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Characterization of Lean Misfire Limits of Mixture Alternative Gaseous Fuels Used for Spark Ignition Engines

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Abstract

Increasing on gaseous fuels as clean, economical and abundant fuels encourages the search for optimum conditions of gas-fueled internal combustion engines. This paper presents the experimental results on the lean operational limits of Recardo E6 engine using gasoline, LPG, NG and hydrogen as fuels. The first appearance of almost motoring cycle was used to define the engine lean limit after the fuel flow was reduced gradually. The effects of compression ratio, engine speed and spark timing on the engine operational limits are presented and discussed in detailed. Increasing compression ratio (CR) extend the lean limits, this appears obviously with hydrogen, which has a wide range of equivalence ratios, while for hydrocarbon fuel octane number affect gasoline, so it can' t work above CR=9:1, and for LPG it reaches CR=12:1, NG reaches CR=15:1 at lean limit operation. Movement from low speeds to medium speeds extended lean misfire limits, while moving from medium to high speeds contracted the lean misfiring limits. NOx, CO and UBHC concentrations increased with CR increase for all fuels, while CO_2 concentrations reduced with this increment. NOx concentration increased for medium speeds and reduced for high speeds, but the resulted concentrations were inconcedrable for these lean limits. CO and CO_2 increased with engine speed increase, while UBHC reduced with this increment. The hydrogen engine runs with zero CO, CO_2 and UNHC concentrations, and altra low levels of NOx concentrations at studied lean misfire limits.

Key words: Misfire limit, LPG, NG, hydrogen

وصف لحدود الأطفاء الضعيفة لأنواع من الوقود الغازي البديل المستخدم في محرك اشتعال بالشرارة

الخلاصة

شجع التأكيد المتواصل على الوقود الغازي كوقود نظيف واقتصادي ومتوفر البحث عن الظروف المثالية لمحركات الاحتراق الداخلي العاملة بالوقود الغازي. وتقدم هذة الورقة النتائج العملية على حدود التشغيل الضعيفة لمحرك ريكاردو E6 يستخدم الكازولين، الغاز النفطي المسال، الغاز الطبيعي والهيدروجين، ولقد استخدمت حالة ظهور أول إطفاء للشرر لتعريف حدود المحرك الفقيرة بعد تقليل تدفق الوقود تدريجيا. درس تأثير نسبة الانضغاط، سرعة دوران المحرك، وتوقيت الشرر على أداء وملوثات المحرك وفحصت بالتفصيل. إن زيادة نسبة الانضغاط تمدد حدود الاشعال الضعيفة، ويظهر هذا الأمر بوضوح عند استخدام الهيدروجين، والذي له مجال نسب مكافئة عريض، بينما لأنواع الوقود الهيدروكاربوني المستخدمة بالبحث كان الرقم الاوكتاني يؤثر على الكازولين لذا لم يعمل محركه أعلى من CR=9:1، أما محرك الغاز النفطي المسال فوصل إلى CR=12:1، وللغاز الطبيعي كانت CR=15:1، كما أن الانتقال من سرع بطيئة الى سرع متوسطة يزيد من الحدود الضعيفة لكل أنواع الوقود المدروسة، بينما التحرك من سرع متوسطة إلى عالية يقلل من حدود الإطفاء للمحرك. تزداد تراكيز NOx، CO و UBHC بزيادة نسب الانضغاط لكل أنواع الوقود المستخدم في الدراسة، بينما تقل تراكيز CO_2 بهذه الزيادة، وتزداد تراكيز NOx للسرع المتوسطة وتقل للسرع العالية، كما تزداد تراكيز CO وCO_2 بزيادة سرعة المحرك، بينما تقل ملوثات UBHC بهذة الزيادة، وبينت الدراسة أن محرك الهيدروجين يعمل عند تراكيز CO، CO_2 و UBHC تساوي صفر، وبمستويات متدنية جدا من تراكيز NOx عند حدود الإطفاء المدروسة.

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الكلمات الدالة: حدود الاطفاء، الغاز النفطي المسال، الغاز الطبيعي، هيدروجين

Nomenclatures

ATDC	After top dead center
BTDC	Before top dead center
ºCA	Crank angle degrees
CO	Carbon monoxide
CO_2	Carbon dioxide
CR	Compression ratio
FAR	Fuel-air ratio
OST	Optimum spark timing
NOx	Nitrogen oxides
UBHC	Unburned hydrocarbons
Bmep	Brake mean effective pressure
bp	Brake power
bsfc	Brake specific fuel consumption
Ø	Equivalence ratio

Introduction

Numerous studies have been conducted to improve engine performance and reduce exhaust emissions. Although there have been advances in both areas, still more research needs to be done in order to meet the strict upcoming emission regulations. There are many alternatives being researched to improve the engine-out emission further. [1]

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. They have distinct superior merits to those of conventional liquid fuels in applications, whether those for spark ignition or compression ignition engines. Additionally, most of these fuels can produce more favorable exhaust emission characteristics that can meet better the ever increasingly stringent emission regulations combined with enhanced power production efficiency and improved engine operational life. [2]

The most common of the alternative fuels is natural gas that is usually made available after processing as "pipeline processed natural gas". It is supplied for engine applications normally as compressed natural gas, (CNG), or occasionally in its cryogenic liquid form, (LNG). The composition of the gas in its natural untreated state varies widely depending on its source, treatment and local conditions. However, after it's processing when destined for transport to its ultimate consumption points its composition becomes less widely variable and made up mostly of methane. [3, 4]

Methane is the principal component of natural gas. Normally more than 90% of natural gas is methane. The auto-ignition temperature for natural gas is higher than gasoline and diesel fuel. Additionally, natural gas is lighter than air and will dissipate upward rapidly if a rupture occurs. Gasoline and diesel will pool on the ground, increasing the danger of fire. Compressed natural gas is non-toxic and will not contaminate groundwater if spilled. Advanced compressed natural gas engines guarantee considerable advantages over conventional gasoline and diesel engines. Compressed natural gas is a largely available form of fossil energy and therefore non-renewable. [5, 6]

Another common source of gaseous fuels involves the higher molecular weight components of natural gas in the form of Liquefied Petroleum Gases, (LPG), which can be liquefied under pressure at ambient temperature. Usually, the main component of these fuel gases is n-propane. On this basis, often engine performance with pure or even commercial propane is considered to be represented adequately by engine operation with LPG gas mixtures. [7, 8]

LPG is blend of the low hydrocarbon molecule compound such as butane, propane, etc. They are produced from natural gas treated factory or oil refinery. LPG has high ignition temperature and it is safety: In the normal temperature it can be liquefied at 1.6MPa - 2MPa. It is easy to be used. It has high heat value, and its gaseous state makes it easy to mix with air. Perfect combustion redounds to improve power output. Good antiknock because of high octane. Soot can be obviously reduced because low carbon compound. Gas fuel will not dilute the engine lubrication oil. The replacement period of lubrication oil could be longer. 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. [9, 10]

Hydrogen as an engine fuel is well recognized to have unique and excellent combustion characteristics while producing effectively negligible exhaust emissions, except for NOx. Moreover, the addition of H_2 to relatively slow burning fuels such as CH_4 was shown to accelerate the flame propagation rates and extend the lean operational limits. This is perhaps despite the well recognized limitations associated with its engine application arising from the need to develop improved methods for its economic manufacture, ease of availability, storage and transport while ensuring the safety of its handling. [11, 12]

In practice, much of the gaseous fuels available are usually mixtures of various fuel and some diluent constituents that can vary widely in nature and concentration depending on the type of fuel and its origin. [13]

The operation of spark ignition engines on fuel lean mixtures rather than stoichiometric combined with catalytic exhaust gas treatment is highly desirable for achieving low exhaust emissions, especially those of NOx, combined with high work output efficiency and improved durability. This is in principle better achieved with gaseous fuels. However, the continued leaning of the intake mixture leads eventually not only highly reduced power output and efficiency but eventually also to increased cyclic irregularity and increased emissions especially those of unburned fuel and carbon monoxide. [14]

Engines operated under very lean conditions may experience ignition failure, flame quenching, and incomplete propagation before the exhaust valve opens, which are prime sources of air pollution. Obviously, the knowledge of such misfire limits help the engine developers to decide on how far they can go in the lean side to achieve best fuel economy and minimum exhaust emissions without worsening drivablity and smooth operation. Moreover, such limits are needed for the prediction of the combustion duration of a given fuel-air mixture. [15]

The definition of an engine lean limit is not universally accepted. For example, the lean flammability limit (LFL) under quiescent conditions is known to be an inherent fuel property and independent of an engine design. The lean ignition limit (LIL), on the other hand, is defined as the leanest mixture that will form a flame kernel for a given ignition system. In spark ignition engines where conditions cannot be idealized, operational lean limits commonly referred to as a lean misfire limit (LML) are given rather than LFL or LIL. The definition and characterization of LML have been somewhat arbitrary. A unique definition is difficult a variety of methods to indirectly measure and define such a limit have been followed. Some examples of those definitions are equivalence ratio at which first misfire occurs, selected frequency of total misfires, selected frequency of cylinder peak pressure, specific amount of hydrocarbon or carbon monoxide content in exhaust gases, number of audible misfires counted over a time period, variation of the area under the measured cylinder p-v diagram, and variation in mean effective pressure. Badr et al. [16] attempted to relate the misfiring limit to the flammability limits under quiescent conditions at cylinder temperatures and pressures similar to those at the time of spark release. [17, 18]

Complete understanding of the phenomena related to the misfiring process in spark ignition engines is not available at this time. Therefore, the main objective of this study to shed light on such limiting phenomena and to examine their relations to engine variables such as speed, spark timing, and compression ratio, and compare the results for the three alternatives with gasoline.

Test setup

A single cylinder, naturally aspirated, four stroke, variable compression ratio, ignition timing and speed, Ricardo E6 was used in this study, and further details regarding this engine are given in table 1.

The engine was operated with gasoline; NG, LPG and pure hydrogen. In practice, much of the gaseous fuels available are usually mixtures of various fuels and some diluents, constituents that can vary widely in nature and concentration, depending on the type of fuel and its origin. In this work the gasoline used was Iraqi Dora refinery production with octane No. 82, the LPG fuel produced from Al Taji Gas Company; consist of ethane 0.8%, 18.47% isobutane, 47.8% propane and 32.45% butane. NG used was produced from Iraqi Northern Gas Company; consist of 84.23% methane, 13.21% ethane, 2.15% propane, 0.15% isobutane, 0.17% n. butane and 0.03% pentane. Hydrogen produced from Al Mansur Company with 99.99% purity.

Gasoline fuel was supplied to the carburetor type Zenith WIP supplied with a choke, 26 mm, consists of a main variable spray, supplied with needle valve to control the gasoline flow rate through spray opening.

Hydrogen and natural gas (NG) were drawn from a high-pressure cylinders; this pressure was reduced to one atm through a pressure regulator. It was then passed through a control valve for regulating the amount of gas, the gas mass flow rate was metered using chocked nozzles meter, which also was used as a flame trap to arrest and control flash back if any.

Commercially available liquefied petroleum gas (LPG) delivery system consists of fuel tank (80 kg of LPG), fuel filter, solenoid valve, gaseous fuel flow rate measuring device (orifice plate), damping tank, gas adaptor.

A piezo electric pickup fitted into the combustion chamber, along with a charge amplifier, oscilloscope and Iwatsu Signal Analyzer enable the measurement of cylinder pressure.

The following instruments were used for the analysis of the emissions:

- A non – dispersing infrared analyzer for CO.
- A magnetic oxygen analyzer for O_2.
- HC analysis by flame ionisation detector, FID.
- A chemiluminescence analyzer for NO and NO_2.

The equivalence ratio which was determined from the measured air and fuel flow rates to the engine, defied as:

$$\emptyset = \frac{\text{stoichiometric fuel/air ratio}}{\text{actual fuel/air ratio}}$$

The following equations were used in calculating engine performance parameters [19]:

1- Brake power

$$bp = \frac{2\pi * N * T}{60 * 1000} \quad kW$$

2- Brake mean effective pressure

$$bmep = bp \times \frac{2 * 60}{V_{sn} * N} \quad kN/m^2$$

3- Fuel mass flow rate

$$\dot{m}_f = \frac{v_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time} \quad {kg}/{sec}$$

4- Air mass flow rate

$$\dot{m}_{a,act.} = \frac{12\sqrt{h_o * 0.85}}{3600} \times \rho_{air} \quad \frac{kg}{sec}$$

$$\dot{m}_{a_{theo.}} = V_{s.n} \times \frac{N}{60 * 2} \times \rho_{air} \quad \frac{kg}{sec}$$

5- Brake specific fuel consumption

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \quad \frac{kg}{kW.hr}$$

6- Total fuel heat

$$Q_t = \dot{m}_f \times LCV \quad kW$$

7- Brake thermal efficiency

$$\eta_{bth.} = \frac{bp}{Q_t} \times 100 \quad \%$$

The experimental tests were conducted at internal combustion engine laboratory in University of Technology. The aim was to study the effect of some engine parameters on the lean limit. The spark timing for all fuels was varied systematically to obtain OST; it was varied with engine speed and compression ratio. The study covered the following parametric ranges:

- Spark timing from 0 to 50 °BTDC.
- Speed from 15 to 50 rps.
- Compression ratio from 6 to 16.

The ambient air conditions were 25°C and 40% relative humidity.

Due to the lack of unique definition for the operation lean limit of an engine. The present study considered the approach based on the pressure–time diagram which was used by ref. [18]. With the engine running at a set of conditions, the fuel flow was reduced gradually until the first appearance of almost motoring cycle. This indicates the loss of power output due to incomplete combustion or failure of ignition. Such a point is named here, the first misfire limit.

Results and Discussion

For 25 rps engine speed, fig.1 shows a continuous decrease in the lean limit with increasing compression ratio, with all fuels kinds. This indicates the dominant effect of the gas temperature and turbulence level at ignition point. The figure shows the high ability of hydrogen to work at extremely lean limits cannot be reached by hydrocarbon fuels, also indicates the high equivalence misfiring ratio for gasoline compared with the alternatives.

On the other hand, Fig.2 shows that for the three alternative the effect of compression ratio on the lean limit is very mild, where the resulted brake power didn't vary a lot, and there values were close to each others, while for gasoline the resulted bp was high, putting in mined, the lean limits for gasoline were varied between (Ø=0.8 to 0.78), compared with the other alternative fuels, as for LPG varied between (Ø=0.71 to 0.66), NG misfire limits varied between (Ø=0.7 to 0.61) and for hydrogen were between (Ø=0.52 to 0.26). Another important parameter was the compression ratio range, it was very narrow for gasoline (CR=5:1 to 9:1), for LPG the range was between (CR=5:1 to 12:1), NG sphere of action was between (CR=5:1 to 14:1) and for hydrogen was (CR=5:1 to 16:1).

Moreover the utilization of lean mixtures, thus reducing brake torque, can be considered as a useful method to control the engine load without throttling, so minimizing the pumping losses.

Compression ratio effect appeared obviously on indicated thermal efficiency, hydrogen had the higher efficiency levels, while NG had the minimum levels, the reason in that is the high flame propagation speed for hydrogen (varied from 2.6 m/s to 3.2 m/s) compared with the low flame speed for NG (0.32 m/s), putting in mined these points were the extra-lean limits for the fuels, this is what fig. 3 shows.

The volumetric efficiency increased with compression ratio increase, and gasoline remain hadding the higher levels, due to its liquid state in nature, so when evaporated it drown its evaporation heat from the air, causing a relatively cooling effect, while the three alternatives are in the gaseous state, they have disadvantage of low volumetric efficiency since they occupy a fraction of intake charge, and this will decrease the fresh air into the cylinder and reduces the power output compared to that of gasoline engine. LPG stayed near gasoline because of its higher molecule weight and heating value, as fig. 4 indicates.

Fig. 5 represents the relation between CR and mechanical efficiency for the fuels at lean misfire limits. Mechanical efficiency is a function of brake power and friction power, so when the friction power high, the efficiency will reduced, and this power depends on engine speed and its brake power. As the figure indicates, hydrogen mechanical efficiency overcomes the other alternatives due to the reduction in friction power. While it's resulted brake power (as figure. 1 shows) for these points was the lowest.

Brake specific fuel consumption (bsfc) reduced with compression ratio increase, as fig.6 shows. LPG obtained the highest values followed by NG, then gasoline and the lowest one was hydrogen. Lean misfire limits for hydrogen were very lean compared with the others, also hydrogen engine characterized by its low fuel consumption.

Compression ratio affect clearly appeared on optimum spark timing, fig 7. OST retarded with CR increse due to temperature increament inside combustion chamber. However, due to the slow burning velocity of natural gas and its poor lean-burn capability, the natural gas spark-ignited engine has the disadvantage of poor lean-burn capability, natural gas OST was highly advanced, and this will decrease the engine power output and increase fuel consumption as mentioned in fig. 6, As the main composition of natural gas, methane has the unique tetrahedral molecular structure with large C–H bond energies, thus demonstrates some unique combustion characteristics such as high ignition temperature and low flame speed.

while for hydrogen which followed NG in the figure, its OST appeared retarded compared with other fuels, as fig. 8 represents, although hydrogen points represents ultra-lean mixtures which cannot be reached with any known fuel, it has no problem with its burning speed because it has a wide flammability range which allows regular combustion for very lean mixtures with respect to gasoline engines, thus resulting in high efficiency and low specific fuel consumption (as fig 6 shows).

NOx concentrations increased with compression ratio increase, as fig. 9 shows, for all fuels due to increase in combustion chamber temperatures, but all the concentrations were low and under permitted limits except for zero NOx limits. Lean combustion is considered as a very promising and attractive approach for high efficiency and low NOx emissions in combustion devices such as gas turbine and internal combustion engines. While, the relatively narrow lean flammability range of hydrocarbon fuel makes it difficult to achieve the stable combustion in the lean burn regime, and this is even more severe for natural gas lean combustion.

Hydrogen engine distinguished by zero CO, CO_2 and unburned hydrocarbons (UBHC), and if there were any traces of these emissions, they resulted from lubricant oil burned inside combustion chamber, as figs. 10,11and 12 indicates.

CO concentrations increased with CR increase especially for NG, fig10. Varying the compression ratio another behavior should be superimposed on the timing dependence. High compression ratio some what increases the rate of expansion and thus the rate of pressure and temperature decay after combustion, leaving less time for oxidation of CO that has escaped the main combustion.

However, looking at fig. 11 this compression ratio dependence is obvious for CO_2, which for the former reason reduced, the increments in CO and UBHC concentrations were from CO_2 share.

The actual drawback is the reduction of the burning rate, mainly due to the lower flame speed, which results in an increase in combustion duration. Once the lean flammability limit is exceeded, the engine stability is affected by cyclic variation, the engine performance drastically drop and the rapid increase in CO and HC emissions can be observed. Anyway it is still possible to fast and efficiently burn very lean mixtures, even though additional conditions have to be created as high turbulence, turbo-charging or high compression ratio as well.

In the case of unburned hydrocarbons, fig. 12, the concentrations increased with CR increase. Although pressure and temperature inside combustion chamber increased giving better burning conditions. Misfire limit caused incomplete burning and resulted in higher UBHC concentrations.

The engine speed, which is an important parameter, indirectly affects the lean limit through its effects on turbulence, flame initiation and propagation, heat transfer, cycle time, spark timing and cyclic variations. In order to explain its role, such combined effects are uncoupled and analyzed qualitatively as follows:

The increase in the engine speed always increases the turbulence level. In the low speed range this may help flame propagation due to better mixing and larger flame front area. In addition, this may help the flame initiation since it drives the flame kernel a little distance away from the electrodes and decrease their contact area and thus reduces heat losses and recombination reactions of active species on the witted electrode surface area. Obviously, these effects reduce the engine lean limit (as fig. 13 indicates). In the high speed range, on the other hand, the very high level of turbulence would have the opposite effect since it may shutter the flame and blow the kernel off the electrodes.

As the engine speed increase, the flame has less time to complete its travel and this may cause incomplete combustion before blow down starts. In addition, the lower temperatures during the expansion stroke may cause flame quenching. Thus, the engine tendency from medium speeds to high speeds increases the lean limit, as fig. 13 indicates.

The net effect of engine speed on the lean limit depends on the relative importance of each of the above-mentioned parameters. Having in mind that the spark timing was the OST for each point for all fuels, the lean limit showed a decrease with speed up to about 30 rps beyond which it remained almost constant or increased slightly, as shown in fig. 13. This variation in lean limit was for each fuel working at HUCR and OST.

The increase in the engine speed shortens the time per cycle for intake and exhaust processes and thus causes an increase in the amount of residuals. This increases the concentrations of both diluents and radicals which have opposing effects on the lean limit, this cause to advance the OST with increasing speed, as fig. 14 represents.

Engine speed effect on NOx concentration was studied, fig. 15, these concentrations increased from low speeds to medium speeds (25 and 30 rps and sometimes for 35 rps) and reduced for high speeds (45 and 50 rps). The maximum cycle temperature was insignificant at low speeds, because of dilution increments, and longer combustion duration. NOx concentrations increased in medium speeds due to increase in cycle's temperatures, and then these concentrations reduced at high speeds, due to reaction time needed for oxygen and nitrogen shortness, and to increment in chemical dissociations due to high rise in maximum cycle temperature. Hydrogen emitted the lowest concentrations due to its altra-lean equivalence ratios, while gasoline engine emitted the highest concentrations.

CO and CO_2 concentration increased with engine speed increase, figs. 16 and 17. Flames misfire accelerate with increasing speed which increase air movement inside combustion chamber. While UBHC reduced with engine speed increased, figure(18), due to increase in mixture turbulence.

Conclusions

A study of lean misfiring limits of gas fueled spark ignition engines was conducted; the study covered a wide range of engine parameters, namely engine speed (15 to 50 rps), spark timing (0 to 50 °BTDC), compression ratio (5 to 16), the study results can be summarized as follows:

- As the compression ratio was increased, the misfiring limits decreased for all fuels, but with relatively different rates.
- Brake power increase somewhat with CR increased.
- Indicated thermal efficiency increased with compression ratio increase, and hydrogen had the highest rate.
- Volumetric efficiency increased with CR increase, and gasoline attained the highest efficiencies, while hydrogen accomplished the lower efficiencies.
- Mechanical efficiency increased with CR increase, and gasoline attained the highest efficiencies, while NG accomplished the lower efficiencies.
- Hydrogen had the lowest bsfc for all used CR's, while LPG had the highest values.
- Hydrogen is characteristic with wide flamability lean limits; it can be worked at altra-lean limits cannot be reached with any other fuel.
- OST advanced for lean limits with increasing engine speeds for all fuels.
- OST retarded with CR increase although the misfire limits were extended at this increment.
- Movement from low speeds to medium speeds extended lean misfire limits, while moving from medium to high speeds contracted the lean misfiring limits.
- NOx, CO and UBHC concentrations increased with CR increase for all fuels, while CO_2 concentrations reduced with this increment.
- NOx concentration increased for medium speeds and reduced for high speeds, but the resulted concentrations were inconcedrable for these lean limits.
- CO and CO_2 increased with engine speed increase, while UBHC reduced with this increment.
- The hydrogen engine charm appeared in zero CO, CO_2 and UNHC concentrations, and altra low levels of NOx concentrations.

The above mentioned results were discussed in relation to chemical reactions rate, turbulence, heat transfer and other parameters.

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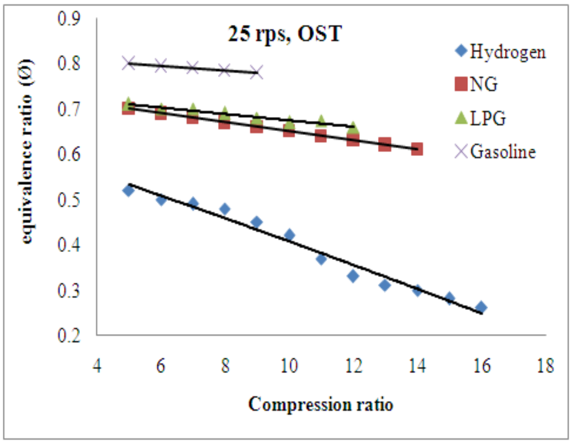
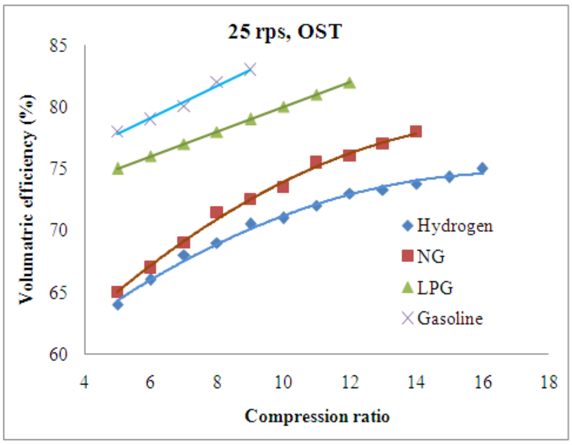
Table 1: Ricardo E6 engine geometry and operating parameters.

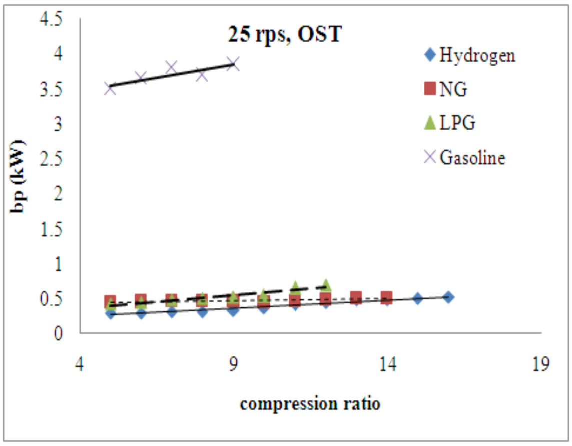
Model	Recardo E6
Displaced Volume	504 cm3
Bore	76.2mm
Stroke	111.1mm
Exhaust Valve Open	43° BBDC (at 5 mm lift)
Exhaust Valve Close	6° ATDC (at 5 mm lift)
Inlet Valve Open	8° BTDC (at 5 mm lift)
Inlet Valve Close	36° ABDC (at 5 mm lift)
Speed	1000-3500 RPM



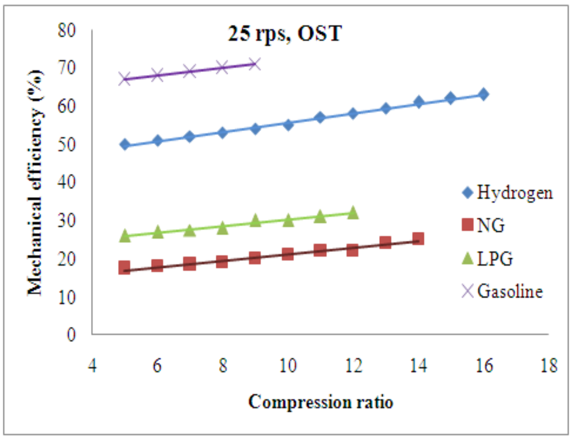
Figure(1): CR effect on lean misfire limit equivalence ratios for the four fuels



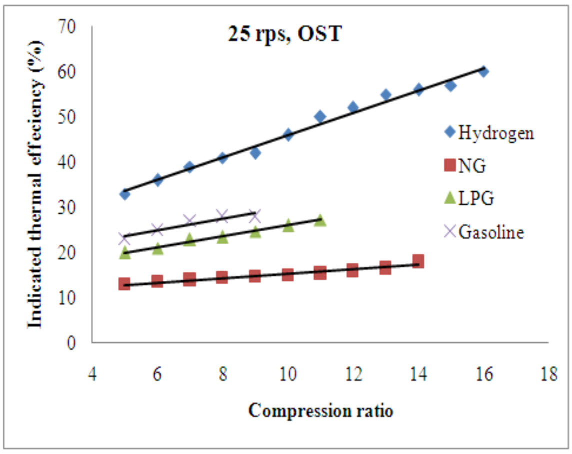
Figure(4):CR effect on lean misfire limit volumetric efficiencies for the four fuels



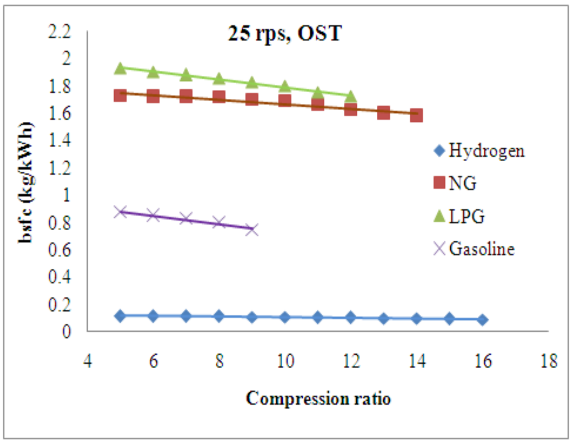
Figure(2): CR effect on lean misfire limit brake power for the four fuels



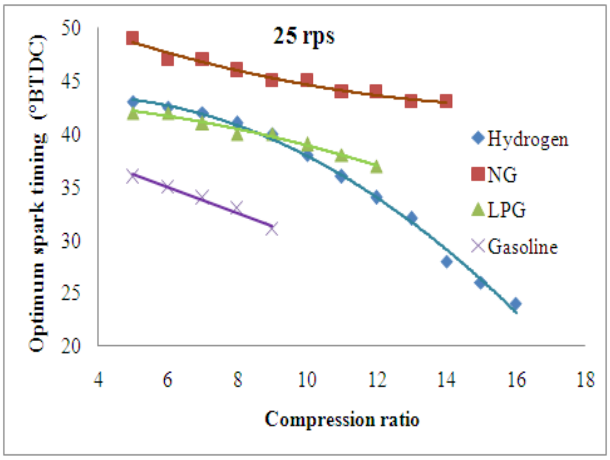
Figure(5): CR effect on lean misfire limits mechanical efficiencies for the four



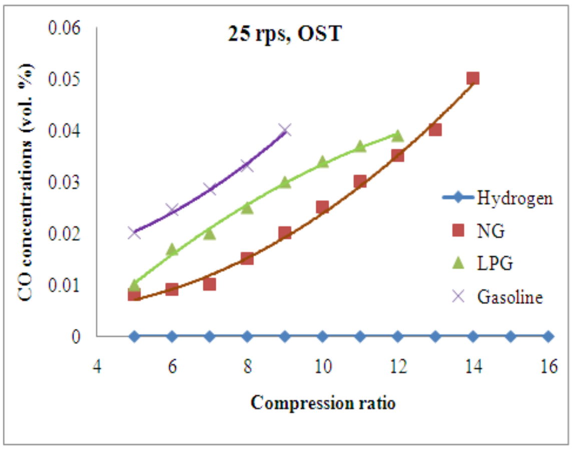
Figure(3): CR effect on lean misfire limits indicted thermal efficiencies for the four fuels



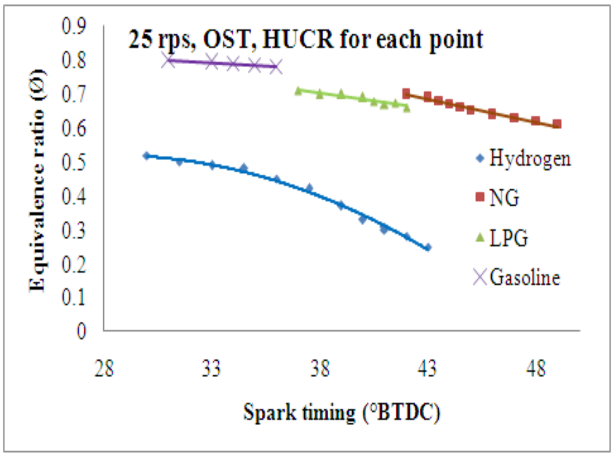
Figure(6): CR effect on lean misfire limits brake specific fuel consumption for the four fuels



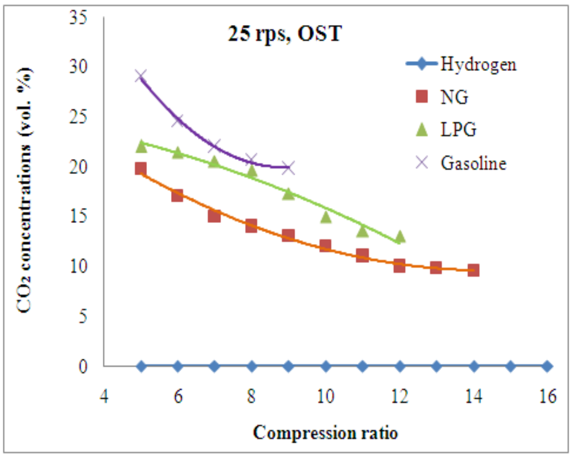
Figure(7):CR effect on OST for lean misfire limit equivalence ratios for the



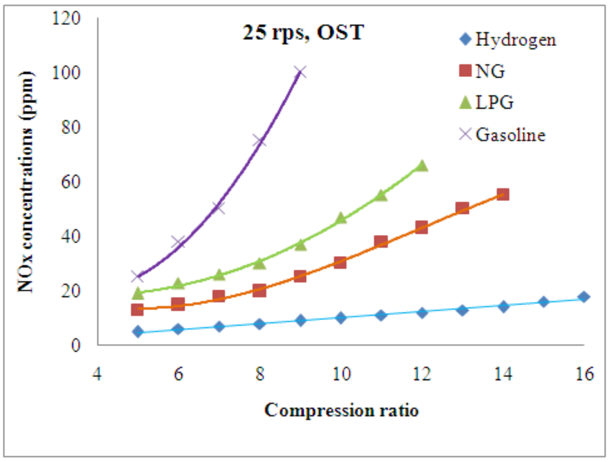
Figure(10): CR effect on CO concentrations for lean misfire limit equivalence ratios for the four fuels



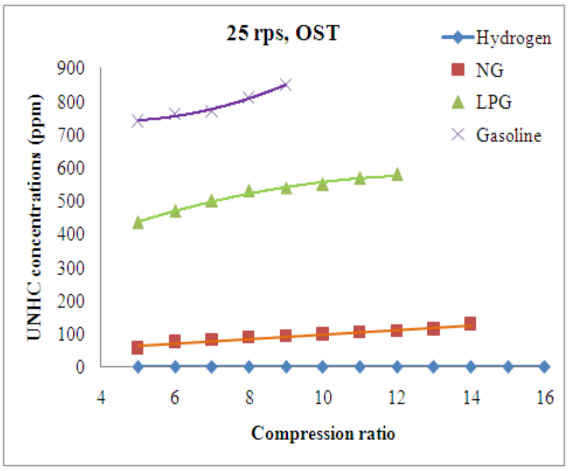
Figure(8): Spark timing effect on lean misfire limit equivalence ratios for the four



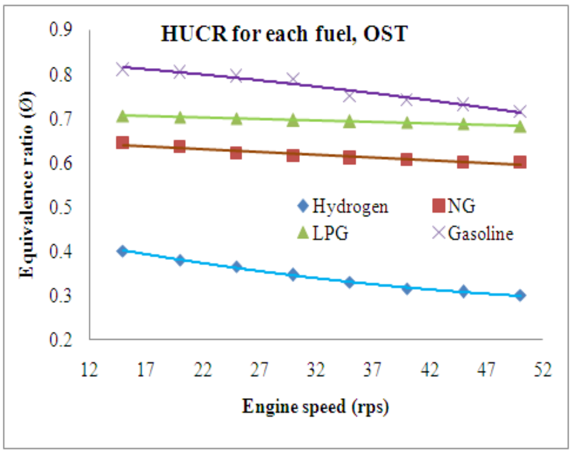
Figure(11): CR effect on CO_2 concentrations for lean misfire limit equivalence ratios for the four fuels



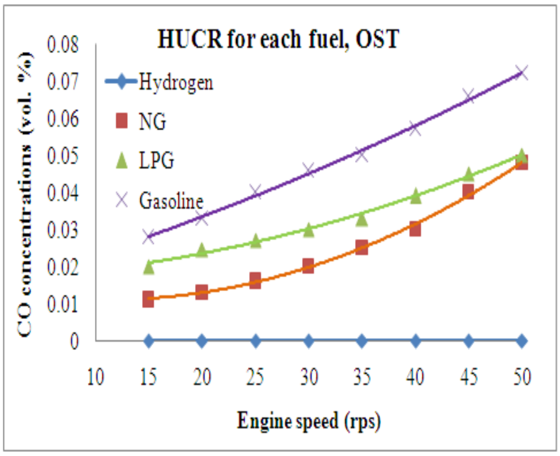
Figure(9): CR effect on NOx concentrations for lean misfire limit equivalence ratios for the four fuels



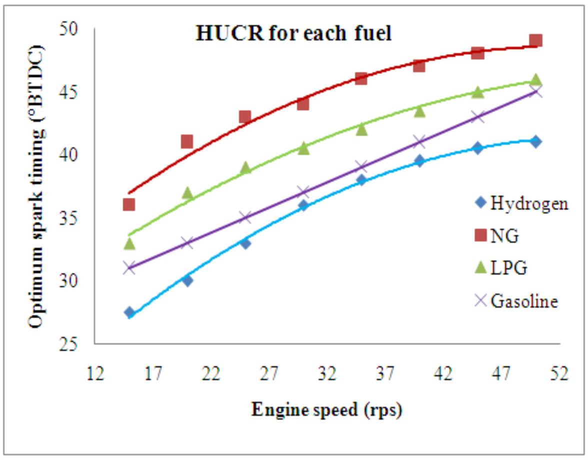
Figure(12): CR effect on UBHC concentrations for lean misfire limit equivalence ratios for the four fuels



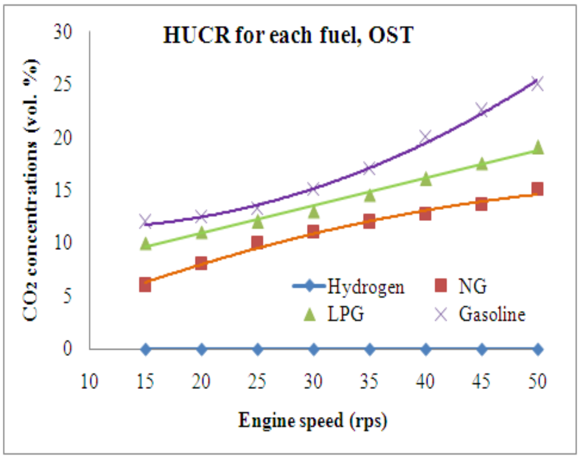
Figure(13): Engine speed effect on lean misfire limit equivalence ratios for the



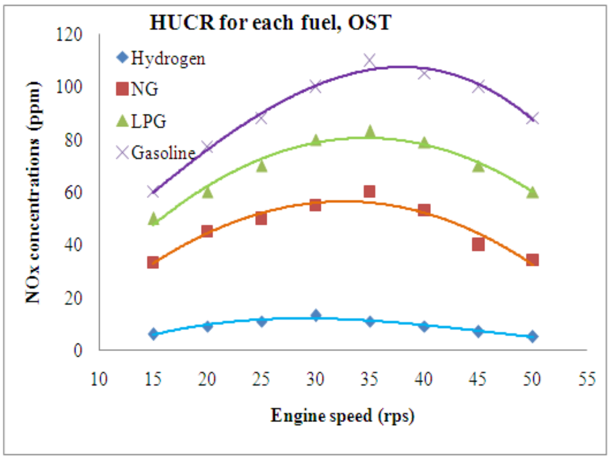
Figure(16):Engine speed effect on CO concentrations for lean misfire limit equivalence ratios for the four fuels



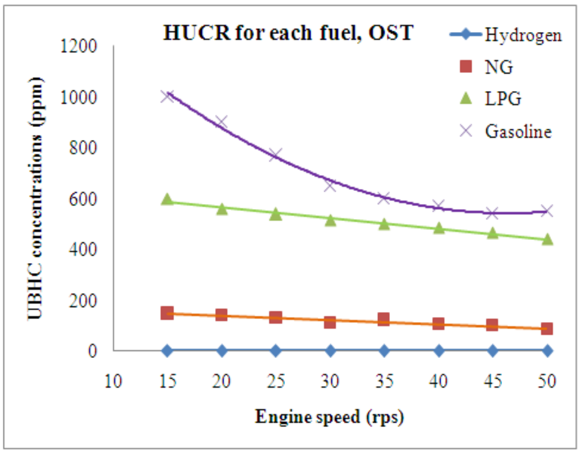
Figure(14): Engine speed effect on OST of lean misfire limit equivalence ratios



Figure(17): Engine speed effect on CO_2 concentrations for lean misfire limit equivalence ratios for the four fuels



Figure(15): Engine speed effect on NOx concentrations for lean misfire limit equivalence ratios for the four fuels



Figure(18): Engine speed effect on UBHC concentrations for lean misfire limit equivalence ratios for the four fuels