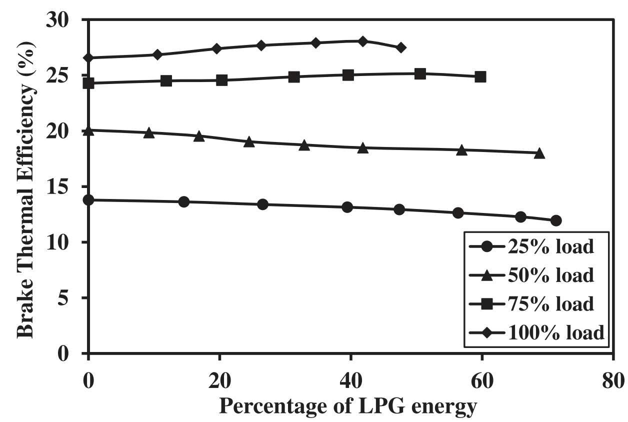

Fig. 2 Variation of brake thermal efficiency with LPG-RSO dual-fuel engine operation at different loads

Figure 3 illustrates the variation in exhaust gas temperature (EGT) with LPG substitution. As the load on the engine increases, the EGT also increases. However, the EGT increases marginally with an increase in LPG energy share. The maximum amount of LPG that can be utilized for every load is limited as depicted in Fig. 3 after which the combustion became unstable and the engine started knocking. The exhaust gas temperature at 25% load and 71.28% LPG share is 245 °C, at 50% load and 68.76% LPG energy share is 290 °C, and at 75% load and 59.77% LPG energy share, it is 378 °C. At the maximum BTE point, the EGT is 432 °C. The EGT rises probably due to the higher gas temperatures reached with LPG induction. Adjustment of the injection timing may reduce the EGT and also improve the thermal efficiency at these conditions.

Emission Parameters. The modulation of unburned hydrocarbon emission with LPG energy share at various loads is depicted in Fig. 4. The HC level reduces with the increase in load and

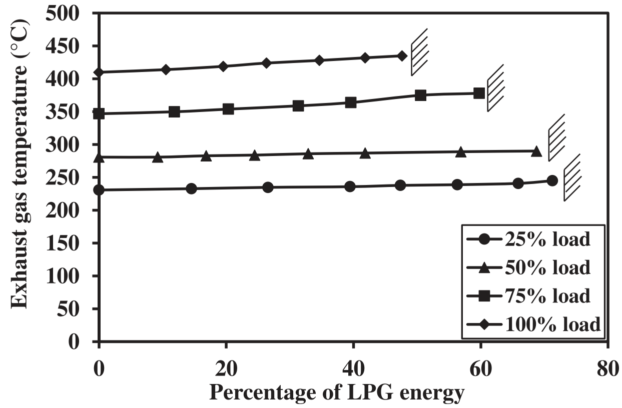

Fig. 3 Variation of exhaust gas temperature with LPG-RSO dual-fuel engine operation at different loads

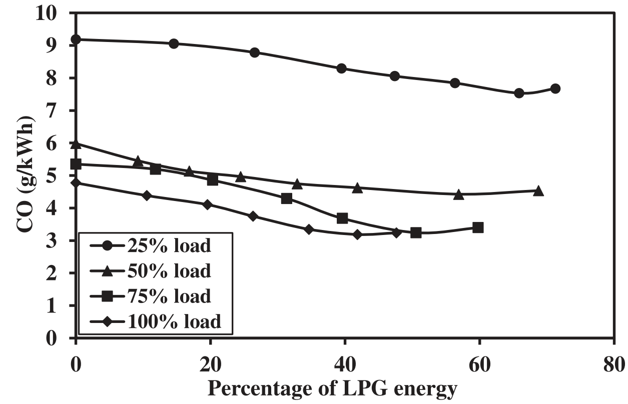

Fig. 5 Variation of CO emission with LPG-RSO dual-fuel engine operation at different loads

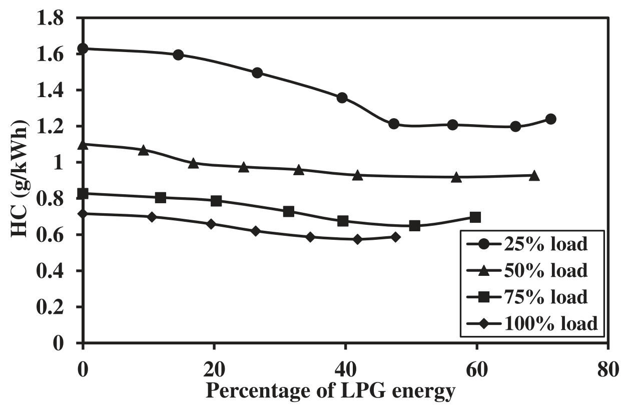

Fig. 4 Variation of unburned hydrocarbon emission with LPG-RSO dual-fuel engine operation at different loads

percentage of LPG energy share. The HC level falls from 0.82 g/kWh to 0.65 g/kWh at 75% load and from 0.71 g/kWh to lowest of 0.57 g/kWh at full load at LPG energy share rate of 50.5% and 41.84%, respectively. At higher loads, the injected LPG air mixture is sufficiently rich to enhance the combustion rate, thus enabling more complete oxidation of hydrocarbon. Lata et al. [16] used diesel as the pilot fuel and observed that at higher load conditions, LPG displaces the air in the intake process resulting in large consumption of oxygen and less oxygen is available for pilot fuel that leads to the increase in HC emission. However, in the present study, the pilot fuel is RSO which is an oxygenated fuel; hence, the deficiency of oxygen is compensated by RSO. The HC emission at full load with diesel operation is 0.5 g/kWh, which is a notch lower than maximum brake thermal efficiency point for LPG induction. At loads lower than 50% of full load, the increase in LPG admission decreases the hydrocarbon emissions. The hydrocarbon emission at 25% load is 1.19 g/kWh and at 50% load is 0.92 g/kWh with LPG energy share of 65.9% and 56.9% respectively.

Figure 5 shows the decrease in carbon monoxide emission with an increase in LPG energy share and load. At 25% load, CO emission reduces from 9.18 g/kWh to 7.53 g/kWh at LPG energy share of 65.9%. At 50% load, CO emission reduces from 5.98 g/kWh to 4.42 g/kWh with the LPG energy share of 56.9%. At 75% load, CO emission reduces from 5.35 g/kWh to 3.24 g/kWh with the LPG energy share of 50.5%. At 100% load, CO emission reduces from 4.77 g/kWh to lowest of 3.18 g/kWh with the LPG energy share of 41.84%. At full load, diesel engine operation resulted in 3.14 g/kWh CO emission, which is marginally lower than LPG induction value at maximum brake thermal efficiency point. LPG has a higher flame velocity compared with diesel, which helps in improving the oxidation rate of carbon monoxide. In addition, the increase in LPG share reduces the overall amount of carbon present in the charge. This reduces the pilot fuel quantity, which in turn reduces the carbon monoxide emission [27].

Figure 6 shows the effect of LPG induction rate on NOx emissions. At maximum BTE point, the NOx level is 9.76 g/kWh, which is lower than the diesel-fuelled engine operation (10.7 g/kWh) at full load. The NOx emissions decrease even though the exhaust gas temperature for LPG induction (Fig. 3) is higher than the diesel at the same point (Table 4). The NOx formation depends on the amount of oxygen available, which is low in this case because LPG replaces the part of the air. This reduces the NOx level and increases HC and CO level, as shown in Figs. 4 and 5, respectively. With an increase in the LPG energy share, the NOx emission increases at all the loads [28]. The NOx level increases from 6.9 g/kWh at no LPG induction to 10.37 g/kWh at 47.63% LPG energy share at full load. The enhanced combustion rate due to premixing of LPG increases the cycle temperature and increases the NOx emission. At 50% load and at 68.76% LPG energy share, the NOx emission is 12.16 g/kWh. At 25% load and at 71.28% LPG energy share, the maximum NOx value is 13.96 g/kWh.

The effect of LPG induction and engine load on the smoke level is shown in Fig. 7. The smoke level reduces drastically at all loads. Premixed charge combustion engines produce low smoke as the combustion takes place of a near homogeneous fuel-air mixture [17]. The smoke level, at full load, reduced from 6.1BSU to 3.9BSU at the highest possible energy share. This is due to the admission of a higher amount of gaseous fuel and the corresponding reduction in RSO. Also, LPG has lower carbon to hydrogen ratio [29]. In

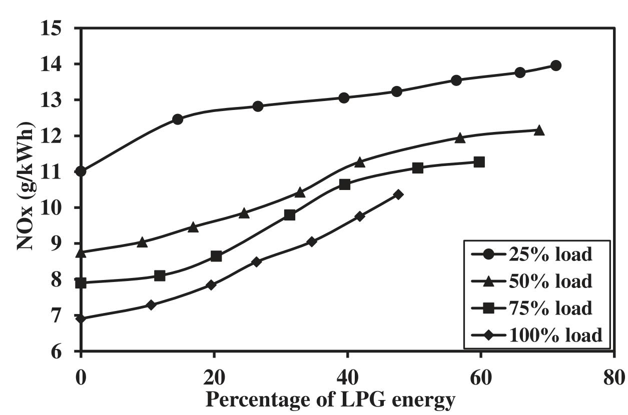

Fig. 6 Variation of NOx emission with LPG-RSO dual-fuel engine operation at different loads

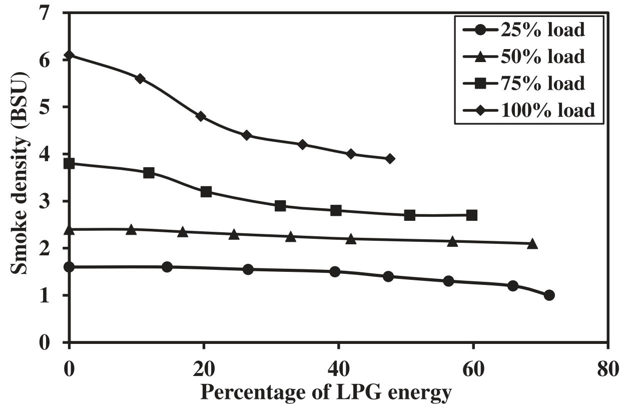

Fig. 7 Variation of smoke emission with LPG-RSO dual-fuel engine operation at different loads

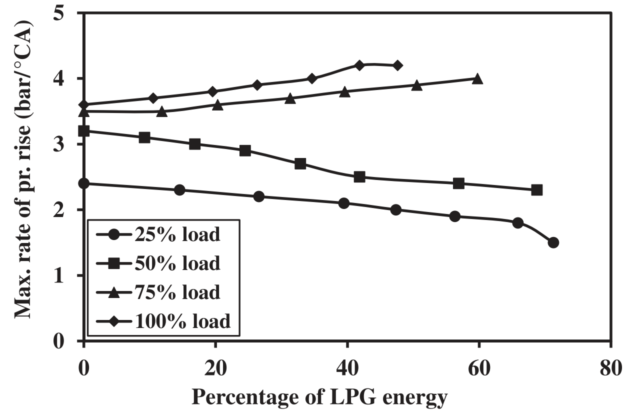

Fig. 9 Variation of the maximum rate of pressure rise with LPG-RSO dual-fuel engine operation at different loads

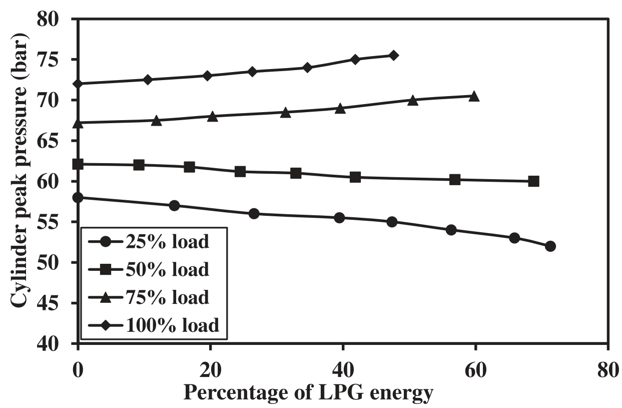

Fig. 8 Variation of cylinder peak pressure with LPG-RSO dual-fuel engine operation at different loads

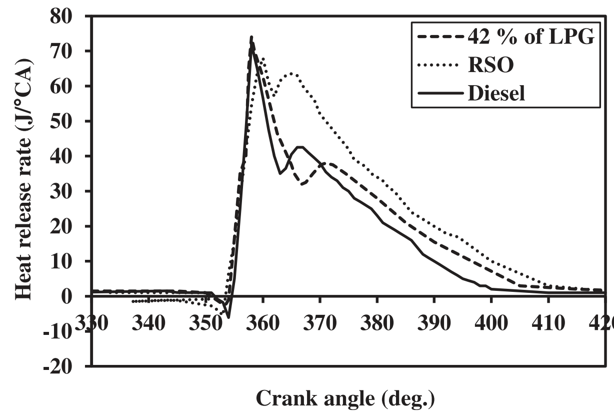

Fig. 10 Variation of heat release rate with LPG-RSO dual-fuel engine operation at full load

addition, the molecular weight of LPG is low and the carbon–carbon bond is smaller in number [30] resulting in lowering of smoke levels. At maximum BTE point, the smoke level is 4BSU, and with diesel-fuelled engine operation, the smoke level is 3.4BSU. RSO as pilot fuel has a high viscosity resulting in poor spray characteristics, and hence, some molecules are not able to participate in combustion resulting in higher smoke. At maximum energy share, smoke level observed at 25% load is 1BSU (lowest), at 50% load, it is 2.1BSU, and at 75% load, it is 2.7BSU.

Combustion Parameters. The peak pressure at full load, depicted in Fig. 8, increases as the amount of LPG inducted increases. The LPG entrained with the pilot fuel also burns in the initial combustion phase and increases the combustion rate. A similar result was obtained by Mohamed Selim [31]. The peak pressure observed at maximum BTE point is 75 bar. It is also observed that below 75% engine load, the peak pressure reduces significantly with the increase in LPG induction rate. The rate of pressure rise as seen in Fig. 9 shows similar trends. The pilot fuel quantity at low loads is less resulting in a lower number of ignition centers, which reduces the rate of pressure rise, and hence, the peak pressure is also reduced as the flame has to propagate a longer distance to consume the mixture [32]. When the LPG induction at low loads is very high, the combustion becomes sluggish resulting in the reduced peak pressure. At maximum energy share and 75%, 50%, and 25% load, the peak pressure is 70.5 bar, 60 bar, and 52 bar, respectively.

The heat release rate at full load with neat diesel, neat RSO, and 42% addition of LPG is depicted in Fig. 10. The heat release rate with diesel is high due to high-uncontrolled combustion phase. However, RSO has lower premixed combustion phase because of its high viscosity as it reduces the fuel and air mixing rate. Therefore, less fuel is prepared for the starting of combustion during ignition delay. The second peak with RSO engine operation indicates a higher diffusion combustion phase as more amount of fuel burns in this stage. With LPG induction, the premixed phase combustion rate increases as the accumulated fuel burns with the gaseous fuel. Moreover, the diffusion combustion rate gradually decreases, and the rate of burning is controlled by the rate of air and fuel-vapor mixing. Stewart et al. [33] and Goldsworthy [34] also observed the similar results.

Figure 11 shows that the ignition delay, at all loads, decreases with the increase in load, whereas with an increase in the LPG energy share, the delay increases. The primary fuel displaces some amount of air, and hence, the preflame reactions of the pilot fuel will be decelerated. In pre-ignition reaction, gaseous fuel also takes part unknowingly in the pre-ignition chemical process that affects combustion and increases ignition delay. Wattanavichien [32] also observed an increase in ignition delay with LPG induction. The author argued that due to the change in the specific heat of the mixture, the combustion temperature is lowered resulting in an increase in ignition delay. At full load, ignition delay increased from 8 °CA to 10 °CA with 47.63% energy substitution of LPG. At 25% load, the ignition delay increased from 13 °CA to 17 °CA with 71.28% LPG energy share, since the mixture formed is rich due to the induction of large amount of LPG. The ignition characteristics and the temperature of the charge at low loads cannot improve due to the low rate of energy release.

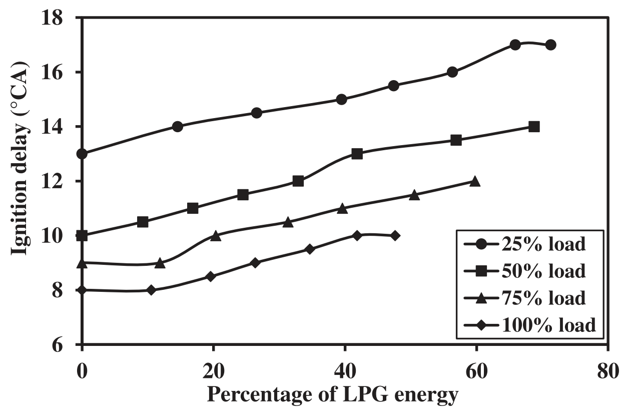

Fig. 11 Variation of ignition delay with LPG-RSO dual-fuel engine operation at different loads

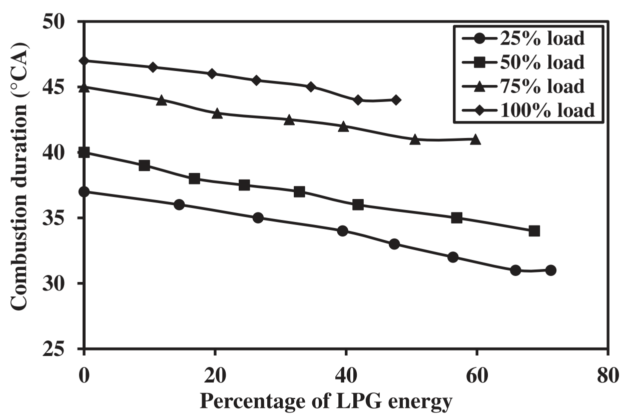

Fig. 12 Variation of combustion duration with LPG-RSO dual-fuel engine operation at different loads

The duration of combustion decreases as the LPG rate increases, shown in Fig. 12. At high loads, as the flame velocity of gaseous LPG is high, the combustion duration is less. Moreover, the amount of pilot fuel injected is also high enhancing the number of ignition centers and increasing the rate of combustion [14]. Figure 10 clearly depicts the improvement in heat release caused by LPG induction, the brake thermal efficiency also improves at high loads. The combustion duration at maximum brake thermal efficiency point is 44 °CA, which is 3 °CA higher than the diesel operation at full load. At low loads, combustion duration decreases with increase in the percentage of LPG up to a particular level and after that it increases. Reduced combustion rate due to the lower amount of pilot fuel injection is the probable reason for the trend [17]. At 25% load and 71.28% LPG energy share, the combustion duration is 31 °CA.

Conclusion

Based on results presented in the earlier sections, the following conclusions may be drawn:

- BTE at all loads with neat RSO is lower than diesel. LPG as a secondary fuel enhances the BTE of RSO at high loads, whereas it reduces the BTE at low loads. Maximum BTE observed is 28% at full load with 41.84% LPG energy share. Lowest efficiency obtained is 11.9% at 25% load with 71.28% LPG energy share.
- The maximum LPG energy share at a full load that the engine can tolerate without knocking was maximum with diesel (59%) and it decreases to 49% with RSO operation.
- Unburnt carbon monoxide and hydrocarbon emission are higher with neat RSO operation. The introduction of LPG in intake manifold reduces both the emissions at all loads. This is due to the improvement in the combustion rate. At maximum BTE point, CO and HC emissions reduced by 33.3% and 19.8%, respectively. These values are similar to diesel fuel operation at full load, without LPG induction.
- Neat RSO operation resulted in lower NOx emission than neat diesel operation. On induction of LPG, NOx emission increased as the LPG energy share increased at all loads. NOx emission at full load increased from 6.9 g/kWh with no premixing of LPG to 10.36 g/kWh with 47.63% of LPG energy share. This can be attributed to an increase in combustion temperature caused by high-premixed combustion.
- Smoke emissions are high with neat RSO operation at all loads. LPG induction reduced the smoke emissions. At full load, it reduced from 6.1BSU for no induction of LPG to 4BSU at maximum BTE point. This is due to a reduction in the amount of pilot fuel injected.
- The maximum rate of pressure rise and peak pressure at high loads increases with LPG induction. At low loads, the maximum rate of pressure rise and peak pressure reduces with LPG energy share because of weak ignition sources.
- The delay in ignition increases with LPG induction at all loads. It increases from 8 °CA to 10 °CA at maximum BTE point with LPG. The increase in ignition delay at initial stages results in high heat release.
- The combustion duration at all the loads decreases as the LPG energy share increases. It reduces from 47 °CA to 44 °CA at maximum BTE point with LPG. Fast diffusion combustion causes a decrease in combustion duration with LPG induction as compared with neat RSO.
- Premixed charge heat release rate is significantly different from neat RSO engine operation. The premixed burning rate is higher as both the accumulated injected fuel and the entrained LPG are burnt, whereas in neat RSO, the diffusion combustion is high as fuel cannot ignite properly in initial stages of combustion.

Thus, premixed LPG can be used to improve the performance of rubber seed oil-fuelled diesel engine. This method effectively improves the BTE and reduces the HC, CO, and smoke emission. NOx emission is a major drawback with LPG induction, which can be taken care by using exhaust gas recirculation.

Nomenclature

BSEC = brake specific energy consumption
BSU = Bosch Smoke Unit
BTE = brake thermal energy
CI = compression ignition
CO = carbon monoxide
DEE = di-ethyl ether
EGT = exhaust gas temperature
HC = unburned hydrocarbons
LPG = liquefied petroleum gas
NOx = oxides of nitrogen
RSO = rubber seed oil

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