

EXHAUST ANALYSIS AND PERFORMANCE OF A SINGLE CYLINDER DIESEL ENGINE RUN ON DUAL FUELS MODE

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ABSTRACT

Generally fossil based fuels are used in internal combustion engines as an energy source. Excessive use of fossil based fuels diminishes present reserves and increases the air pollution in urban areas. This enhances the importance of the effective use of present reserves and/or to develop new alternative fuels, which are environment friendly. Use of alternative fuel is a way of emission control. The term "Alternative Gaseous Fuels" relates to a wide range of fuels that are in the gaseous state at ambient conditions, whether when used on their own or as components of mixtures with other fuels.

In this study, a single cylinder diesel engine was modified to use LPG in dual fuel mode to study the performance, emission, and combustion characteristics. The primary fuel, liquefied petroleum gas (LPG), was mixed with air, compressed, and ignited by a small pilot spray of diesel. Dual fuel engine showed a reduction in oxides of nitrogen in the entire load range. The brake thermal efficiency improved by 3% in dual fuel mode, especially at low load, and also reduced the hydrocarbon, carbon monoxide, and CO₂ emissions.

الخلاصة

يسنخدم الوقود الاحفوري بشكل عام كمصدر طاقة لمحركات الاحتراق الداخلي، وتسبب الاستخدام المفرط لهذا الوقود بنضوب معظم مصادرة اضافة الى تلوث الهواء في المساحات المدنية، ولهذا تركز الاهتمام على الاستخدام الفعال للمصادر الحالية للطاقة أو تطوير مصادر وقود جديدة، تكون صديقة للبيئة. إن استخدام الوقود البديل هو طريقة للسيطرة على التلوث، وتعبير " الوقود الغازي البديل" يرتبط بعدة أنواع من الوقود تكون بالحالة الغازية في ظروف الجو الاعتيادية، سواء استخدمت بمفردها أو مشتركة ومخلوطة مع وقود آخر.

في هذه الدراسة، تم تحويل محرك ديزل أحادي الاسطوانة ليستخدم الغاز النفطي المسال بأسلوب الوقود الثنائي، لدراسة خصائص الاحتراق كداء وملوثات المحرك. يتم خلط الغاز النفطي المسال (الوقود الابتدائي) بالهواء، يضغط بعدها ثم يشعل بواسطة كمية صغيرة من وقود ديزل محقون. أظهر محرك الوقود الثنائي نقصاناً في ملوثات أكاسيد النيتروجين لكافة الاحمال المستخدمة، كما تحسنت الكفاءة الحرارية بمقدار 3% في هذه الحالة، كما قلت ملوثات الهيدروكربونات غير المحترقة وأول أكسيد الكربون وثاني أكسيد الكربون.

KEY WORDS

Diesel engine; Dual fuel engine; Liquefied petroleum gas (LPG), Exhaust emissions; Alternative fuel, reduction in pollutant emissions and fuel consumption

INTRODUCTION

Global pollution caused by transport consists primarily of the emissions from transport systems during manufacture, operation and disposal of the greenhouse gases. These gases include the direct greenhouse gases such as carbon dioxide and methane, the ozone precursors such as hydrocarbons and the nitrogen oxides, and carbon monoxide which have an indirect effect on greenhouse gas production. The problem of global pollution from transport activities is centered on the use of fossil fuels for transport. These fuels are mainly in the form of gasoline and diesel oil for road transport, kerosene for air transport, and diesel oil for rail transport, industrial application, shipping etc (Selim, 2005 and Lakshmanan, 2009).

Gaseous fuels promise to be suitable for higher compression ratios engines, since it is known that they resist knock more than conventional liquid fuels, as well as producing less polluting exhaust gases if appropriate conditions are satisfied for its mixing and combustion (Yousufuddin, 2008). Therefore, it is more economical and of environmental advantage to use gaseous fuel in diesel engines that use the dual fuel concept. There have been many published works on the use of gaseous fuels in dual fuel engines. Natural gas use in dual fuel engines has been studied from the combustion duration and ignition delay point of view (Stanislav, 2001) and from the performance and emissions point of view (Abd Alla, 2002). Combustion and thermal loading and temperature distribution have also been studied for dual engines (Karim, 1991). Pure methane has also been studied in dual fuel engines from the flame spread limits point of view (Papagiannakis, 2003) and performance and emissions point of view (Selim, 2004).

The potential benefits of using LPG in diesel engines are both economical and environmental. In the dual fuel 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 (Vijayabalan, 2009). With reduced energy consumption, the dual fuel engine shows a significant reduction in smoke density, oxides of nitrogen emission, and improved brake thermal efficiency. 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 & CO emissions at low loads (Ma, 2007 and Salman, 2004).

The increase in pilot diesel improves the brake thermal efficiency at low loads. At higher loads, it reduces efficiency due to rapid combustion. Low efficiency and poor emissions at light loads can be improved significantly by advancing injection timing of the pilot fuel. Any measures that lower the effective lean flammability limit of charge and promote flame propagation will improve part load performance (Heng, 2008).

According to the characteristics of LPG (table 1) there are following specialties in the engine:

- LPG high ignition temperature and safety: In the normal temperature it can be liquefied at 1.6MPa 2Mpa which makes it easy to use.
- LPG has high heat value, with its gaseous state makes it easy to mix with air. It has perfect combustion redounds to improve power output, as well as, good antiknock because of high octane (Saleh, 2008).
- Soot can be obviously reduced because low carbon compound. Also the accumulative soot layer can be reduced in combustion chamber (Benea, 2007).
- Gas fuel will not dilute the engine lubrication oil. The replacement period of lubrication oil could be longer (Qi, 2007).

But ignition time delay of LPG is longer because the low cetane number. Moreover the

volumetric efficiency of LPG is lower. It must make fully use of advantages of LPG and improve the power and economy of engine and reduce the pollutant. In order to reach this object, it must optimize structure and operating parameters of the engine (Sethi, 2004).

The present study is concerned with the reduction of diesel engine exhaust emissions. For this purpose, a single cylinder, indirect injection diesel engine was modified to operate with dual fuel. During the experimental study the engine was run with (30% and 60%) LPG and (70% or 40%) diesel fuel (by volume) and changes in engine performance and exhaust emissions were observed.

EXPERIMENTAL TECHNIQUE

The test rig used in the present study is the Ricardo E6 single cylinder variable compression indirect injection diesel engine. The specifications of the engine are listed in **Table 2**. The engine cylinder head has a Ricardo Comet Mk V compression swirl combustion chamber. This type of combustion system consists of two parts. The swirl chamber in the head has a top half of spherical form and the lower half is a truncated cone which communicates with the cylinder by means of a narrow passage or

throat. The second part consists of special cavities cut into the crown of the piston. The engine is capable to run on 100% diesel fuel or dual fuel. The engine is converted to run on dual fuel by introducing the gaseous fuel, LPG in the present work, in the intake manifold by means of a gas adapter. The gas is injected at a pressure slightly higher than atmospheric pressure.

The schematic diagram for the engine test rig is shown in **Fig. 1**. The engine is loaded by an electrical dynamometer rated at 22 kW and 420 V. The engine is fully equipped for measurements of all operating parameters. The liquid fuel flow rate is measured by means of two tanks, main tank (9 liters) and secondary tank (1 liter), with a set of valves to close and open fuel line, and a fuel flow measurement device. The gaseous fuel flow rate is measured by using an orifice meter connected to electronic partial pressure transducer that is connected to a digital pressure meter.

The Multigas mode 4880 emissions analyzer was used to measure the concentration of nitrogen oxide (NO_x), unburned total hydrocarbon (HC), CO₂ and CO. This device was calibrated at Central Organization for Standardization and Quality Control in Baghdad-Iraq.

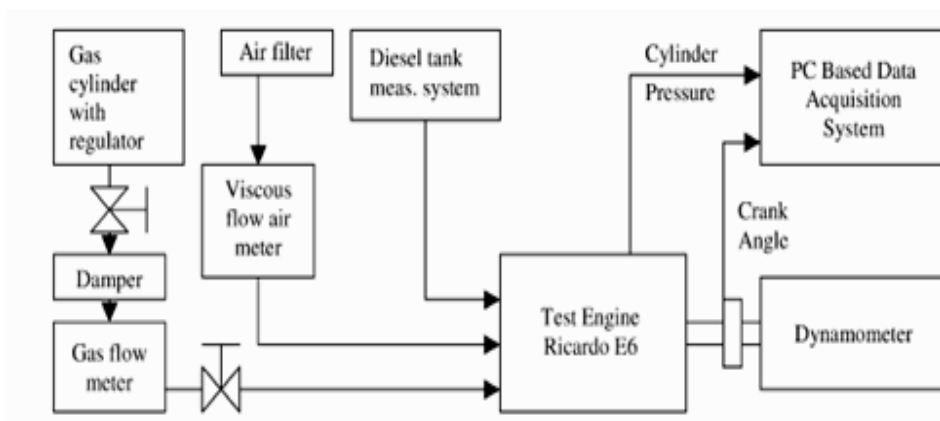


Fig. 1, Schematic diagram of the engine test rig.

Experiments have been carried out after running the engine for some time until it

reaches steady state and oil temperature is at $60^{\circ}\text{C}\pm 5$, and cooling water temperature is at

70 °C±5. The engine design and operating parameters have been varied at the following levels:

1. Type of fuel included pure diesel fuel (base case as normal diesel engine), dual fuel of diesel and LPG.
2. The engine load was varied from no load to full load.
3. The pilot diesel fuel injection timing was varied from 20 to 45 °BTDC in steps of 5°.
4. The engine speed was varied from 1000 to 2100 rpm.

In the first case, engine was operated with liquefied petroleum gas (LPG), which was mixed with suction air having 30 and 60% LPG on volume basis along with the pure diesel through timed cylinder injection. The fuel injection system was adjusted to supply lesser diesel during the operation with air-LPG mixture for smooth operation. Tests of engine performance and exhaust emissions on diesel fuel alone were conducted as a basis for comparison. Engine was run on no load condition and its speed was adjusted to 1500 rpm + 20 rpm by adjusting the screw provided with the fuel injector pump. The engine was run to attain uniform speed, and then it was gradually loaded. The experiments were conducted at full load level, and the engine was run for at least 3 min at that load. The experiments were replicated three times.

A simple, low cost air-LPG mixing device, designed as shown in **Fig. 2**, was used to mix LPG with inlet air during suction stroke.

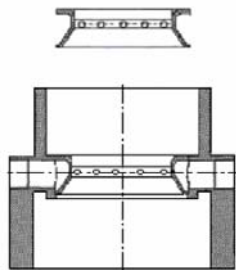


Fig.2 The cross sectional drawing of the mixer

RESULTS and DISCUSSIONS

LPG introduced as gas in the combustion chamber, so its proportion took a part of air

share, this reduction in air volume reflected as a reduction in volumetric efficiency of dual engine compared with diesel engine, as **Fig. 3** represents.

For the load characteristics at 1500 rpm, a comparison between the bsfc of LPG–diesel dual fuel and diesel is shown in **Fig. 4**. The bsfc of both engines are almost equal when the load ratio is less than 35 per cent, when the load ratio is larger than 35 per cent, the bsfc of dual fuel is slightly less than that of diesel. The dual-fuel can be electronically controlled that the engine is fuelled with dual fuel when its load ratio is larger than 35 percent and with diesel when its load ratio is less than 35 percent.

Fig. 5 shows the relationship of brake thermal efficiency and brake power for the two fuel systems. Increasing LPG portion to 60% increased indicated thermal efficiency, due to high calorific value of LPG with low mass used in this mode, compared with diesel fuel.

Fig. 6 represents the relationship between exhaust gas temperatures and bp. Exhaust gas temperatures increased with load increase, and reduced with increasing LPG in mixture. Engine operation at optimum injection timing with LPG presence, gave fast and efficient combustion, so when the piston run down in power stroke, all the fuel had been burned, and all burned gases would be cooled in this stroke before exhaust valve opened, emitting lower exhaust gas temperatures.

The comparison between bmep of the external characteristics of LPG–diesel dual fuel and diesel at variable engine speeds is shown in **Fig. 7**. The bmep of the dual fuel is slightly less than that of diesel at low speeds, so the dynamic characteristic of the dual-fuel engine almost does not decline. Although the LPG pre-mixture causes a small decrease in the engine's intake air amount, the torque of dual fuel is almost equal to that of diesel which makes the heat value of the mixture in the cylinder recover, and dual fuel has a higher combustion efficiency than diesel.

The comparison between the bsfc's of LPG–diesel dual fuel and diesel is shown in **Fig. 8**. The bsfc of dual fuel is less than that of diesel. Because of the good quality of the mixture and high combustion efficiency of dual fuel, the bsfc of dual fuel at full load is low.

During the experiment, injection timings were adjusted based on the opening time of the needle 5° crank angle (CA) before top dead center (BTDC) each time. **Fig. 9** shows engine brake specific fuel consumption versus fuel injection timings fueled with two dual fuels and neat diesel. The results show that the brake specific fuel consumption decreases remarkably when operating on the dual mode. The brake specific fuel consumption of the fuels has the minimum value at the fuel injection timing of 27° BTDC (in 60% LPG case) and at 30° BTDC (in 30% LPG cases). The brake specific fuel consumption increased sharply while retarding the fuel injection timing. It is noted that further retarding fuel injection timing was not appointed in this study due to the rapid increase in brake specific fuel consumption under the speed conditions. The brake specific fuel consumption decreased with the increase of LPG proportion in the mixture and the decreasing trend becomes gentle.

Fig. 10 shows the bmep versus the fuel injection timings. Retarding injection timing increases the cylinder air temperature at the timing of fuel injection beginning. This will cause high increment in vaporization of injected fuel inside the cylinder. This leads to decrease in both the physical preparing time and the chemical reaction time before ignition starts, resulting in shortening the ignition delay. Advancing injection timing in the other hand will improve combustion, especially for LPG mixtures, causing higher bmep.

Retarding injection timing increased exhaust gas temperatures, as well as, increasing the LPG proportion in the blends, as **Fig. 11** indicates. Retarding injection timing decreases the ignition delay and leading the fuel to burn after the piston move from TDC, resulting in increments in the

exhaust gas temperatures. The good mixing of LPG in air increases the combustible mixture available during the ignition delay period. This leads to increase heat released in the premixed combustion phase, and increases the peak cylinder pressure and maximum heat release rates at combustion period. These gases will run out at lower temperatures after the results are being cooled at power stroke, and at exhaust stroke.

CO concentrations for dual mode approached the resulted concentrations for diesel with little reductions at some F/A ratios, as **Fig. 12** shows, the reason for lower emission is the increased burning temperature which created local turbulence and increased flame velocity.

The effect of engine speed on CO emissions is shown in **Fig. 13**. At full load, The CO concentrations of dual fuel is almost exceeded to that of diesel fuel at low speeds, and reduced away from them at medium and high speeds. The formation of CO is related to the mixture concentration and fuel composition.

Fig. 14 shows the effect of fuel injection timings on exhaust CO emission. Exhaust CO emission increased while retarding the fuel injection timings, and decreased with increasing the LPG proportion in the blends. Retarding the fuel injection timing decreases the amount of fuel burned in the premixed combustion phase and increases the amount of fuel burned in the subsequent diffusive combustion phase. The latter phase always takes place in a rich mixture environment and easily produces the incomplete burning product CO. Increasing the LPG proportion in the blends decreases the carbon fraction in the blends.

CO₂ concentrations reduced with dual mode for all examined F/A ratios, as **Fig. 15** represents. LPG existence increased hydrogen to carbon percentage in the fuel, which improved the combustion and reduced resulted CO₂.

Fig. 16 shows the effects of fuel injection timings on exhaust CO₂ emissions. CO₂ emission increased while retarding the fuel

injection timings and it decreased with increasing the LPG proportion in the fuel mixture, and when working with optimum injection timing. The molecular structure of LPG is simpler than that of diesel fuel and can be oxidized more easily, as well as increasing the LPG proportion in the blends decreased the carbon fraction in the blends.

The variation of NO_x emission with load at variable fuel- air ratios is shown in **Fig. 17**. It increased marginally in the case dual fuel operation as well as in case of neat diesel fuel, but for dual mode the resulted concentrations were less than that for diesel. LPG fuel forms a homogeneous mixture with air, and it leads to nearly complete combustion, resulting in high temperature inside the engine during combustion. It increases the possibility of NO_x formation, but high temperatures leads to high dissociation rates. The resultants of these reactions will freeze when the piston goes down in power stroke, and combustion chamber cooled by cooling water. Also, the pre-chamber combustion figure improves turbulence and complete burning. Moreover working with OIT reduced the time available for NO_x formation. The resultant NO_x concentrations were the outcome of all these parameters.

The comparison between the nitrogen oxides (NO_x) emissions of the dual fuel and diesel at variable engine speeds is shown in **Fig. 18**. The NO_x emission of dual fuel is almost less than that of diesel at many speeds ranges. The change in the creation of NO_x is determined by factors such as the mixture concentration and the combustion temperature, which are slightly changed. Therefore, the NO_x emission is slightly changed.

The effect of fuel injection timings on NO_x emission is given in **Fig. 19**. The study shows that both the fuel injection timing and LPG proportion in the mixtures affect exhaust NO_x emission remarkably. The concentration of NO_x decreased while retarding the fuel injection timings and increased with increasing the LPG proportion in the blends. Retarding the fuel injection timing reduces ignition delay period, causing reductions in the amount of heat released, cylinder gas

temperature and the exhausted NO_x. Increasing LPG proportion increases the ignition delay period, the amount of heat release, and the cylinder gas temperatures. This will contribute to increase NO_x emission. The emitted NO_x concentration will be the resultant of these two parameters. So, some measures need to be implemented to reduce NO_x emission when operating with the diesel-LPG blends such as exhaust gas recirculation.

Fig. 20 shows unburned hydrocarbons (UBHC) concentrations for tested fuels at variable fuel-air ratios, the results prove that using LPG in dual mode reduced UBHC for all tested ratios, contrary to reference (Gunea, 1998) who demonstrated increase in HC emissions with dual mode, may be because the researchers didn't use optimum injection timing for tested fuels as took place in this study.

The comparison between the HC emissions of the dual fuel and diesel for variable engine speeds is shown in **Fig. 21**. At full load, the HC emission of dual fuel is higher than that of diesel, which is caused by two key factors; the first is that the LPG pre-mixture is scavenged to outside from the cylinder in the overlap period of the valves. The second is that the LPG pre-mixture that is pressed into the cooled crevices during compression stroke is difficult to burn.

Fig. 22 shows the effects of fuel injection timings on exhaust HC emission. HC emission increases while retarding the fuel injection timings and it decreases with increasing the LPG proportion in the fuel mixture. Over-lean and over-rich mixtures will increase exhaust HC emission. The high cylinder gas temperature will promote the post flame HC oxidation. Furthermore, the molecular structure of LPG is simpler than that of diesel fuel and can be oxidized more easily. The total HC emission decreases while increasing the LPG proportion in the blends. The study also indicates that the effect of LPG in the mixtures on the reduction of HC emission is stronger than that from advancing the fuel injection timing. Thus, diesel-LPG dual fuel is beneficial to the reduction of HC emission.



CONCLUSIONS

Experiments were conducted to study the performance and emission characteristics of IDI diesel engine in dual fuel mode of operation. Liquefied petroleum gas was introduced in the inlet manifold for various loads, with diesel as an ignition source. The following conclusions have been arrived at,

1. Dual fuel operation of LPG exhibits lower exhaust gas temperatures as compared to diesel operation.
2. A perceivable reduction in HC, CO and CO₂ emissions was observed with LPG operated dual fuel mode.
3. The brake specific fuel consumption of the engine fueled with diesel-LPG blends is lower than that of diesel fuel under the optimum injection timings.
4. At the same engine speed and brake mean effective pressure, engine exhausts HC, CO, and CO₂ emissions increased and exhaust NO_x emission decreased while retarding the fuel injection timing. Exhaust HC, CO, and smoke emissions decreased and exhaust NO_x emission increased while increasing the LPG proportion in the mixtures.

The exhaust emissions (NO_x and UBHC) were improved in dual fuel operation. This indicates that fuel property is one of the most important parameters, which affects the exhaust emissions. Diesel powered engines can be converted to operate with dual fuel. So the air quality will be better. LPG can be introduced to the intake manifold to obtain the precise control of the amount of fuel, admitting to the cylinder. Also, by increasing the LPG proportion in dual fuel operation a further improves in exhaust emissions can be obtained.

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Table 1 Properties of fuels used

Property	Diesel	LPG
Calorific value , kJ/kg	44500	50000
Self ignition temperature, °C	725	525
Boiling point range , °C	260-320	-34
Ignition delay period, s	0.002	-
Flame propagation rate, cm/s	10.5	83.7
Flame temperature, °C	1715	1985
Surface tension, dynes	32	-
Viscosity at 39 °C, centistokes	2.7	-
Specific gravity at 32 °C	0.83	0.43
Sulpher content by weight, %	0.8	0.0112

Table 2 Engine characteristics

Model	Ricardo E6
Type	IDI with the pre-combustion chamber
Number of cylinders	1
Bore × Stroke (mm)	76.2×111.1
Cycle	4-stroke
Compression ratio	From 5 to 22
Maximum power (kW)	9 naturally aspirated
Maximum speed (rpm)	3000
Injection timing	Variable



Nomenclature

ATDC	After top dead center
BTDC	Before top dead center
°CA	Crank angle degrees
CO	Carbon monoxide
CO ₂	Carbon dioxide
CR	Compression ratio
FAR	Fuel-air ratio
OIT	Optimum injection timing
NO _x	Nitrogen oxides
UBHC	Unburned hydrocarbons
bmep	Brake mean effective pressure
bp	Brake power
bsfc	Brake specific fuel consumption

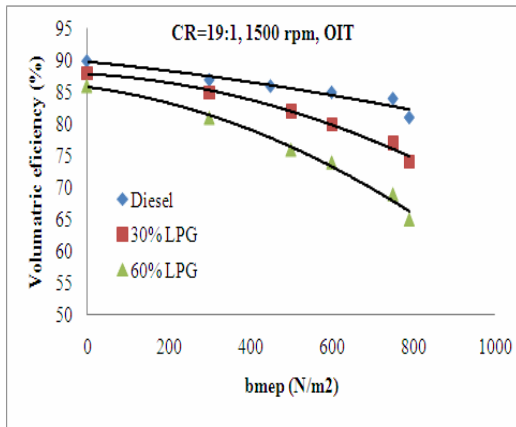


Fig. 3, Effect of LPG addition on volumetric efficiency at variable loads

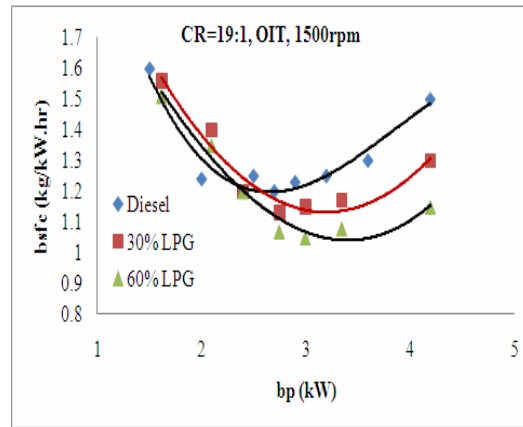


Fig. 4, Effect of LPG addition on bsfc at variable brake powers

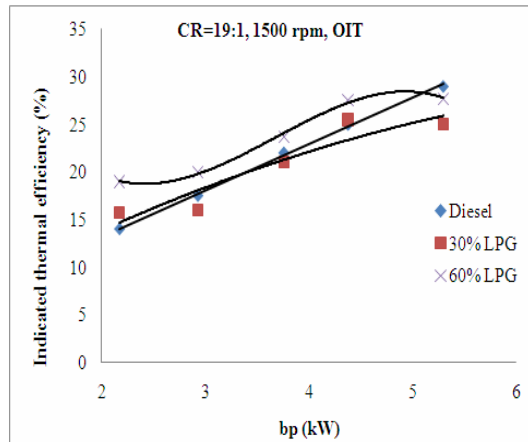


Fig. 5, Effect of LPG addition on indicated thermal efficiency at variable brake powers

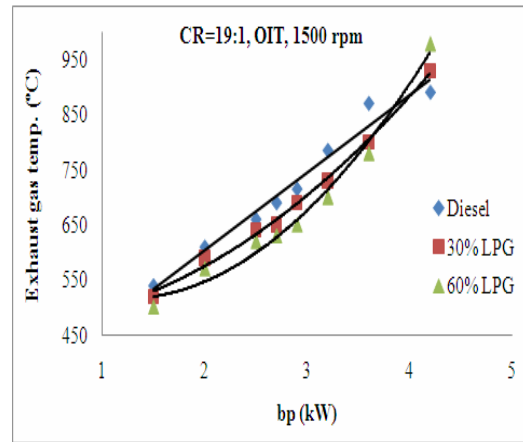


Fig. 6 Effect of LPG addition on exhaust gas temperatures at variable loads

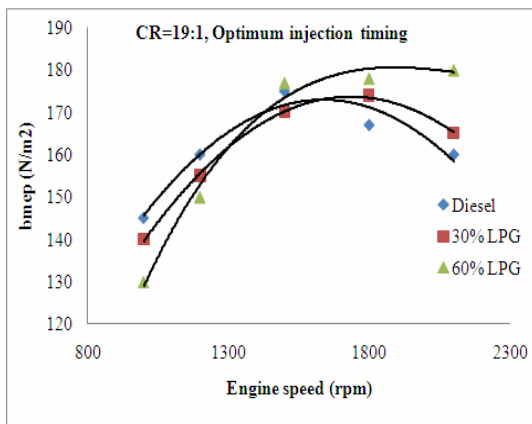


Fig. 7, Effect of LPG addition on bmep at variable engine speeds

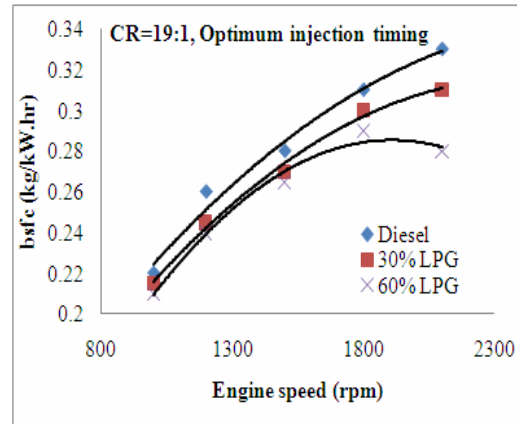


Fig. 8, Effect of LPG addition on bsfc at variable engine speeds

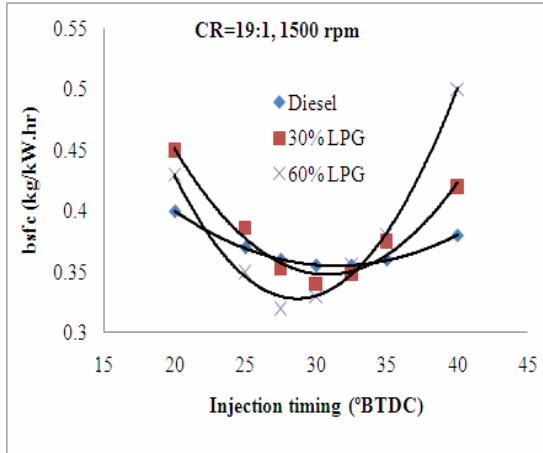


Fig. 9, Effect of LPG addition on bsfc at variable injection timing

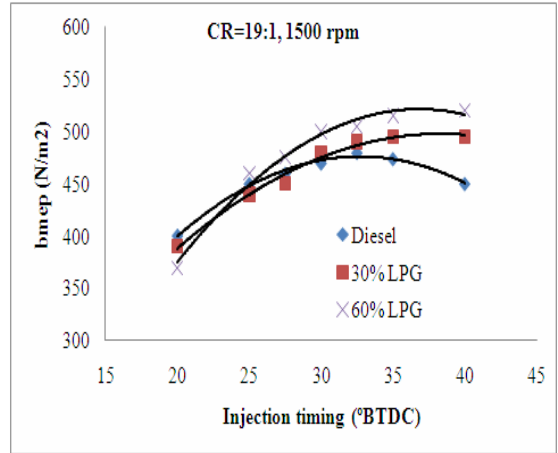


Fig. 10, Effect of LPG addition on bmep at variable injection timing

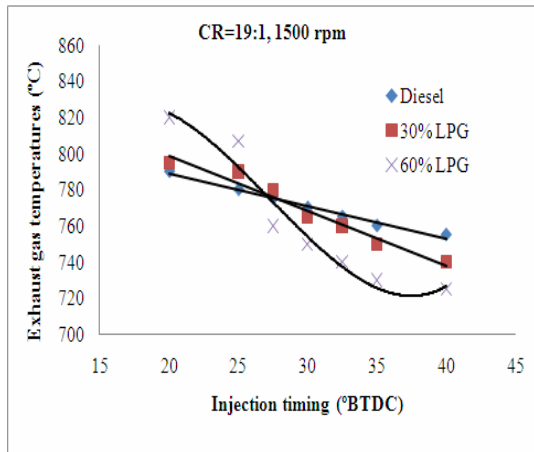


Fig. 11, Effect of LPG addition on exhaust gas temperatures at variable injection timing

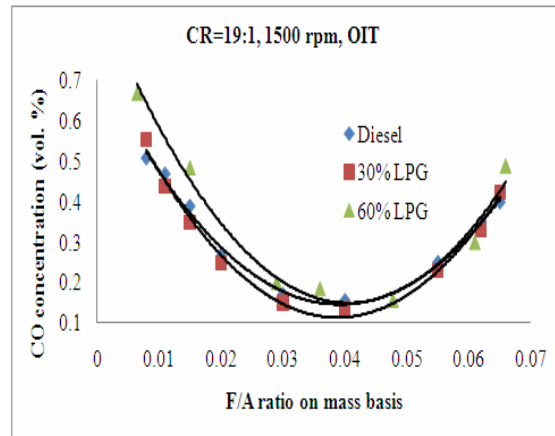


Fig. 12, Effect of LPG addition on CO concentrations at variable fuel-air ratios

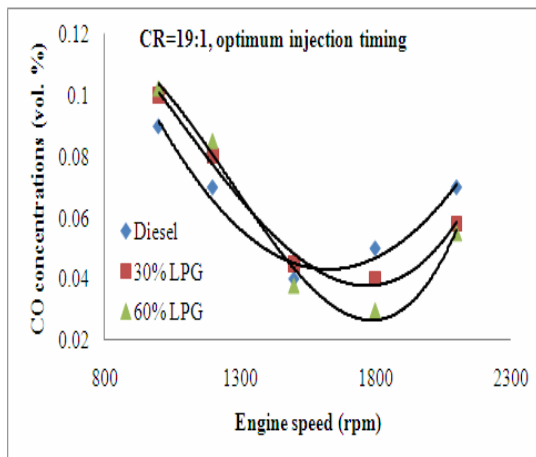


Fig. 13, Effect of LPG addition on CO at variable engine speeds

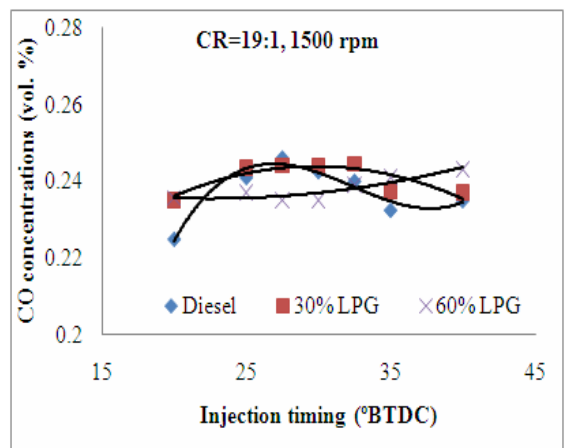


Fig. 14, Effect of LPG addition on CO at variable injection timing

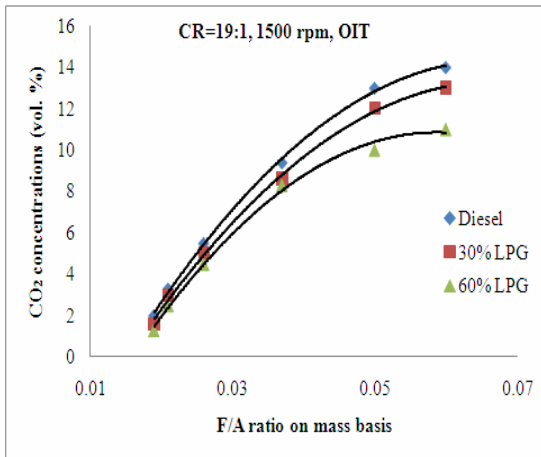


Fig. 15, Effect of LPG addition on CO₂ concentrations at variable fuel-air ratios

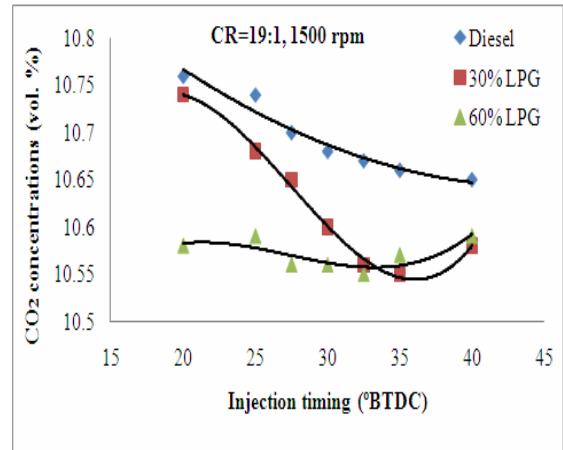


Fig. 16, Effect of LPG addition on CO₂ at variable injection timing

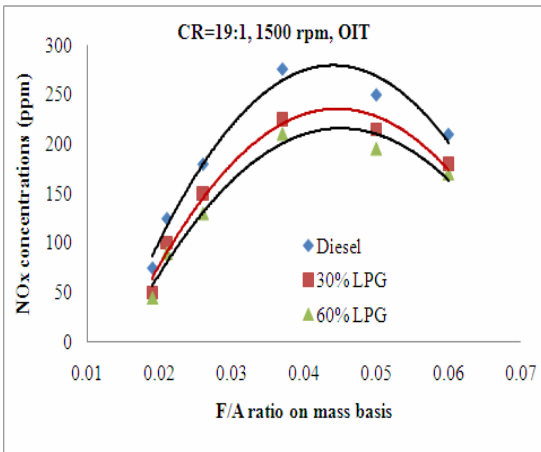


Fig. 17, Effect of LPG addition on NO_x concentrations at variable fuel-

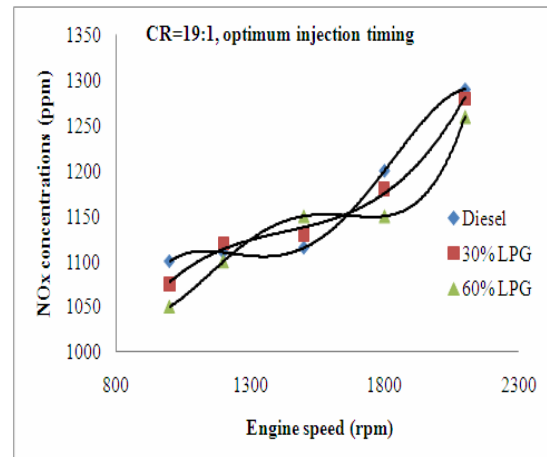


Fig. 18, Effect of LPG addition on NO_x at variable engine speeds

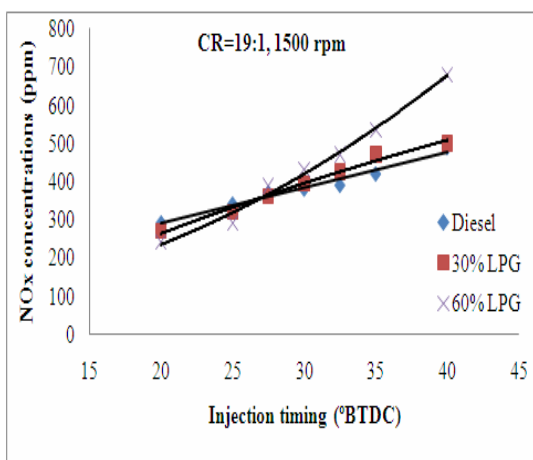


Fig. 19, Effect of LPG addition on NO_x at variable injection timing

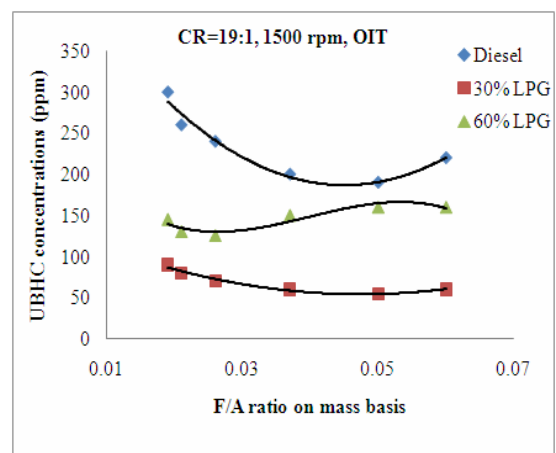


Fig. 20, Effect of LPG addition on UBHC concentrations at variable fuel-air ratios

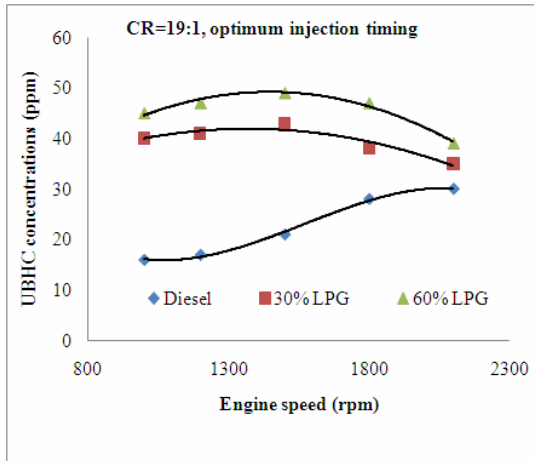


Fig. 21, Effect of LPG addition on UBHC at variable engine speeds

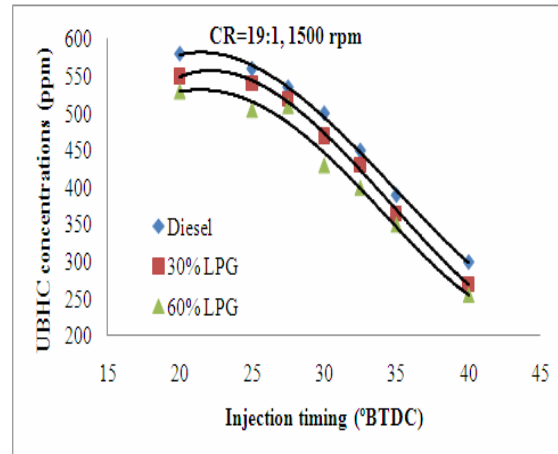


Fig. 22, Effect of LPG addition on UBHC at variable injection timing