Effect on Burning Period of LPG based dual fuel diesel engine supplemented by hydrogen

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Abstract

This paper presents experimental investigations on dual fuel operation of a 4 cylinder (turbocharged and intercooled) 62.5 kW gen-set diesel engine. The detailed investigations have been carried out with, liquefied petroleum gas (LPG) and mixture of LPG and hydrogen used as secondary fuels. The paper is projected to present crucial information regarding Burning period of the dual fuel diesel engine at a wide range of load conditions with different gaseous fuel substitutions. When only LPG is used as 30% of secondary fuel, the Burning period is increased by 5 ^{0}CA at 80% load condition. The major finding of the present work is major improvement in combustion parameters of dual fuel engine when hydrogen-LPG mixture is used as the secondary fuel. The best part of this Trial is that when 40% of mixture of LPG and hydrogen (7:3) is used as secondary fuel, burning period reduces by 4 ^{0}CA as compare to pure diesel operation.

Keywords: Burning period, dual fuel engine, hydrogen, LPG.

1. Introduction

Since periods of centuries, the fossil fuels not only fulfill our basic energy demand, but also have major contribution in the development of society. Due to their ever increasing demand and higher consumption rate along with rapid exhaustion of the reservoirs, fossil fuels make an alarming sound to the researchers. The effective approach to overcome this problem is to conserve conventional fuels by utilizing renewable energy sources or other non-commercial energy sources [1].

The limited stock of fossil fuels and their uneven distribution on earth are responsible to raise its cost in multiple times. This leads to severe economical imbalances in developing countries, which fulfill its main fuel demands by oil imports. In due course of time, this tendency is likely to worsen further causing greater dearth and adversity [2].

The second effect of fossil fuel is due to its combustion, which causes damage to the environment. The technology used to extract fossil fuel, transportation, processing, storage of oil and gas, its spillways and leakages, and their combustion have harmful effect on the environment and causes both air and water pollution [3]. Mostly harm to the environment is exaggerated by combustion of fossil fuels. The main constituents of these fossil fuels are carbon, hydrogen, small amounts sulfur and oxygenated additives which are mixed in the fossil fuels for further enhancement of their performances. These constituents produce various gases like CO_2, HC, CO, NO_x etc. along with soot, ash and other organic compounds during combustion which when emitted into atmosphere cause degradation of air quality [4].

In this context, the dual fuel engine, not only maintains most of the positive features of diesel engine operation [5], but also its performances are better than diesel. It develops higher power outputs and efficiencies. This all is achieved without considerable amount of particulates emissions and reduced NO_x generation, along with reduction in peak cylinder pressure and smooth operation.

In dual-fuel engine, a carbureted mixture of air and high octane index gaseous fuel, which is compressed like in a regular diesel engine, is used. The compressed gaseous fuel-air mixture does not ignite spontaneously due to its higher self ignition temperature. Hence, it is fired by a small liquid fuel injection which ignites rapidly at the end of compression process. The main advantage of this type of engine is that it utilizes the diversity in flammability limits of two different types of fuel [6].

Normally, gaseous fuel like CNG, LPG, bio gas, producer gas, hydrogen etc., and fumigated liquid fuels like gasoline, methanol etc. in air are used as secondary fuels. The pilot diesel fuel like in normal diesel engine ignites due to higher temperature of the compressed charge, initiates combustion in primary fuels (diesel). A large number of ignition centre (nuclei) formed by diesel spray reduces the time and flame travel distance and makes sure knock free operation at higher compression ratio as compared to SI engine [7].

Nieminen et al. [8] developed some models for comparative combustion characteristics of gasoline and hydrogen fuelled spark ignition engine. The analysis of those models indicated that approximately 6.42% increase in thermal efficiency for the hydrogen fuelled engine due to less exhaust blow down, less heat rejection during the exhaust stroke, and its shorter Burning period closer to TDC.

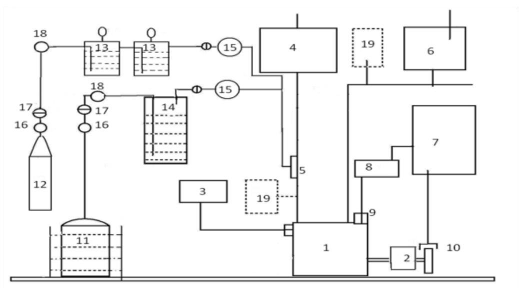
Yamin et al. [9] analytically discussed the effect of compression ratio, equivalence ratio, spark plug location, spark timing and engine speed on Burning period. It was observed that Burning period has significant effect on both performance and emission characteristics of the engine.

2. Experimentation

A schematic layout of the diesel engine Trial setup used during the experiments is shown in Fig 1. Table 1 shows the engine geometry and operating parameters for the present work. The diesel engine is modified to work on dual fuel mode by connecting hydrogen and LPG gas cylinders in association with the intake manifold through flame traps, mass flow meters, followed by a one way non-return valve and common flame arrestor by keeping turbocharger and its bypass active. The engine was coupled to a 62.5 kW D.C. generator. The load on the engine was varied by introducing five water pumps and twelve 3 kW industrial water heaters in a set of four each. The power is based on the electrical output of the installation. The engine was run at a constant speed of 1500 RPM [10].

The mass flow rate of hydrogen and LPG was measured by volume flow meters in liters per minute. To ensure repeatability the experiments were carried out for five times.

The experiments were performed on the Trial engine under the following four conditions.



1-Engine, 2-Gen-set, 3- Diesel tank and measurement system, 4- Air tank and measurement system, 5-Gas mixture, 6- Gas analyzer, 7- PC based data acquisition system, 8- Charge amplifier, 9- Cylinder pressure sensor, 10-Crank angle encoder, 11-LPG gas cylinder, 12- Hydrogen gas cylinder, 13-Hydrogen gas flame trap, 14- LPG gas flame trap, 15- Gas flow meter, 16- Gas cylinder control valve, 17- Pressure regulator, 18- Solenoid switch valve, 19- Temperature and Pressure measurement locations.
Fig. 1 Experimental Set up.

Table 1. Engine Specifications.

Sr. No.	Parameter	Engine A Specification
1.	Make and Model	Ashok Leyland ALU WO4CT Turbocharged, inter-cooler, Gen- Set
2.	General Details	Four Stroke, Compression Ignition, Constant Speed, vertical, water-cooled, direct injection, turbo charger, Intercooler, Gen-Set
3.	No. of Cylinder	4
4.	Bore mm	104
5.	Stroke mm	113
6.	Rated Speed (rpm)	1500
7.	Swept volume (cc)	3839.67
8.	Clearance volume (cc)	84.90
9.	Compression ratio	17.5:1
10.	Injection Pressure (bar)	260

(i) Trial I: Engine runs on diesel only.

(ii) Trial III: Engine runs on diesel as pilot fuel and LPG as secondary fuel.

(iii) Trial IV: Engine runs on diesel as pilot fuel and LPG plus hydrogen as secondary fuel.

The uncertainty in the calculated values such as brake power, brake thermal efficiency was analyzed. The average percentage uncertainty in brake power was ± 1.44%, while average percentage uncertainty in brake thermal efficiency was ± 0.51% with variable brake power, mass of diesel, LPG and hydrogen. The percentage uncertainties of the measuring instruments are as follows: mass flow rate of gaseous fuels ± 4%, mass flow rate of diesel ± 3%, speed ± 1%, HC ± 2%, CO ± 3%, NO_x ± 3%, and crank angle ± 1^0.

3. Results and Discussion

The experimental results at rated speed of 1500 rpm, injection pressure 260 bar and injection timing 16^0 BTDC are presented for Trials II, III, and IV at different load conditions. The quantities of hydrogen and LPG in Trials II, III and IV were supplied at pre calculated rate as given in Trial matrix [10] and diesel fuel was automatically supplied through fuel metering and governing system at all load conditions. Zero percent gaseous fuel substitution represents pure diesel operation. Due to knocking, the maximum amount of hydrogen and LPG substitution was limited to 50% and 70% respectively.

The mixture of LPG and hydrogen was varied in the following proportions in each combination (M): (LPG-90% + H_2-10%), (LPG-80% + H_2-20%), (LPG-70% + H_2-30%), (LPG-60% + H_2-40%) [10].

The Burning period can be calculated from “mass fraction burnt curve”. Two aspects of mass fraction burnt curve are used to characterize ignition delay and Burning period. Ignition delay is defined on the basis of mass fraction burnt curve as the angle between the time of injection and the time at which 1% of the mass fraction is burned.

While, burning period is defined as the angle change from 1% to 90% of mass fraction burnt [11].

Figure 3 shows the Burning period for the Trials I and II at 10%, 40% and 80% load conditions. The Burning period for 30% of hydrogen substitution at 10% load condition is 29 ^{0}CA as compared to 24 ^{0}CA of Trial I operation. This may be due to sluggish flame propagation in

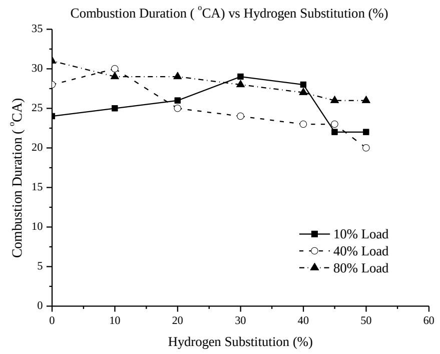


Fig. 3 Burning period (^{0}CA) vs. hydrogen substitution (%).

the lean gaseous fuel-air mixture [12]. The sluggish spread of flame front in dual fuel mode at light load condition results in poor brake thermal efficiency also. Similarly, the Burning period at 40% load condition is 24 ^{0}CA as compared to 28 ^{0}CA of Trial I operation. The 50% of hydrogen substitution shows 7 ^{0}CA and 4 ^{0}CA falls in Burning period in Trial II operation at 10% and 40% load conditions respectively.

At 80% load condition, the 30% and 50% hydrogen substitution shows Burning period of 28 ^{0}CA and 26 ^{0}CA respectively as compared to 31 ^{0}CA of Trial I operation. The drop in Burning period of 5 ^{0}CA is due to higher flame velocity of hydrogen (265 cm/sec) that leads to higher combustion rate (flame propagation) and lower Burning period. Further, in Trial I, the injected fuel burns in diffusive mode and therefore, longer Burning period is observed. The larger diesel quantity at higher load condition makes higher ignition sources and burning of gaseous fuel is faster due to its higher laminar velocity which also leads to higher brake thermal efficiency. It is further observed that shorter Burning period leads to higher brake thermal efficiency.
The Burning period at 30% and 50% of LPG substitutions are found to be 27 ^{0}CA, 26^{0}CA, 35 ^{0}CA and 31^{0}CA, 24 ^{0}CA and 29 ^{0}CA respectively at 10%,

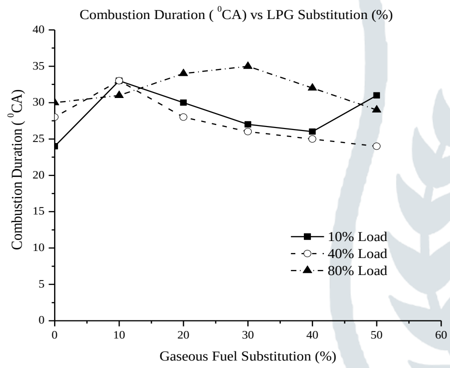


Fig. 4 Burning period (^{0}CA) vs. LPG substitutions (%).

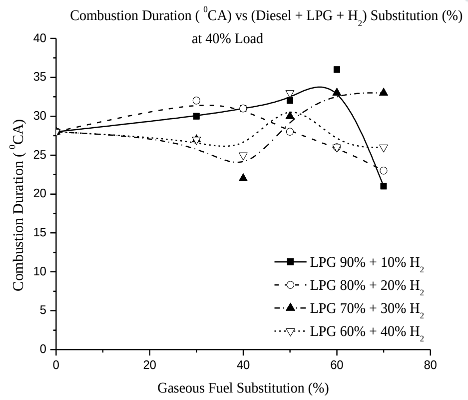


Fig. 5 Burning period (^{0}CA) vs. Diesel + LPG + hydrogen fuel substitution (%) at 40% load.

40% and 80% load conditions as compared to 24 ^{0}CA, 28 ^{0}CA and 30 ^{0}CA of Trial I operation Fig. 4. At lower load condition deterioration in ignition sources is caused due to less pilot diesel quantity that results in increase in Burning period. The combustion is mainly due to flame propagation through gaseous-air mixture. After ignition delay, a part of heat release is affected by flame propagation which dominates first phase of combustion. At 50% of LPG substitution, less availability of oxygen causes rich gaseous fuel-air mixture which results in higher Burning period than Trial I operation.

At 80% load condition, the most of the injected pilot diesel fuel burns in the diffusive mode and therefore, longer Burning period is observed in Trial III up to 30% of LPG substitution. Further increase in LPG substitution causes a decrease in Burning period due to lower ignition delay and higher flame velocity [13].

At higher quantity of pilot diesel fuel, the combustion is mainly due to premixed burning of the pilot fuel in first phase of combustion along with diffusive burning of pilot and simultaneous burning of gaseous fuel in second phase of combustion. The combustion in second phase of combustion is rapid and hence, the Burning period at higher substitution is lower than diesel operation.

The Burning period at 40% and 80% load conditions are shown in Figs.5-6. The Burning period at 40% load condition (at 30% gaseous fuel substitution) and with 10% to 40% hydrogen in LPG-hydrogen mixture shows 30, 32, 27 and 27 ^{0}CA respectively as compared to 28 ^{0}CA of Trial I operation. The Burning period in Trial IV at initial condition is more than Trial I operation due to sluggish flame propagation occurring in the lean gaseous-air mixture. As the pilot quantity increases, Burning period decreases due to presence of a strong ignition source.

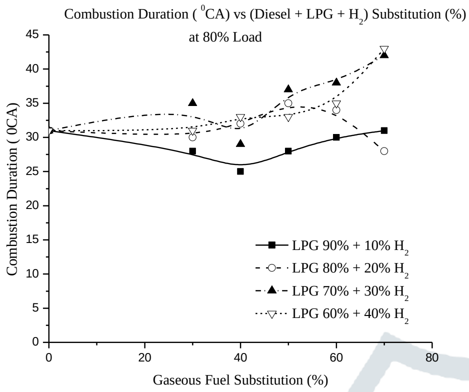


Fig. 6 Burning period (^{0}CA) vs. Diesel + LPG + hydrogen fuel substitution (%) at 80% load

Hydrogen has higher diffusivity as compared to hydrocarbon fuels, which improves mixing, enhances turbulences and increase homogeneity in charge.

At 80% load condition as in Trial IV, Burning period is 28, 29, 35 and 31 ^{0}CA as compared to 32 ^{0}CA of Trial I operation at 30% of gaseous fuel substitutions. As hydrogen concentration in the mixture increases, Burning period initially decreases up to 30% of hydrogen substitution. Beyond 30% it increases due to rapid flame propagation in the gaseous-air mixture after ignition. The Burning period decreases in Trial IV due to increase in load also.

The presence of hydrogen in the gaseous fuel-air mixture (Diesel + LPG + hydrogen) increases combustion flammability. Hydrogen has high flame speed more than five times than the hydrocarbon fuels. Also it has lean limits (mixture at which flame will not propagate due to excess air) of equivalence ratio $(\phi = 0.1)$, much lower than theoretical limit of the same for diesel ($\phi = 0.6$). Theoretically, it is possible to extend the lean limit of the mixture by adding a small amount of hydrogen to a liquid or gaseous hydrocarbon fuel.

4. Conclusions

Experiments were performed on 4 cylinder turbocharged, intercooled with 62.5 kW gen-set diesel engine using hydrogen, LPG and mixture of LPG and hydrogen as secondary fuels.

On the basis of the results and discussions presented above, the following conclusions may be drawn:

1. The Burning period at lighter load condition in Case II is more than Case I operation.
2. At 80% load condition and 50% of hydrogen substitution, the Burning period is reduced by 5 ^{0}CA than Case I operation.
3. The Burning period is larger in dual fuel mode as compared to diesel operation at lower substitution of LPG at all load conditions due to lower temperature in the combustion chamber and higher ignition delay.
4. The Burning period at 30% and 50% of LPG substitutions are found to be 27 ^{0}CA, 26 ^{0}CA, 35 ^{0}CA and 31^{0}CA, 24 ^{0}CA and 29 ^{0}CA respectively at 10%, 40% and 80% load conditions as compared to 24 ^{0}CA, 28 ^{0}CA and 30 ^{0}CA of Trial I operation.
5. At higher load condition the Burning period decreases at higher substitution of gaseous fuel due to lower ignition delay period, higher combustion rate, and higher flame propagation velocity.
6. At low loads, the Burning period decreases with an increase in gaseous fuel substitution due to higher combustion rate in the gaseous fuel mixture.
7. At 80% load condition as in Trial IV, Burning period is 28, 29, 35 and 31 ^{0}CA as compared to 32 ^{0}CA of Trial I operation as hydrogen percentage increases from 10 to 40%. As hydrogen concentration in the mixture increases, Burning period initially decreases up to 30% of hydrogen substitution then increases.
8. At higher loads, the Burning period is lower than diesel up to 40% of gaseous fuel mixture substitution. The Burning period decreases due to higher temperature and higher loads which generates high flame propagation velocity.

It may be conclude that at lower gaseous fuel substitution dual fuel engine shows more Burning period, while at higher substitutions Burning period get decrease.

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