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Investigation on Engine Performance and Exhaust Emissions in an LPG Diesel Dual Fuel Engine

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ABSTRACT

In a dual fuel engine, two fuels are used simultaneously. The primary fuel is usually gaseous forms the major content of the total energy supplied to the engine. The secondary fuel, i.e., pilot fuel, is injected after compression of the primary fuel air mixture. Much of the energy release comes from the combustion of the gaseous fuel and a small amount of diesel fuel provides ignition through timed cylinder injection. In the present work, single-cylinder, compression ignition, direct injection diesel engine has been used for the investigations of exhaust emissions when the engine is operating as a dual-fuel engine using diesel as pilot fuel and Liquefied Petroleum Gas (LPG) as secondary fuel. The influence of major engine operating parameters, such as the pilot fuel quantity, intake air temperature, Exhaust Gas Recirculation (EGR), intake air throttling and rate of injection on the exhaust emissions was investigated. Diesel fuel was used as the pilot fuel, while LPG was used as the main fuel which was inducted in the intake manifold. The experimental investigations showed that the poor exhaust emissions at light loads can be improved by employing larger pilot fuel quantity, using EGR, increasing intake temperature and well adjusted rate of injection.

Keywords : Dual fuel, LPG, Thermal Efficiency, Diesel and Performance

I. INTRODUCTION

LPG is an alternative fuel for both spark ignition and compression ignition engine. In the case of compression ignition engine, LPG is supplied in suction stroke mixed with air while diesel is injected at the end of compression stroke to initiate the combustion process. This type of engine operation is known as dual fuel operation. These engines retain the injection of a small quantity of liquid diesel fuel as a pilot fuel to initiate the combustion while introducing gaseous fuel in suction stroke with the air to provide the most of the energy release. In dual-fuel engines, the gaseous fuel replaces most of the diesel fuel, with only a small pilot injection of diesel used to ignite the gas [1].

LPG can be used in the diesel engines without much modification in the engine system. Dual-fuel engines, fuelled with various gaseous fuel resources, produce less exhaust emissions than conventional diesel engines without any substantial increase in operating and capital cost. Smoke and

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A. S. Jethoo is with the Department of Civil Engineering at the Malaviya National Institute of Technology, Jaipur, India (email: asjethoo@gmail.com). supply system which is able to economically convert conventional diesel engines into dual-fuel engines like LPG/diesel engines and CNG/diesel engines. This engine is particulate emissions from dual-fuel engines generally tend to be much lower than with the corresponding neat diesel operation [2]. Reduction in air flow due to addition of LPG into the engine slightly increases carbon monoxide and hydrocarbon emissions. At light load, dual-fuel engines usually exhibit a drop in brake thermal efficiency and power output with respect to corresponding diesel operation. The emissions of unburned hydrocarbons and carbon monoxide under dual-fuel operation are higher by comparison with diesel operation [3, 4].

Barata [5] has studied the exhaust emission of a dual-fuel engine with propane as the primary fuel. He reported that the carbon monoxide concentrations were higher at all loads. This effect was more pronounced for light loads, i.e., less than 50 per cent of maximum rated power output. In contrary to this, the NOx levels were always lower with the introduction of gaseous fuel.

A small percentage of exhaust gases recirculation into the cylinder in dual mode operation shall improve engine performance in terms of engine brake thermal efficiency. Recirculation of exhaust gases into the engine increases the intake temperature. This also brings down the level of smoke because unburned of previous cycle may re-burn in the forthcoming cycles [6]. As the Exhaust gas ratio increases, the hydrocarbon emissions decrease. This is mainly due to the presence of radicals, initiation of combustion process become fast [7]. Reid and Stephenson [8] carried out limited amount of air throttling work and indicated substantial improvements in thermal efficiency at part load with high gas substitution levels in dual fuel engines.

Simonson [9] confirmed that the intake air restriction gave higher thermal efficiency. Rao *et al*. [10] have indicated that with the dual-fuel mode of operation, precious diesel could be conserved up to 80%. However, in their work, it could be done only up to 45% due to severe engine vibrations. The brake power of the engine was found to be about 15% more on the dual-fuel operation, while the brake specific fuel consumption was found to be about 30% lower than diesel fuel mode of operation. This could be due to better mixing of air and LPG and improved combustion efficiency. Srinivasa Rao *et al*. [11] have made experimental investigations on a dual-fuel engine with LPG as the main fuel and diesel as the pilot fuel. The engine was run under different operating conditions of load and speed. The optimum combination of fuels was evaluated at each of the conditions. The exhaust smoke level, exhaust gas temperature and other performance parameters were calculated and a cylinder pressure trace was obtained using a data acquisition system.

Dong Jian *et al*. [12] developed a new type of dual fuel capable of either using single diesel fuel or using dual-fuel including both diesel and CNG fuel and both diesel and LPG fuel. Vezir Ayhan *et al*. [13] investigated the effects of LPG injection during air inlet period on emissions and performance characteristics. The engine has been modified to determine the best LPG substitution for dual operation in order to improve the emissions quality while maintaining high thermal efficiency in comparison to a conventional diesel engine. An electronic controlled LPG injection system has been developed for this purpose. LPG injection rate were selected as 5, 10, 15 and 25% on a mass basis. Minimum SFC and maximum brake efficiency obtained with 15% LPG between 1400 and 1800 rpm engine speeds. Optimum injection rates is found at 5% LPG in terms of exhausts emissions and performance.

Qi *et al*. [14] conducted an experimental investigation on a single cylinder DI diesel engine modified to operate under LPG–Diesel dual fuel mode. Using LPG–Diesel blends of various rates; 0, 10, 20, 30 and 40%, they compressed LPG by pressured N_2 gas to mix with the diesel fuel in a liquid form. They concluded that LPG-Diesel blended fuel combustion is a promising technique for controlling both NOx and smoke emissions even on existing DI diesel engines.

LPG can be used in dual fuel compression ignition engines as a primary fuel (Mohamed *et al*., [15]; Miller *et al*., [16]; Li *et al*., In a dual fuel compressed ignition system (CI), the engine is operated with LPG as primary fuel providing with a pilot amount of diesel fuel is used as an ignition source. Due to its high auto-ignition temperature, the mixture of air and LPG does not auto-ignite. The pilot Diesel fuel injected for ignition can only contribute to a small fraction of the engine power output.

The nature and extent of the exhaust emissions from dual-fuel engines has been the subject of investigation over the years. In the present study, the effects of load, EGR, intake air throttling and rate of injection and the percent energy substitution by LPG on the exhaust emissions and engine performance were studied at optimum engine operating conditions.

II. APPARATUS AND EXPERIMENTAL PROCEDURE

A single cylinder, direct injection, water cooled diesel engine was used for this experimental work. The specifications of the engine are given in Table I. The schematic layout of the test setup used is indicated in Fig.1.

TABLE I: ENGINE DETAILS

Engine Type	Single cylinder, four strpke, direct injection diesel engine
Rated Power	3.7 kW @ 1500 RPM
Bore × Stroke	80 mm × 110 mm
Compression Ratio	15: 1
Injection Timing and Injector Opening Pressure	27.4 º BTDC (Static), 170 bar

An LPG connection was made on the intake manifold. Governor varied pilot diesel flow while LPG flow rate was varied manually. Based on a previous experimental program injection timing of 27.4º BTDC and 24º BTDC were selected for dual fuel and diesel operation respectively. Based on the same work injection pressure of 170 bar used. Arrangements have been made to control the intake air supply at light load conditions. The developed throttling mechanism had eight equal steps from 100% throttle closing to 0% throttle closing. The optimum pilot fuel quantities and intake temperatures set for the experimentation at different loads are given in Table II.

1-engine 2-dynamometer 3-diesel tank and measurement system

4-air flow surge tank and meter 5-air pre heater

6-hc/co analyzer 7-pc based data acquisition system

8-charge amplifier 9-pressure pickup 10-shaft position encoder

11-lpg cylinder in a constant temperature water bath

12-control valve 13-pressure regulator

14-positive displacement gas flow meter 15-rotometer

16-flame trap 17-control valve 18-gas mixer 19-exhaust outlet

20-air inlet 21-temperature and pressure measurement points

22-egr valve 23-throttle valve

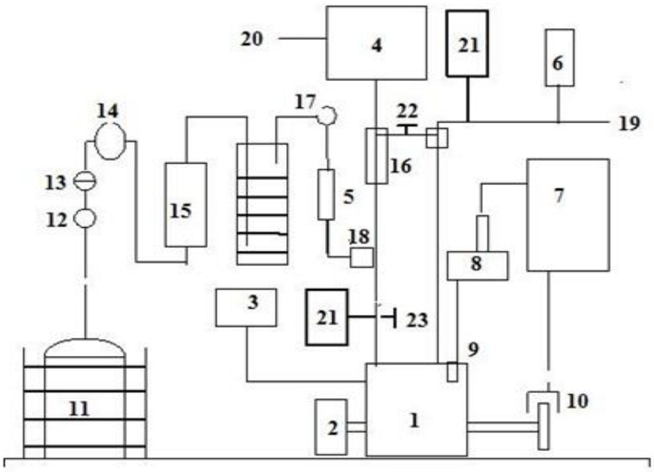

Figure 1. Experimental set up

TABLE II : OPTIMUM ENGINE OPERATING CONDITIONS

Load in %	Intake Temp. (º C)	Pilot Fuel Quantity mg/cycle	Throttle Opening in %	Optimum EGR by Volume in %	% LPG Substitution
20	70	8.4	50	18	25.0
40	70	10.7	62.5	10	36.0
60	60	10.7	100	8	100
80	60	5.9	100	6	100
100	40	7.1	100	4	100

III. VARIATION OF INLET AIR THROTTLE

It was thought worth to investigate the combined effect of intake air heating; EGR and throttling the intake charge on the engine exhaust emissions in dual fuel mode at light load. In the present investigations, effect of intake air throttling at 20% and 40% load on the engine emission were studied in the dual fuel mode moreover with diesel for comparison. The intake manifold depression was varied from 10 cm (0% throttle closing) to 60 cm (75% throttle closing) of water with the help of a throttle. The throttle position was varied from 75% throttle closing to 0% closing with the help of a manually controlled lever. The engine speed was maintained constant at 1500 rev. /min. The tests were conducted only at 20% and 40% of full load at optimum injector opening and injection timing.

IV. RESULTS AND DISCUSSIONS

At low loads, the fuel air ratio of the inducted mixture is very low in the case of dual fuel engine. In this mixture the flame propagation is not complete and most of the fresh air-gas mixture remains unburnt resulting in lower brake thermal efficiency and high HC emission. Adopting inlet air throttling tends to increase the effective fuel air ratio of the charge by reducing the amount of air sucked per stroke. The tests were conducted only at 20 and 40% load at optimum intake temperature with and without optimum EGR at 170 bar injector opening pressure and 27.4° BTDC injection timing. At higher loads, throttling may deteriorate the engine performance due to decrease in volumetric efficiency and hence it was not studied. The hydrocarbon emission was studied in the dual fuel mode.

Fig. 2 and Fig. 3 show the effect of intake air throttling at 20% and 40% load respectively. Tests were conducted with optimum injector opening pressure (170bar) and injection timing of 27.4° BTDC keeping optimum pilot fuel quantity and with optimum EGR and without EGR. 18% EGR was found to be optimum from previous tests at 20% load. This 18% EGR is based on actual mass of air inducted with throttling. The brake thermal increased and HC emissions decreased significantly by throttling the air at 20% load with 0% EGR up to 50% throttle closing. The HC emissions were reduced by 130 ppm by throttling with 18% EGR as compared to 50 ppm reduction with 0% EGR. Inlet air throttling tends to increase the effective fuel air ratio of the charge by reducing the amount of air sucked per stroke resulting in better combustion in first and second stage. Addition of hot EGR further improves the effective fuel air ratio by replacing the air. It seems that after 50% throttle closing, the ignition delay becomes very long resulting in lower peak pressure and brake thermal efficiency. This may be due to lower compression pressure and very rich mixtures of gas and air.

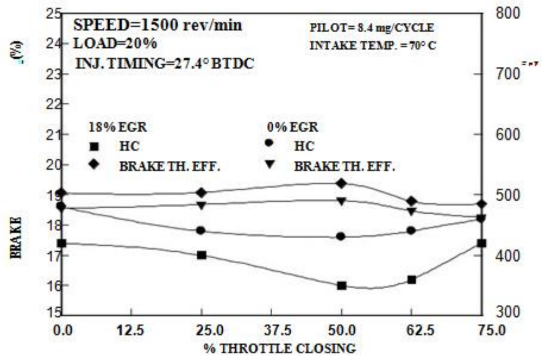

Figure 2. Effect of throttling on brake thermal efficiency and HC emissions

At 40% load the same trends have been observed as in the case of 20% load as shown in Fig. 3. Due to increase in the quantity of gas, the improvement in brake thermal efficiency and HC emissions are less at 40% loads, the mixture became richer approximately by 2.6% with throttling. The engine running was found to be very smooth due to reduction in peak cycle pressure. The optimum throttle position found to be 50 %(manifold depression 21 cm of water) and 37.5% (manifold depression 15 cm of water) at 20% and 40% load respectively. The exhaust gas temperature was found to increase at too high throttling due to delayed combustion in expansion stroke at both 20 and 40% loads.

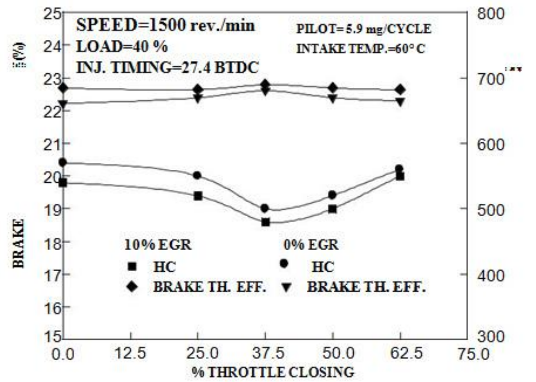

Figure 3. Effect of throttling on brake thermal efficiency and hydrocarbon emissions

To estimate the effect of throttling on the gas substitution at low loads, tests were conducted at the optimum throttle opening of 50% and varying gas substitution at 20% load as seen in Fig. 4. In these tests, the intake temperature and EGR were kept at the optimum values of 70˚ C and 18% respectively (in the case of 20% load). For the purpose of comparison, the results without throttling have also been plotted. It is seen that as the gas substitution increases the brake thermal efficiency and HC emissions deteriorate with increasing gas substitution beyond the optimum value of 25% (8.4 mg/cycle pilot) obtained in the other study. With optimum throttling, at every substitution, the engine performance was found to be better as compared to that without throttling. This does indicate the ability of air throttling to improve the engine performance at higher gas substitution at light loads. HC emissions are also lower with throttling at every gas substitution value. Similar trends are observed at 40 % load as shown in Fig. 5. However, the effect on brake thermal efficiency is not very significant at 40 % load for reasons indicated earlier.

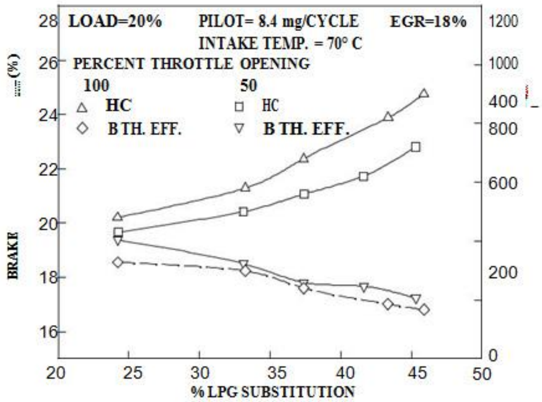

Figure 4. Effect of LPG substitution on brake thermal efficiency and UN burnt hydrocarbons at load 20%

Experiments were also conducted at optimum operating conditions as indicated in Table II while varying the rate of injection of pilot fuel. The results of the brake thermal efficiency for the two plungers (6 mm and 5 mm plunger diameter) at 1500 rev./min are given in Fig. 6. The optimum injection timing (static) of 27.4 ˚BTDC for 6mm plunger was maintained. With the 5 mm plunger optimum injection timing was 37.5 ˚ BTDC. It is observed that the brake thermal efficiency at 80% load is highest in both cases. It is 32% with 6 mm plunger (higher rate of injection) as compared to 30 % load with 5 mm plunger (lower rate of injection) at 80% load.

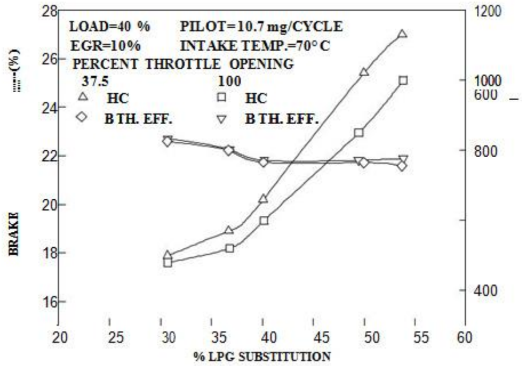

Figure 5. Effect of LPG substitution on brake thermal efficiency and UN burnt hydrocarbons at load 40%

Even though it was thought that at low outputs a lower rate of injection will improve performance by resulting in an improved source of ignition, it did not help as the spray formation deteriorated. The low brake thermal efficiency with lower rate of injection may also be attributed to improper mixture formation. The lower value of brake thermal efficiency at reduced rate of injection is mainly due to longer dynamic injection duration. It is observed that with lower injection rate, the dynamic injection duration has increased up to 12˚ of crank angle and even with very high advanced static injection; the same dynamic timing that was observed with 6 mm plunger could not be obtained. To obtain the same dynamic timing, the injector opening pressure was further reduced to 125 bars by keeping other parameters constant.

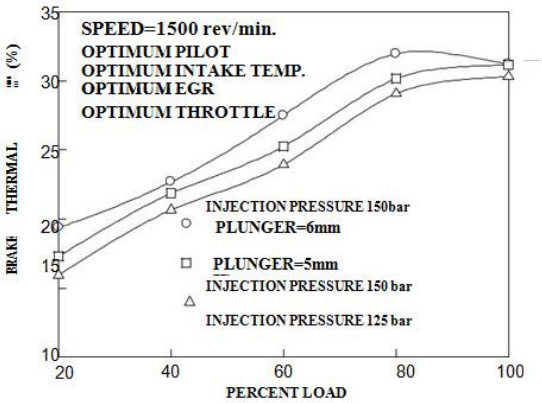

Figure 6. Load v/s break thermal efficiency at optimum points

The ignition delay periods were also to be observed very high with low rate of injection (5-8˚ longer). However at 100 % load the brake thermal efficiency of the two versions is nearly comparable. At this load due to higher temperature and pressure of the cycle, the role of the pilot fuel is not very significant. With lower injection pressure (125 bar),the brake thermal efficiency of the engine further deteriorated due to improper mixture formation. For a given load, with lower rate of injection, the exhaust gas temperature and HC emissions were found to be very high.

V. CONCLUSION

Through experimental investigations, a direct injection dual-fuel engine fuelled with LPG and diesel were tested to determine its performance and exhaust emission characteristics with the objectives of improving engine efficiency and exhaust emissions at part loads. The parameters considered to achieve these objectives were gaseous fuel quantity, pilot fuel quantity, pilot fuel injection rate, intake air throttling, EGR and the intake air temperature. First engine operation was optimised at different operating conditions and then percent gas substitution was varied at optimum conditions. The main conclusions of the present study are summarized as follows:

1. Intake air throttling at low loads can improve both engine efficiency and HC emissions. However the effect is more significant at 20% load as compared to 40% load.
2. It is seen that as the percent gas substitution increases the brake thermal efficiency and HC emissions deteriorate with increasing gas substitution beyond the optimum value of 25% (8.4 mg/cycle pilot). However this effect is not significant at 40% load.
3. With optimum throttling, at every gas substitution, the engine performance was found to be better as compared to that of without throttling. This does indicate the ability of air throttling to improve the engine performance at higher gas substitution at light loads. HC emissions were also found to be lower at every gas substitution value.
4. Lower rate of injection with 5 mm plunger, results in poor brake thermal efficiency due to improper mixture formation. Longer dynamic injection timing and ignition delay period is observed to be the cause of poor brake thermal efficiency and deteriorated HC emissions.
5. An improvement in the combustion process was achieved by advancing the injection timing because of the corresponding high pressure and temperature, with no decrease in the combustion duration. The increase in NOx emissions with advance of the injection timing was attributed to the increase in the maximum temperature of the charge. Meanwhile, the reduction in carbon monoxide and unburned hydrocarbons was also observed due to increase in temperature.

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