# PERFORMANCE, EMISSION, AND COMBUSTION CHARACTERISTICS OF A CI ENGINE USING LIQUID PETROLEUM GAS AND NEEM OIL IN DUAL FUEL MODE

by

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*Increased environmental awareness and depletion of resources are driving the industries to develop viable alternative fuels like vegetable oils, compresed natural gas, liquid petroleum gas, producer gas, and biogas in order to provide suitable substitute to diesel for compression ignition engine. In this investigation, a single cylinder, vertical, air-cooled diesel engine was modified to use liquid petroleum gas in dual fuel mode. The liquefied petroleum gas, was mixed with air and supplied through intake manifold. The liquid fuel neem oil or diesel was injected into the combustion chamber. The performance, emission, and combustion characteristics were studied and compared for neat fuel and dual fuel mode. The experimental results on dual fuel engine show a reduction in oxides of nitrogen up to 70% of the rated power and smoke in the entire power range. However the brake thermal efficiency was found decreased in low power range due to lower calorific value of liquid petroleum gas, and increase in higher power range due to the complete burning of liquid petroleum gas. Hydrocarbon and carbon monoxide emissions were increased significantly at lower power range and marginal variation in higher power range.*

Key words: *liquid petroleum gas, neem oil, diesel, dual fuel, combustion, performance,emission.*

## 1. Introduction

Use of diesel in a compression ignition engine is a well-proven technology. The use of plant oil as fuel for compression ignition engine is not new. Dr. Rudolf Diesel (inventor of diesel engine) demonstrated his engine in Paris in 1900 using groundnut oil as fuel. The plant oil fuels were not accepted much at that time. It was found that all the properties of plant oils were close to diesel except viscosity and volatility. Diesel could be replaced by the plant oil with satisfactory engine performance [1]. Various plant oils such as honge, rice bran, and neem oils are being investigated for their suitability as diesel engine fuel in neat fuel and dual fuel mode [2-4].

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Available technologies for reciprocating internal combustion engines are generally divided in two categories: compression ignition (CI) and spark ignition (SI) engines. In CI engines (diesel engines), air is compressed at pressures and temperatures at which the injected liquid fuel fires easily and burns progressively after ignition. Whereas, SI engines (Otto engines) that runs according to the Beau de Rochas cycle [5], the carburetted mixture of air and vaporized fuel (high octane index) is compressed under its ignition point and then fired at a chosen instant by an independent means. In a dual-fuel engine, both types of above combustion coexist together, *i. e.* a carburetted mixture of air and high octane index gaseous fuel is compressed like in a conventional diesel engine. The compressed mixture of air and gaseous fuel does not auto-ignite due to its high auto-ignition temperature. Hence, it is fired by a small liquid fuel injection which ignites spontaneously at the end of compression phase. The advantage of this type of engine is that, it uses the difference of flammability of two used fuels. Again, in case of lack of gaseous fuel, this engine runs according to the diesel cycle by switching from dual-fuel mode. The disadvantage is the necessity to have liquid diesel fuel available for the dual-fuel engine operation [6].

Liquid petroleum gas (LPG) vehicles are being rapidly developed as an economical and a low pollution car [7, 8]. The potential benefits of using LPG in diesel engines are both economical and environmental [9]. In dual fuel gas engines, the gaseous fuel is inducted along with the air and this mixture of air and gas is compressed like in conventional diesel engines. A small amount of diesel, usually called the pilot, is sprayed near the end of the compression stroke to initiate the combustion of the inducted gas air mixture [10, 11]. With reduced energy consumption, the dual fuel engine shows a significant reduction in smoke density, oxides of nitrogen emission, and improved brake thermal efficiency [12]. The combustion of this pilot diesel leads to flame propagation and combustion of the gaseous fuel. The engine can be run in the dual fuel mode without any major modification, but is usually associated with poor brake thermal efficiency and high HC and CO emissions at low loads [13].

Many researchers have performed tests on CI engine with different vegetable oils. The botanical name of neem oils is *Azadirachta indica*. It is extracted from the seeds of neem tree. Higher viscosity, lower volatility, and polyunsaturated character of neat vegetable oils pose serious engine operational problems. Several researchers have undertaken works related to conservation of diesel fuel in dual fuel engines using various vegetable oils. Dual fuel combinations of producer gas–honge oil, producer gas–rice bran oil, and producer gas–neem oil have been extensively studied. From the point of view of fuel flexibility, dual fuel engines are highly acceptable. Modification of existing diesel engine for dual fuel operation with producer gas is simple [14].

The objective of the present work is to study the performance, emission, and combustion of diesel and neem oil in single fuel mode and in combinations with LPG:

diesel (100%) CI engine power
neem oil (100%) CI Engine power
diesel 7.6 mg/cycle rest LPG CI engine power
neem oil 7.6 mg/cycle rest LPG CI engine power

## Fuel properties

The various properties studied were kinematic viscosity, density, calorific value, flash point, and cetane number are summarized in tab. 1.

**Table 1. Fuel properties**

| Properties | Diesel | Neem oil | LPG |
|---|---|---|---|
| Kinematic viscosity at 40 °C [cSt] | 2.6 | 21.5 | – |
| Density [kgm$^{-3}$] | 830 | 910 | – |
| Calorific value [MJkg$^{-1}$] | 42.5 | 39.5 | 47.7 |
| Flash point [°C] | 50 | 203 | – |
| Cetane number | 40 to 60 | 47 | <3 |

## Experimental set-up and experimentation

A single cylinder, 3.7 kW, four stroke, direct injection (DI), air-cooled diesel engine coupled to an electrical dynamometer was used for the experiments. The specifications of the DI diesel engine are shown in tab. 2.

**Table 2. Specifications of the test engine**

| | |
|---|---|
| General detail | Single cylinder, four stroke, compression ignition, constant speed, vertical, air cooled, direct injection |
| Bore | 80 mm |
| Stroke | 110 mm |
| Swept volume | 553 cm$^3$ |
| Clearance volume | 36.86 cm$^3$ |
| Compression ratio | 16.5:1 |
| Rated output | 3.7 kW at 1500 rpm |
| Rated speed | 1500 rpm |
| Opening pressure of injection nozzle | 200 bar |
| Fuel injection timing | 24° bTDC(neat fuel) ; 27° bTDC (dual fuel) |
| Type of combustion chamber | Hemispherical open combustion chamber |
| Connecting rod length | 235 mm |
| Lubricating oil | SAE 40 |

The engine was modified to work in the dual fuel mode by connecting LPG line to the intake manifold with a flame trap, non-return valve, needle valve, and mixing unit [13]. A digital type platform weighing machine having an accuracy of 2 mg was used to measure the LPG fuel flow by difference in weight method. Time taken for fuel consumption was measured with a help of a digital stopwatch. An orifice meter was used to measure air consumption of the engine with the help of a U tube manometer. The surge tank fixed on the inlet side of an engine maintains a constant airflow through the orifice meter. CO and unburned HC emissions were measured using a NDIR gas analyser. $NO_x$ emissions from the engine were measured using a Crypton make analyser. Smoke emissions were measured by means of a Diesel tune smoke meter. Two separate sampling probes were used to receive sample exhaust gas from the engine for measuring emission and smoke intensity. The technical specifications of emission analyser are shown in tab. 3. Chromel-alumel (K-type) thermocouple was used to measure the exhaust gas temperature with a digital temperature indicator. The brake thermal efficiency was calculated by considering the calorific value and mass flow rate of both fuels. A Kistler make piezo electric transducer with a sensitivity of 14.2 pC/bar was installed with a Kistler charge amplifier for monitoring the cylinder pressure. A crank angle (CA) encoder was used to sense the top dead centre position. Output from the CA encoder and pressure transducer was recorded in a personal computer through A/D converter.

$$\text{Brake thermal efficiency} = \frac{\text{brake power}}{(m_\text{f} CV)_\text{LPG} \quad (m_f CV)_\text{diesel(or)neem}}$$

**Table 3. Technical specifications of emission analyser**

| Measuring parameter | Range | Resolution | Accuracy |
|---|---|---|---|
| Carbon monoxide | 0-10% | 0.01% | 0.01% |
| Hydro carbon | 0-10000 ppm | 1 ppm vol. | 1 ppm |
| Carbon dioxide | 0-20% | 0.1% vol. | 0.1% |
| Oxygen | 0-25% | 0.01% vol. | 0.01% |
| Nitric oxide | 0-5000 ppm | 1 ppm | 12 ppm |
| Engine rpm | 0-10000 with 2 /4 stroke selection | 1/minute | 10 |
| Oil temperature | 0-120 °C | 1 °C | 5° |

The schematic representation of the experimental set-up is shown in fig. 1.

The experimental procedure consists of the following steps.

- Initially engine was tested using the base fuel diesel at all loads to determine the engine operating characteristics and pollutant emissions. The engine speed was maintained constant throughout the entire engine operation at 1500 rpm.
- The engine was tested using neat neem oil for the same operating condition.
- The same procedure was repeated in dual fuel mode with an optimum pilot quantity of 7.6 mg/cycle diesel with LPG [15] and 7.6 mg/cycle neem oil with LPG. The injection timing was advanced 3 °CA for dual fuel operation (27° bTDC). The liquid fuel quantity was maintained constantly for the entire load range by varying the flow rate of LPG for each load condition. The mass fraction of LPG in the blend ($Z_1$) and neem oil in the blend ($Z_2$) is shown in tab. 4. $Z_1$ and $Z_2$ are defined as:

$$Z_1 \quad \frac{\dot{m}_\text{LPG}}{\dot{m}_\text{diesel} \quad \dot{m}_\text{LPG}} 100\%$$

$$Z_2 \quad \frac{\dot{m}_\text{LPG}}{\dot{m}_\text{neem} \quad \dot{m}_\text{LPG}} 100\%$$

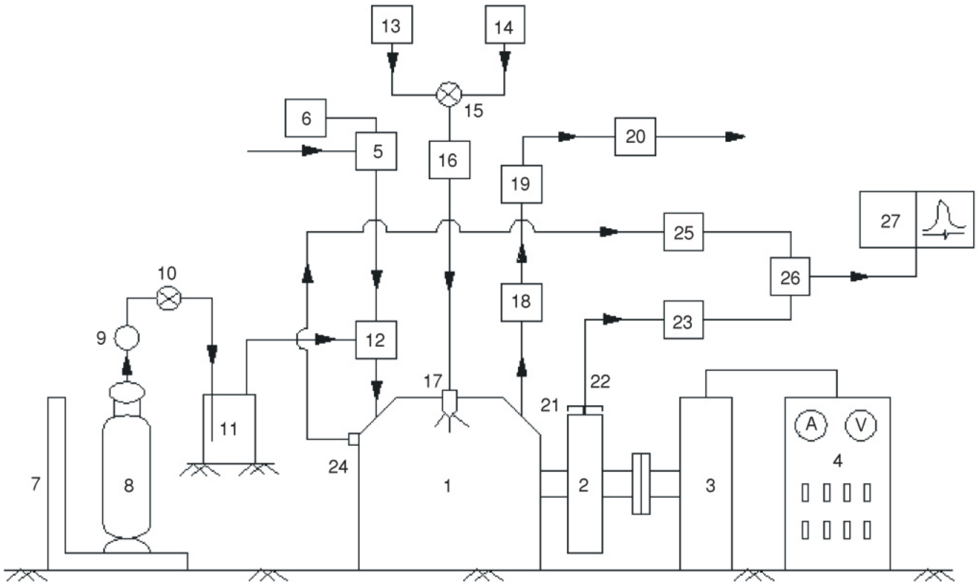


**Figure 1. Schematic of the experimental set-up**
*1 – Engine, 2 – Flywheel, 3 – Dynamometer, 4 – Control panel, 5 – Air box, 6 – U-tube manometer, 7 – Weighing machine, 8 – LPG pressure tank, 9 – Pressure regulator, 10 – Flow control valve, 11 – Flame trap, 12 – Mixing unit, 13 – Diesel tank, 14 – Biodiesel tank, 15 – Three way valve, 16 – Fuel measurement, 17 – Injector, 18 – Temperature measurement, 19 – Exhaust gas analyser, 20 – Smoke meter, 21 – TDC marker, 22 – TDC position encoder, 23 – TDC amplifier circuit, 24 – Pressure pickup, 25 – Charge amplifier, 26 – A/D card, 27 – Personal computer*

**Table 4. Mass fraction of LPG in the blend**

| Load [%] | 0 | 20 | 40 | 60 | 80 | 100 |
|---|---|---|---|---|---|---|
| Neem with LPG ($Z_1$) | 33.8 | 47.8 | 56.8 | 61.9 | 66.5 | 68.9 |
| Diesel with LPG ($Z_2$) | 29.3 | 32.8 | 48.5 | 54.8 | 60.4 | 63.6 |

During the engine test conditions the cylinder pressure, exhaust gas temperature, mass flow rate of air, fuel consumption, exhaust smoke, and exhaust gas emissions were recorded at all the loads.

## Error analysis

Errors and uncertainties in the experiments can arise from instrument selection, condition, calibration, environment, observation, reading, and test planning. Uncertainty analysis is needed to prove the accuracy of the experiments. An uncertainty analysis was performed using the method described by Holman [16].

Percentage uncertainties of various parameters like total fuel consumption, brake power; specific fuel consumption, and brake thermal efficiency were calculated using the percentage uncertainties of various instruments given in tab. 5. An uncertainty analysis was performed using the equation:

$$Y \quad \sqrt{X_1^2 \quad X_2^2 \quad X_3^2 \quad X_4^2 \quad X_5^2 \quad X_6^2 \quad X_7^2 \quad X_8^2 \quad X_9^2 \quad X_{10}^2}$$

where $Y$ is the total percentages uncertainty, $X_1$ – the uncertainty of total fuel consumption, $X_2$ – the uncertainty of brake power, $X_3$ – the uncertainty of specific fuel consumption, $X_4$ – the uncertainty of brake thermal efficiency, $X_5$ – the uncertainty of CO, $X_6$ – the uncertainty of unburned hydrocarbon, $X_7$ – the uncertainty of $NO_x$, $X_8$ – the uncertainty of smoke number, $X_9$ – the uncertainty of exhaust gas temperature, and $X_{10}$ – the uncertainty of pressure pickup.

Total percentage uncertainty ($Y$) of this experiment is:

$$Y \quad \sqrt{(1)^2 \quad (0.2)^2 \quad (1)^2 \quad (1)^2 \quad (0.12)^2 \quad (0.12)^2 \quad (0.2)^2 \quad (1)^2 \quad (0.15)^2 \quad (1)^2} \quad 2.6\%$$

The total uncertainty for the whole experiment is obtained to be 2.6%

**Table 5. List of instruments and its range, accuracy, and percentage uncertainties**

| Instruments | Range | Accuracy | Percentage uncertainties |
|---|---|---|---|
| Smoke level measuring instrument (Diesel tune DX230) | BSU 0-10 m$^{-1}$ | +0.1 to –0.1 | +1 to –1 |
| Exhaust gas temperature indicator | 0-900 °C | +1 °C to –1°C | +0.15 to –0.15 |
| Speed measuring unit | 0-10000 rpm | +10 rpm to –10 rpm | +0.1 to –0.1 |
| Burette for fuel measurement | – | +0.1cm$^3$ to –0.1cm$^3$ | +1 to –1 |
| Digital stop watch | – | +0.6 s to –0.6 s | +0.2 to –0.2 |
| Manometer | – | +1 mm to –1 mm | +1 to –1 |
| Pressure pickup | 0-110 bar | +0.1 kg to –0.1 kg | +0.1 to –0.1 |
| CA encoder | – | +1° to –1° | +0.2 to –0.2 |

## Results and discussion

The results of dual fuel operation of neem -LPG, diesel-LPG are compared to neat neem oil and diesel are presented.

### *Combustion analysis*

#### *Pressure CA diagram*

Figure 2 shows the variation of cylinder pressure with CA for neat fuel and dual fuel mode at 20% of full load. Peak pressure of a CI engine depends on the combustion rate in the initial stages, which is influenced by the amount of fuel burnt in the premixed combustion. The premixed combustion is dependant on the delay period and the mixture preparation [5]. The peak pressure of diesel is high and it is followed by neem oil, diesel-LPG, and neem-LPG. The peak pressure values are 62 bars, 59 bar, 51 bar, and 47 bar, respectively. In the dual fuel mode at low loads there is a drop in peak pressure. At low loads, the gaseous fuel air mixture is lean and ignition source is also weak. This will result in slower combustion rate and lower rate of pressure rise. This is also believed to be one of the reasons for the reduced brake thermal efficiency at this condition [17].

The cylinder pressure obtained at full load is high for dual fuel operation compared to neat fuel operation. Figure 3 shows the cylinder pressure with CA for neat fuel and dual fuel mode at full load. As load increases, the mixture temperature will rise. This will lead to reduction in the ignition delay. The gas to air ratio also increases with load when pilot quantity is constant. This will lead to increase in the ignition delay. Both these effects counter each other and it seems that the ignition delay gets affected more by the gas to air ratio of the inducted mixture [18]. Longer ignition delay at high load range increases the peak pressure in dual fuel mode than that of neat fuel. The peak pressures of diesel-LPG and neem-LPG are 72 bar and 66 bar, respectively. This is high compared to the neat fuel operation of diesel, neem oil, whose values are 69 bar and 63 bar, respectively. At higher loads, the introduction of LPG leads to rapid combustion and high cylinder pressure and temperature. Rapid combustion occurs due to rise in the flame propagation rate in the LPG air mixture. This is also the reason for high peak pressure in dual fuel operation.

### *Cylinder peak pressure*

The cylinder peak pressure variation with brake power is shown in fig. 4. In the dual fuel operation there is a drop in peak pressure in case of low outputs. This is due to weakening of the ignition source, as a result of lean gas air mixture. The peak pressure of dual fuel operation increases with brake power

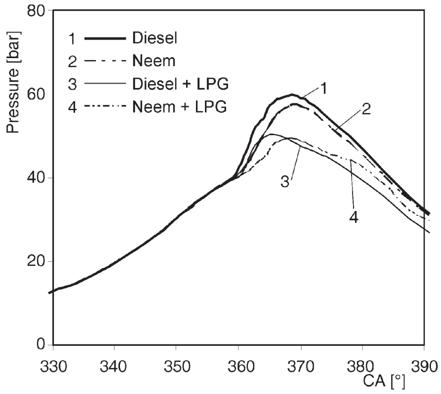


**Figure 2. Variation of pressure with CA at 20% load**

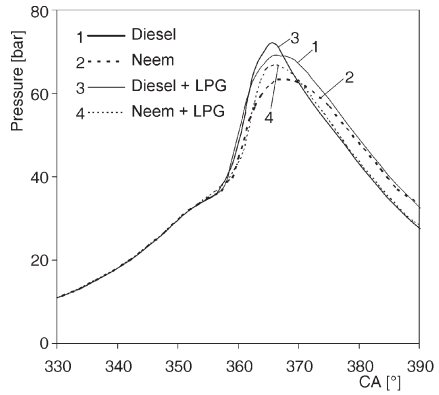


**Figure 3. Variation of pressure with CA at full load**

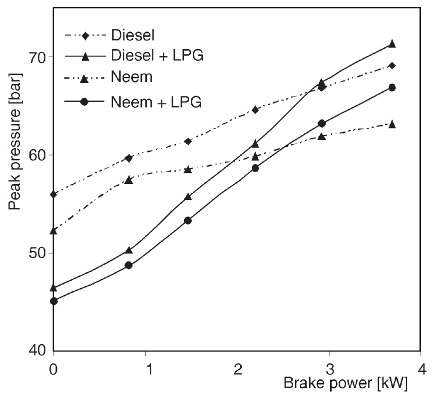


**Figure 4. Variation of peak pressure with brake power**

and it is high in high power range compared with neat fuel operation. An increase in diesel substitution leads to relatively rich gas air mixture which causes high rate of pressure rise in the maximum power range.

### *Heat release rate*

The rate of heat release curves are drawn using pressure and CA value in the existing software.

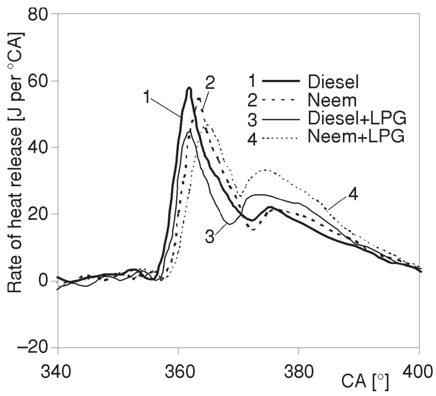


**Figure 5. Variation of heat release rate with CA at 20% load**

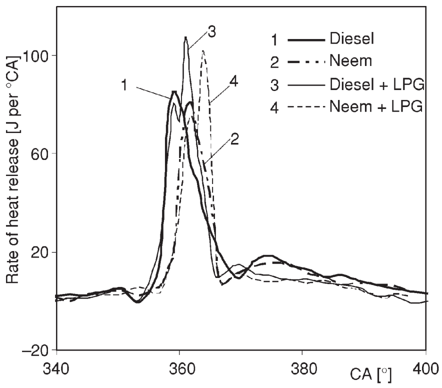


**Figure 6. Variation of heat release rate with CA at full load**

The heat release rate of the neat fuel and dual fuel mode at 20% of full load are given in fig. 5. The first stage starts from the start of ignition to the point where the heat release rate drops and this is depends on the ignition of fuel air mixture prepared during the delay period. The second stage starts from the end of the first stage to the end of combustion [5]. The maximum heat release rate is 45 J/°CA for diesel-LPG and 46 J/°CA for neem-LPG whereas 58 J/°CA for diesel and 54 J/°CA for neem. The heat release rate during the first stage decreases in the dual fuel operation because the amount of fuel prepared for burning during the delay period decreases and also the contribution of gaseous fuel to heat release during this stage is not significant. This behaviour is observed at light loads only with lean gas-air mixture. The second stage of combustion is also sluggish, because the source of ignition for the gas-air mixture becomes weak.

Diesel shows the lowest heat release rate at initial stage and longer combustion duration at rated power. The heat release rate of the neat fuel and dual fuel mode at full load is given in fig. 6. The maximum heat released for dual fuel operation is high compared to neat fuel. It can be noticed that in dual fuel operation, most of the heat release occurs only during the premixed combustion. Longer ignition delay results in higher heat release during the premixed combustion phase. The higher heat release rate lead to an increase in exhaust gas temperature. The maximum heat release rate is 107 J/°CA for diesel-LPG and 101 J/°CA for neem oil-LPG.

## ***Performance analysis***

### *Brake thermal efficiency*

The variation of brake thermal efficiency with brake power is shown in fig. 7. It is observed that in dual fuel mode, the brake thermal efficiency is less at low power range and it significantly increases in high power range than neat fuel. In dual fuel operation, the insufficient ignition source for gaseous fuel leads to a drop in efficiency at low power range. The brake thermal efficiency of neem-LPG and diesel-LPG at 20% of rated power are 9.4% and 10.8%, respectively. The poor performance of gas engine at low power range is due to the effect of gas residuals and low cylinder temperature. It is also due to the reduction in combustion efficiency caused by reduced flame propagation speed and increased compression work resulting from the large amount of air-gas inducted [2]. The complete burning of LPG at elevated temperature promotes the efficiency at higher power range. The brake thermal efficiency of neem-LPG and diesel-LPG at maximum power are 25.6% and 28.6%, respectively. Neat neem oil shows a lower efficiency in the entire power range than diesel, due to the lower calorific value of neem oil than diesel. The brake thermal efficiency of neat neem oil and diesel at the maximum power are 23.2% and 26.7%, respectively.

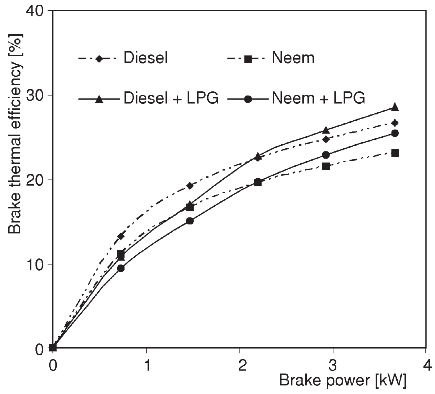


**Figure 7. Variation of brake thermal efficiency with brake power**

### *Exhaust gas temperature*

Figure 8 shows the variation of exhaust gas temperature with brake power. A marginal increase in exhaust gas temperature is noticed throughout the dual fuel mode of operation. The exhaust gas temperature varies from 271 °C at no load to 602 °C at rated power for neem-LPG, whereas in the case of diesel-LPG, it varies from 262 °C to 589 °C. The increase in exhaust gas temperature with engine load is obvious from the simple fact that more amount of fuel was required by the engine to generate extra power needed to take up the additional loading. The uncontrolled combustion, late burning of fuel air mixture, higher heat release rate, and steep increase in mean gas temperature are the major reasons for higher exhaust gas temperature in dual fuel operation. The exhaust gas temperature varies from 231 °C at no load to 581 °C at rated power for neem oil, whereas in the case of diesel, it varies from 224 °C to 571 °C. The exhaust gas temperature is not affected significantly in the neem oil compared to diesel.

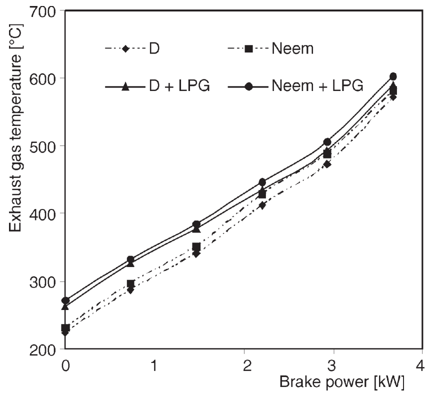


**Figure 8. Variation of exhaust gas temperature with brake power**

## *Emission parameters*

### *HC and CO emissions*

Unburnt HC emissions are the direct result of incomplete combustion. The term HC means organic compounds via gaseous state. Solid HC are the particulate matter. Figure 9

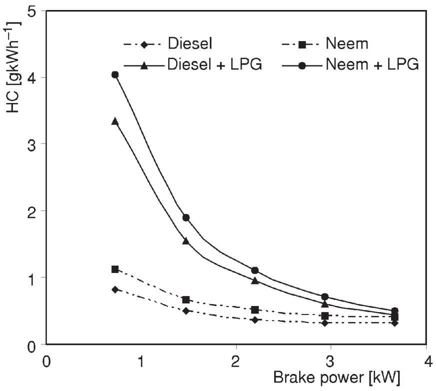


Figure 9. Variation of HC with brake power

shows the variation of hydrocarbon emission against brake power. The hydrocarbon emission is high in dual fuel mode compared with neat fuel mode in the entire power range. The difference in HC emission is high in low power range and narrow in high power range. Incomplete combustion of LPG due to insufficient ignition sources in the low power range is the reason for high HC emission. The emission range in the dual fuel mode of neem-LPG and deisel-LPG at 20% of maximum power are 4.0 g/kWh and 3.3 g/kWh, respectively. At low loads, the inducted LPG air mixture is too lean and this causes the flame to quench near the combustion chamber walls which leads to high HC emission. As the load increases, the HC level reduces. This is because combustion is complete due to high gas temperature in the cylinder. At the maximum power, neem-LPG, and diesel-LPG emissions are 0.51 g/kWh and 0.44 g/kWh, respectively. The variation is narrow between the neat fuels throughout the power range of 0.40 g/kWh and 0.32 g/kWh for neem and diesel, respectively, at maximum power range. The neem oil shows slightly higher emissions than diesel due to the higher viscosity of neem oil which results in improper atomization.

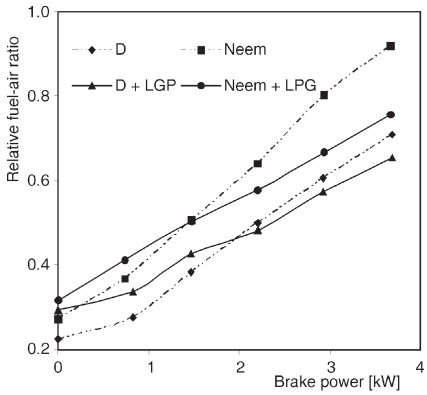


Figure 10. Variation of relative fuel-air ratio with brake power

The variation of relative fuel air ratio with brake power is shown in fig. 10. It can be seen that the fuel air ratio increases with brake power for all the cases. In dual fuel operation, the variation is high in low power range compared with neat fuel operation, but it is less in high power range. At light load, when very lean gaseous fuel-air mixtures are used, significant proportion of the low concentration of the gaseous fuel will not burn completely, this is despite the presence of much excess air leads more HC emissions [18]. The ignition of the diesel pilot releasing energy and raising the effective temperature locally cannot permit widespread flame propagation. A flame starting from any pilot ignition centre quenched quickly leaving some of the gaseous fuel unoxidized leads to a significant increase in CO emission.

The CO is a product of incomplete combustion due to insufficient amount of air in the air-fuel mixture or insufficient time in the cycle for completion of combustion. CO emission is toxic and must be controlled. Generally CI engines operate with lean mixtures and hence CO emission will be low. The variation of CO emissions with brake power is shown in fig. 11. The CO emission is increased throughout the engine operation in dual fuel mode compared with neat fuel mode. The variation is more in low power range and it shows a slight difference in high power range. The CO emission greatly depends on the air fuel ratio. The LPG was inducted through the inlet manifold, the oxygen availability in the intake mixture was less, which results in higher emission. The emission range for neem-LPG, diesel-LPG at 20%

of rated power is 68.4 g/kWh and 58.6 g/kWh, and at the maximum power – 18.4 g/kWh and 14.6 g/kWh, respectively. At high loads, the increase in CO level is not as high as compared to the light loads operation. In general the trend of CO emission follows the same pattern of HC emissions. The variation is narrow between the neat fuels and it is 13.3 g/kWh and 11.0 g/kWh for neem oil and diesel, respectively, at maximum power range.

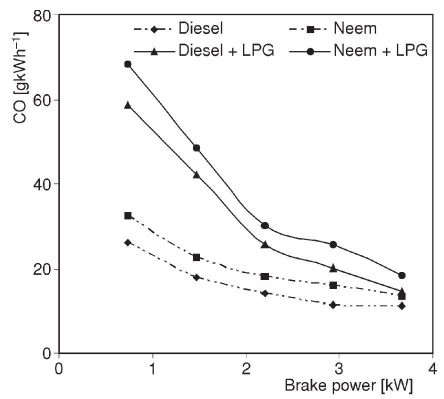


**Figure 11. Variation of CO with brake power**

## *$NO_x$ emission*

The oxides of nitrogen ($NO_x$) occur in the engine exhaust are the combination of nitric oxide (NO) and nitrogen dioxide ($NO_2$). Nitrogen and oxygen react at relatively high temperature. Therefore high temperature and availability of oxygen are the two main reasons for the formation of $NO_x$ (14). The variation of $NO_x$ emission with brake power is shown in fig. 12. $NO_x$ level is reduced in dual fuel mode up to 70% of output. This is because of reduction in the combustion rate, lower peak pressure and insufficient amount of oxygen in the air fuel mixture. At higher loads high amount of LPG leads to rapid combustion, high cylinder pressure and temperature increases $NO_x$ emission marginally. The emission ranges for neem-LPG is between 4.9 g/kWh and 6.4 g/kWh, and diesel-LPG is between 4.7 g/kWh and 6.8 g/kWh. The variation is narrow between the neat fuels and it shows the maximum value of 7.7 g/kWh and 8.2 g/kW at 20% of rated power and 5.9 g/kWh and 6.4 g/kW at rated power for neem oil and diesel, respectively.

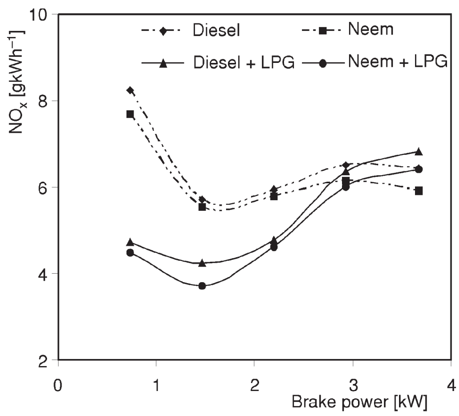


**Figure 12. Variation of $NO_x$ with brake power**

## *Smoke*

The effect of brake power on smoke emissions is shown in fig. 13. The smoke was reduced drastically in the dual fuel mode. The reduction of smoke is due to lower carbon/hydrogen ratio of LPG and the availability of premixed, homogeneous charge inside the engine well before the commencement of combustion. Higher combustion temperature and rapid flame propagation

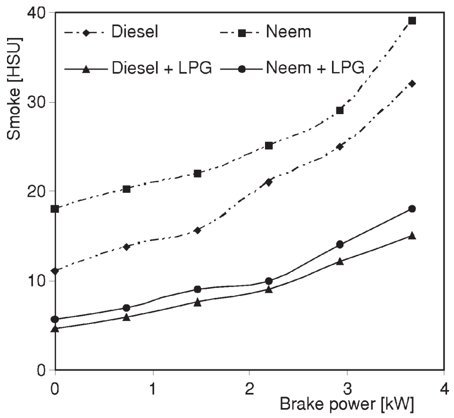


**Figure 13. Variation of smoke with brake power**

also facilitates the reduction in smoke. The emission ranges for neem-LPG is between 5.6 HSU and 18 HSU and of diesel-LPG is between 4.5 HSU and 15 HSU. The neat RBO emits slightly higher smoke than neat diesel due to its heavier molecular structure. It is 39 HSU, and 32 HSU for neem oil and diesel, respectively, at rated power. The neem oil shows the poor combustion in both single and dual fuel mode. This is due to poor atomization and mixture preparation with air, resulting in higher smoke compared to diesel.

## Conclusions

The following conclusions are drawn based on the experimental investigation

The performance study of CI engine operated on diesel and neem oil in single fuel mode and with LPG in dual fuel mode shows no major modification required in an existing diesel engine.

The peak pressures were lower in the dual fuel operation up to 70% load than neat fuel operation; it shows marginal increase at full loads. Similar trends have been observed in peak heat release rate.

The brake thermal efficiency of engine with single fuel operation is high at low power ranges and dual fuel operation is high at high power range. The maximum brake thermal efficiency for diesel-LPG, neem-LPG, neem oil, and diesel are 28.6%, 25.6%, 23.2%, and 26.7%, respectively.

HC and CO emissions of dual fuel operation were found to be more than single fuel operation.

The $NO_x$ emissions were reduced up to 70% of the rated power and smoke emissions were reduced by 62% in the entire power range in the dual fuel mode of operation.

In general dual fuel engine improves the performance in full load with a significant reduction in smoke emission to the entire load range and $NO_x$ emission up to 70% of rated power.

## Nomenclature

$\dot{m}_{\mathrm{diesel}}$ – mass flow rate of diesel, [kgs$^{-1}$]
$mf$ – mss of the fuel, [kgs$^{-1}$]
$\dot{m}_{\mathrm{LPG}}$ – mass flow rate of LPG, [kgs$^{-1}$]
$\dot{m}_{\mathrm{neem}}$ – mass flow rate of neem oil, [kgs$^{-1}$]

*Acronims*

*A/D* – analog to digital
bTDC – before top dead centre
CA – crank angle
CI – compression ignition engine
CNG – compressed natural gas
CV – calorific value, [MJkg$^{-1}$]
HSU – Hartridge smoke unit
LPG – liquid petroleum gas
rpm – revolutions per minute
SI – spark ignition engine

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