DOI:10.21276/sjeat.2016.1.3.7

Saudi Journal of Engineering and Technology
Scholars Middle East Publishers
Dubai, United Arab Emirates
Website: http://scholarsmepub.com/

ISSN 2415-6272 (Print)
ISSN 2415-6264 (Online)

Research Article

Spark Ignition Engine Performance When Fueled with NG, LPG and Gasolin

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Abstract: Natural gas and liquefied petroleum gas are the most important alternative fuels for gasoline in spark ignition engines, for many reasons, such as large world reserve of these gasses, high heating value, high octane number, low emissions emitted from burning them in engines, and their small prices compared with gasoline. The practical study conducted with these two fuels to operate single cylinder with variable compression ratio, speed and spark timing Ricardo E6/US, and its performance was compared with that resulted from running the engine with gasoline. The results appeared that the HUCR (higher useful compression ratio) for gasoline was 8:1, 10.5:1 for LPG and 13:1 for natural gas. Results appeared that spark timing was advanced when using NG more than other used fuels, because of its low flame speed propagation. The study conducted that brake power of LPG and NG were less than that for gasoline at CR=8:1, but they became closer when the engine operated at HUCR for each fuel. The results showed that specific fuel consumption for NG less than that for LPG, which was less than that for gasoline on a mass basis. Also, the exhaust gas temperature for NG was found to be less than LPG, and it was for LPG less than that for gasoline.
Keywords: LPG, NG, gasoline, SIE, performance, compression ratio, equivalence ratio.

INTRODUCTION

Internal combustion engines differ from other types of engines in the manner of operation, since the thermal energy liberated in these engines by igniting the fuel in the engine cylinder. Then, fast chemical reactions conduct, however, the time it takes to create the appropriate mixture of fuel and air operation depends mainly on the nature of the fuel used and its entering method into the combustion chamber. For this reason, there are conditions and requirements for fuel used in internal combustion engines. The used fuel has to have certain physical and chemical properties, such as high combustion energy, high thermal stability, the tendency to form little sediment, suitability of the engine. The fuel must be safe against fire, little toxicity, low pollution, and easy to transport in portable tanks with the engine [1].

Gasoline is the largest industrial product from the raw oil and is used as a fuel for most of the spark ignition engines. Gasoline is a mixture of different types of hydrocarbons, such as paraffin, olefins, naphthenes, aromatics, and benzene. The gasoline components ratios vary depending on the crude oil source and the refinery operations [2].

The liquefied petroleum gas (LPG) is a hydrocarbon mixture, also. The essential ingredient of LPG is the commercial propane and butane, with portion determined by the seasons and the field of use. LPG can be liquefied at normal temperatures using a pressure of approximately (8.5 to 9.0) atmosphere. It is characterized by of being free of water and hydrogen sulfide, and it is always under its vapor pressure. This pressure is influenced by surrounding temperatures, as the vapor pressure of propane and butane are not equal at the same temperature, so the mixing ratios of these gasses affect the total liquefied petroleum gas pressure and thermal energy [3].

There is a natural gas under the ground on two terrain bodies, the first where the gas is accompanying crude oil, which is with the oil directly or dissolved in it. It is separated during the early stages of the production. In the second form, the natural gas exists in a layer above the direct oil layer, in this case without the presence of oil, it is known as gas Dry. Natural gas is composed of several components, as methane which is mainly (98% in the dry gas), and other compounds as ethane (C_2H_6), propane (C_3H_8), butane (C_4H_{10}), in addition to carbon dioxide CO_2, and hydrogen H_2, and nitrogen N_2.

There are two main demands for the fuel used in the internal combustion engine, which are:
1.The period it takes for the combustion process inside the cylinder must be short as much as the possible and maximum amount of thermal energy must be released in this time to give a maximum capacity.

2.The fuel has to compose a homogeneous mixture with air, and provides a homogeneous distribution of fuel in the intake manifold, so as to minimize the difference from cycle to cycle.

Ref. [4] confirmed that the use of gaseous fuels leads to a decrease in the resulted brake power of the engine compared to a gasoline engine, due to the decline in volumetric efficiency. The decrement of brake power can be compensated by increasing the engines' compression ratio. Besides, the authors confirmed that the disparity in the brake power per cylinder reduce by using gaseous fuel compared to gasoline. This is due to better air-fuel charge distribution in the gaseous fuel.

Ref. [5] clarified that spark ignition engine could operate at very lean mixture when it is fed with gaseous fuel compared to gasoline operation.

Ref. [6] found that the increase in compression ratio increases engine brake power and thermal efficiency when it is working with propane or methane. However, the engine brake power and efficiency is lower in the case of methane than in propane at specific compression ratio. The reason for this little brake power and efficiency can be referred to the slow flame propagation for methane gas into the combustion chamber compared to the fast spread of the propane flame. Besides, the methane use reduces the volumetric efficiency of the engine greater than propane.

Ref. [7] indicated that the resulted engine brake power reduces when the engine is fueled with natural gas about 15% compared to gasoline. The study revealed that the optimum spark timing for the best torque increases with the decrease of the equivalent ratio, due to the flame speed decrease in the lean mixture for both gasses. The natural gas' optimum spark timing for the best torque was larger by about 6 degrees from those of gasoline, because of the slow flame speed of methane compared to gasoline.

Ref. [8] showed that the extra engine power drop of about 17% when the engine is fueled with natural gas compared to gasoline, due to the decrease in the engine volumetric efficiency. Also, the author mentioned that when the engine is fueled with NG, the sparks timing must be advanced about 10 degrees due to the NG slow flame spread. In the same time, the ignition limits increase with increasing engine compression ratio.

Ref. [9] noted that the engine operation with liquefied natural gas (LNG) reduces the resulted brake power up to 7% compared to the gasoline engine and increases the fuel consumption up to 22%.

Ref. [10] manifested that there is a decrease in the engine brake power when using methane compared to gasoline, due to the low volumetric efficiency. As the gas displaces a portion of the air entering the cylinder and the fuel does not cool the entering air as in the case when using gasoline. For this reason, the volumetric efficiency decreases in the gasses engine. The study concluded that the brake power reduction referred to the slow flame spread of methane, which requires advancing the optimal spark timing.

Ref. [11] revealed that the difference in the indicated thermal efficiency between propane and methane depends on the spark timing and the equivalence ratio. The thermal efficiency of propane is larger than that for methane at constant spark timing, and this difference approached zero when using the optimum spark timing. The authors stated that the high knock resistance of methane is favorable in the engines that use supercharging. Also, there is a reduction in fuel consumption when NG is used up to 8% when the used engine compression ratio was increased from CR=7.7: 1 up to 9.5: 1.

The high growth of cars fueled with gasoline in the streets caused high air pollution and still to this day many people have doubts about using gasses fuel in their cars and whether they can operate the engine as the gasoline fuel. For this purpose, this work aims to use two Iraq gasses products in a laboratory engine to specify these gasses combustion properties and compared it to gasoline. This work is a part of continuous efforts of the Energy and Renewable Energies Technology Center, University of Technology, Iraq to educate the Iraq society about the benefits of alternative fuels on the country economy and environment [12-59].

EXPERIMENTAL SETUP

Internal combustion engine and its accessories

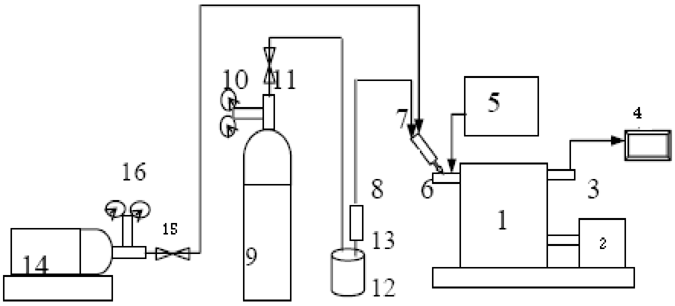
The experiments were conducted in this research using Ricardo E6 engine. This engine is a single-cylinder, four strokes engine, with variable compression ratio, spark timing, equivalence ratio, and speed. The engine is connected an electrical dynamometer. The engine lubricated employing a gear pump operates separately from it. The cooling water is circulated using a centrifugal pump. Fig. 1 represents a scheme diagram of the tests rig while Table 2 illustrates the engine specifications.

Table 1: Ricardo E6 engine geometry and operating parameters

Value	Description
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

The gasoline supply system consists of a major six-liter tank capacity, an auxiliary tank with a capacity of 1 liter, and the gasoline injection system. The LPG supply system consisted of a fuel reservoir, the fuel filter, an electromagnetic valve, LPG carburetor, gaseous fuel flow rate gauge, and damping box. The NG supply system consisted of A methane gas cylinder, a pressure regulator, chock nozzles system to measure NG flow rates, and the NG carburetor. The amount of entering air was measured using an Alock viscous flow meter connected to a flame trap. Engine speed was measured using a calibrated tachometer.

The dynamometer, in addition to its primary function, which is measuring the resulted engine brake power, it was used as an electric engine to rotate the engine at the start of the operation. The dynamometer is used to measure the brake power, the average effective pressure, and power lost due to friction. The exhaust gases temperatures were measured using thermocouple type K (nickel chrome / nickel Alomel). The used thermocouples were calibrated. The following equations were used in calculating engine performance parameters [60]:



1. Single cylinder engine
2. Dynamometer
3. Engine exhausts manifold
4. Exhaust gas analyzer
5. Air drum
6. Engine intake manifold
7. Gas carburetor
8. Solenoid valve
9. NG cylinder
10. Pressure gauge and pressure regulator
11. Non return valve
12. Flame trap
13. Choked nozzles system
14. LPG cylinder
15. LPG flow meter
16. Pressure gauge and pressure regulator

Fig-1: a schematic diagram of the used rig in the experiments

1- Brake power

$$bp = \frac{2\pi * N * T}{60 * 1000} \quad kW \qquad (3)$$

2- Brake mean effective pressure

$$bmep = bp \times \frac{2*60}{V_{sn}*N} \quad kN/m^2 \tag{4}$$

3- Fuel mass flow rate

$$\dot{m}_f = \frac{v_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time} \quad kg/_{sec} \tag{5}$$

4- Air mass flow rate

$$\dot{m}_{a,act.} = \frac{12\sqrt{h_o * 0.85}}{3600} \times \rho_{air} \quad \frac{kg}{sec} \tag{6}$$

$$\dot{m}_{a_{theo.}} = V_{s.n} \times \frac{N}{60*2} \times \rho_{air} \quad \frac{kg}{sec} \tag{7}$$

5- Brake specific fuel consumption

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \quad \frac{kg}{kW.hr} \tag{8}$$

6- Total fuel heat

$$Q_t = \dot{m}_f \times LCV \quad kW \tag{9}$$

7- Brake thermal efficiency

$$\eta_{bth.} = \frac{bp}{Q_t} \times 100 \quad \% \tag{10}$$

MATERIALS

The used liquefied petroleum gas in this research was produced by The Gas Company of Al-Taji, Iraq. This gas consisted of 0.8% ethane, 18.37% iso-butane, propane, 48.7%, and 32.45% N. Butane. The NG used in this research was produced by The North Gas Company, Iraq. The used NG consisted of: 84.23% methane, ethane 13.21%, 2.15% propane, isobutane 0.15%, 0.17% n. Butane, 0.03 Bentane. The Iraqi conventional gasoline which was produced from Al-Doura Refinery was used in this study. The Iraqi gasoline characterized by medium octane number (85 in the recent study), high lead and sulfur content (about 500 ppm) [30]. The fuels specifications were checked in their production sources. Table 2 lists the used fuels specifications.

Table 2: Fuel properties at 25°C and 1 atm

Property	Gasoline	LPG	NG
Density (kg/m^3)	730	1.85/505	0.72
Flammability limits (volume % in air)	1.4-7.6	4.1-74.5	4.3-15
Flammability limits (Ø)	0.7-4	0.7-1.7	0.4-1.6
Autoignition temperature in air (K)	550	588	723
Minimum ignition energy (mJ)	0.24		0.28
Flame velocity (m/s)	0.37-0.43	0.48	0.38
Adiabatic flame temperature (K)	2580	2263	2215
Quenching distance (mm)	2		2.1
Stoichiometric fuel/air mass ratio	0.068	0.064	0.069
Stoichiometric volume fraction (%)	2		9,48
Lower heating value (MJ/kg)	44.79	42.79	45.8
Heat of combustion (MJ/kg air)	2.83	3	2.9

Test procedure

The first tests were conducted to determine the higher useful compression ratio for a wide range of equivalence ratios at engine constant engine speed (1500 rpm) and optimum spark timing. The engine performance was studied in detail at this compression ratio. The engine performance was compared when the engine was operated with each type of fuel alone.

RESULTS AND DISCUSSIONS

Figures 2, 3 and 4 represented the resulting engine bp when it was fueled by gasoline, liquefied petroleum gas, and natural gas, respectively. The tests were conducted for a broad range of equivalence ratios and variable compression ratios at 1500 rpm engine speed and optimum spark timing. The engine bp increased with the equivalent ratio transferred from the lean to the rich side. Higher engine bp was achieved at equivalence ratios between (Ø = 1.1-1.2), as it generated the highest thermal energy from the air-fuel mixture combustion. Most of the oxygen existed in the mixture was used in the combustion process. The bp reduced by increasing the equivalence ratio more (make it richer).

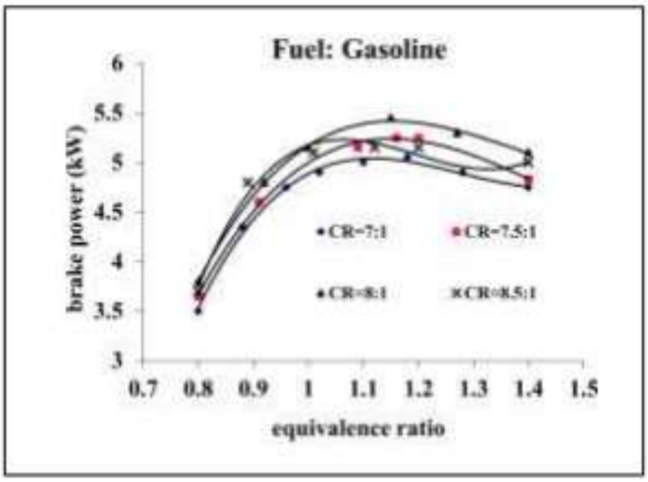


Fig-2: The effect of variable compression ratios on the bp for wide range of equivalence ratios for gasoline

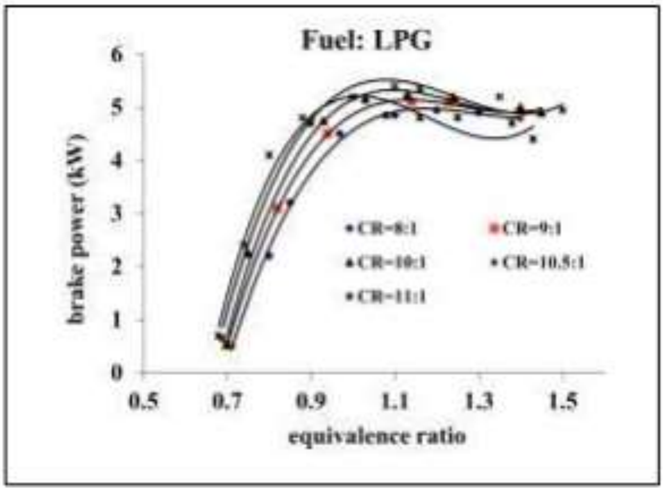


Fig-3: the effect of variable compression ratios on the bp for wide range of equivalence ratios for LPG

The amount of oxygen on the lean side is greater than the amount required for combustion, so a part of the combustion energy is gained by the amount of air excess which will be lost from the engine with exhaust gasses. Besides, the amount of fuel is little. In the rich side, the amount of extra fuel hampered the combustion process, and cause degradation in the bp due to the reduction in the oxygen amount needed for combustion. The figures indicate the possibility of working at equivalence ratios leaner when the compression ratio was increased.

The three figures show a constant trend of the engine performance with the hydrocarbon fuels, as the bp was increased with the compression ratio increase to a certain limit, and then it declined with continuing increasing the compression ratio, due to the knock phenomenon existence, which forced to delay the spark timing, causing bp reduction.

The equivalent ratio at which the maximum bp obtained by changing the compression ratio from the lowest ratio to the higher useful compression ratio for the tested fueled was reduced and approaching the stoichiometric equivalence ratio. For LPG it was dropped from 1.19 to 1.1; for methane from 1.21 to 1.13. The figures show the possibility of operating the engine with wider broader with liquefied petroleum gas compared to gasoline. These borders expanded more when running with natural gas when the engine ran with the HUCR of each fuel.

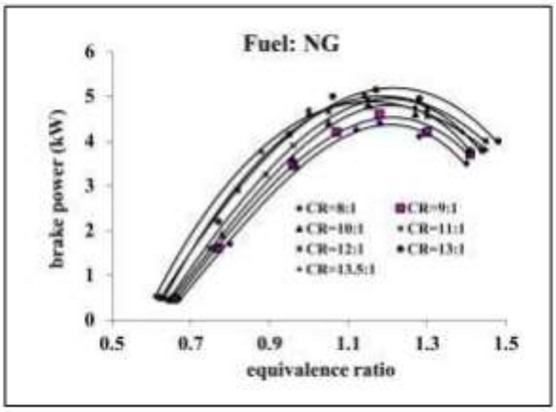


Fig-4: The effect of variable compression ratios on the bp for wide range of equivalence ratios for NG

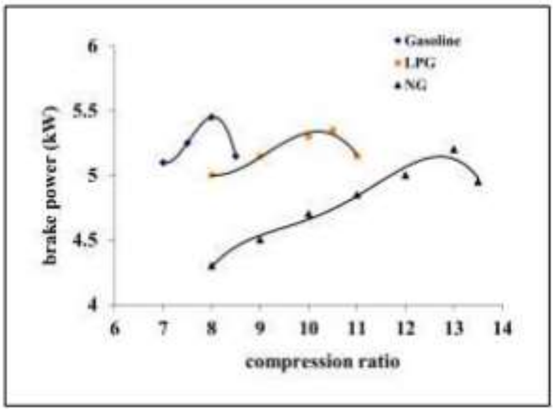


Fig-5: The effect of variable compression ratios on the maximum bp for the tested fuels

The figures clarify that the HUCR for gasoline was 8: 1, and for the gas liquefied petroleum was 10.5: 1, and for natural gas was 13: 1. Fig. 5 shows the relationship between the engine's bp at each engine compression ratio. It is noted that the gasoline's bp is higher than those of the two alternatives. However, these values converge at engine operating at HUCR, and this is confirmed by Figures 6 and 7. Fig. 6 shows the relationship between bp and equivalence ratio for the three fuels when the engine compression is increased from 8: 1, which is the HUCR for gasoline. It appears from the figure that the resulted bp when using liquefied petroleum gas is less than those from gasoline. NG had lower values, because of their small calorific value by volume, low flame speed, and the low volumetric efficiency compared to the other fuels.

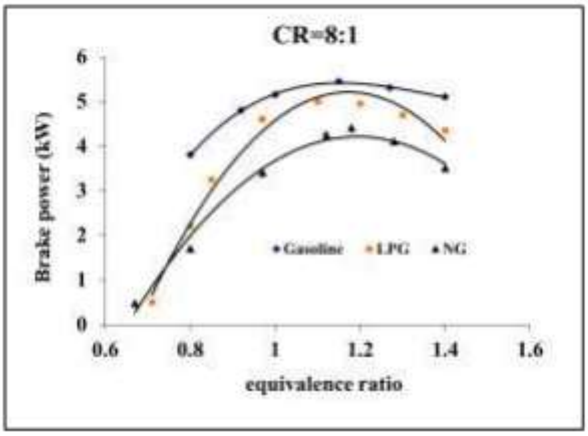


Fig-6: The effect of variable equivalence ratios on the bp for the tested fuel at CR=8:1

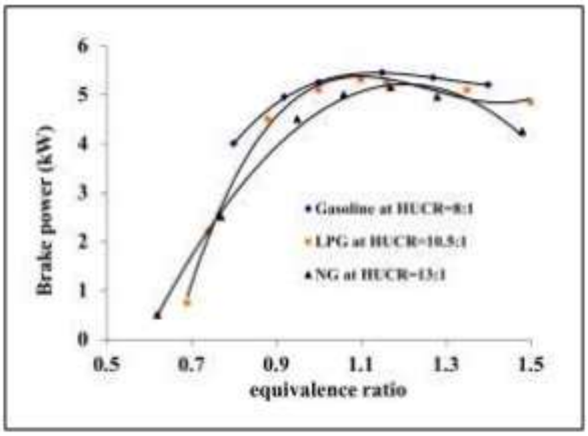


Fig-7: The effect of variable equivalence ratios on the bp for the tested fuels at HUCR for each

Fig. 7 indicates the relationship between bp and the equivalence ratio when the engine operated at the HUCR for each fuel, optimum spark, and constant engine speed (1500 rpm). The resulted brake power for the tested fuels converged because the bp for the LPG was less than that for gasoline about 3% while the NG's bp reduced about 9%.

Fig. 8 shows the relationship between the optimum spark timing and the equivalence ratio at HUCR for each fuel and engine speed of 1500 rpm. The figure indicates a certain spark timing advance when using natural gas, pointing to slow flame spread compared to other fuels, which their optimum spark timings converged due to the convergent flame speed for the two fuels. The gasoline OST delayed from the LPG ones few degrees, due to the improvement in the volumetric efficiency compared to LPG. The figure shows that the spark timing must be advanced by a larger percentage when working with NG compared to its operation with the other fuels. In the same time, the figure indicates that increasing the compression ratio caused retardation to the optimum spark timing; due to the temperature rise inside the combustion chamber which improves the quality of the combustion.

The brake specific fuel consumption was large in lean equivalence ratios; then the consumption reduced to reach its minimum value at an equivalence ratio of about Ø = 0.9. Then, the brake specific fuel consumption increased again with fuel enriched the air-fuel mixture, as Fig. 9 represents. At Ø = 0.9, the best mixing and homogenization that resulted in better and full fuel combustion. As for the rest of the equivalent ratios, the lack of homogeneity of the mixture due to increased oxygen in the combustion chamber in a very lean side, or increasing the fuel in the rich mixture, causing increased specific fuel consumption. The figure clarifies that the specific fuel consumption of natural gas is less than the bsfc of the LPG, which in turn lower than gasoline.

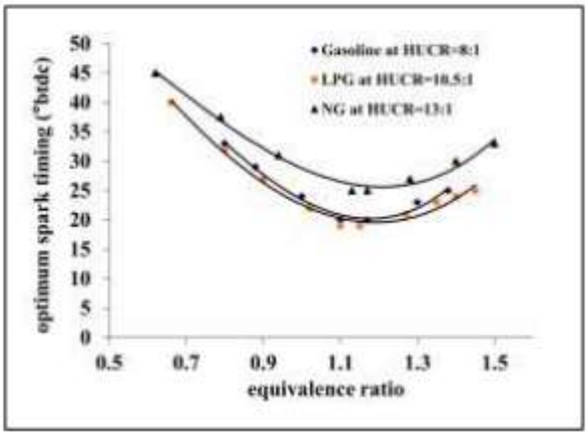


Fig-8: The effect of variable equivalence ratios on the optimum spark timing (OST) of the tested fuels at HUCR for each

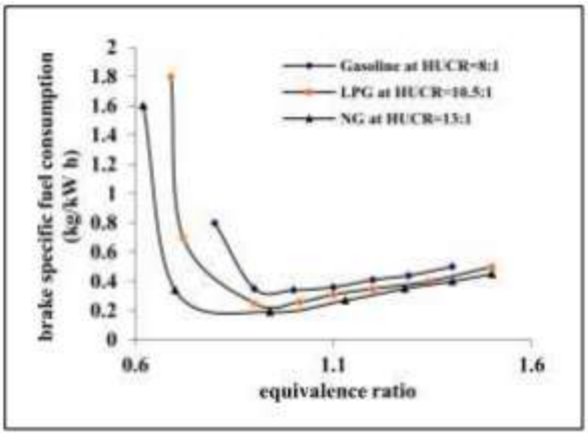


Fig-9: the effect of variable equivalence ratios on the bsfc of the tested fuels at HUCR for each

The exhaust gas temperatures increased with the increase in the equivalence ratio on the lean side at a significant rate until it reached the equivalence ratio at which it achieved the maximum bp, as Fig. 10 reveals. These temperatures reduced by mixture enrichment with a lower rate. The gasoline resulting exhaust gas temperature is higher than the two alternatives temperatures. This is because the maximum combustion temperature is the largest for gasoline compared to those for NG and LPG for all equivalent ratios. The exhaust gas temperatures from the LPG engine were higher than those produced using natural gas, indicating clearly the LPG's high heating value and maximum combustion temperatures compared to natural gas.

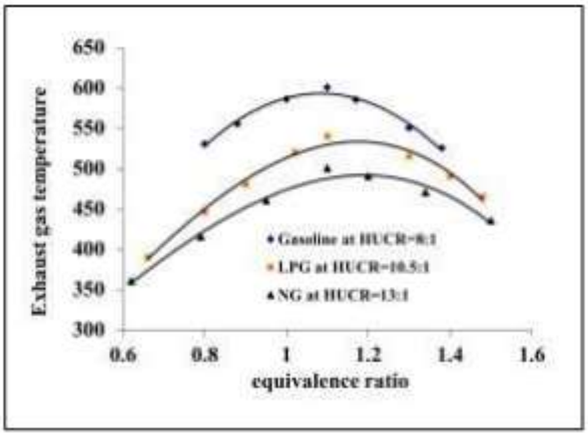


Fig-10: The effect of variable equivalence ratios on the exhaust gas temperatures of the tested fuels at HUCR for each

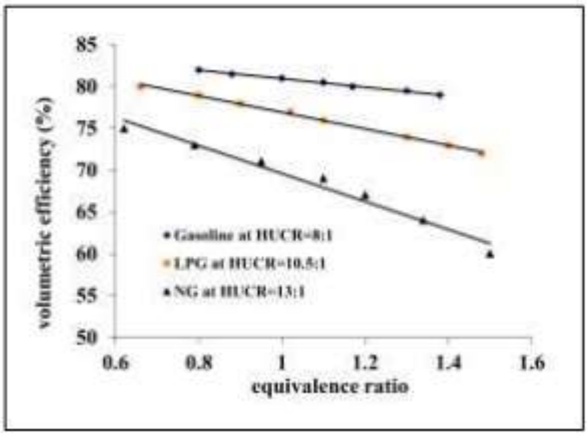


Fig-11: The effect of variable equivalence ratios on the volumetric efficiency of the tested fuels at HUCR for each

Fig. 11 shows the relationship between the equivalence ratio and the volumetric efficiency when the engine is at HUCR for each fuel, OST, and engine speed at 1500 rpm. The volumetric efficiency of natural gas is less than LPG, and in turn, the LPG engine volumetric efficiency is less than that resulted from the gasoline engine. This efficiency reduction results from the gaseous nature of the NG and LPG, which took a greater place on the account of the entering air inside. Besides, the gaseous fuels mix with air without cooling it as in the case of gasoline. The three fuels volumetric efficiencies reduced by enriching the engine with fuel.

Mechanical efficiency is a function of the resulted engine bp, as Fig. 12 manifests. The friction power fixed with the fixing of the engine speed and compression ratio. The mechanical efficiency maximum value was at the equivalent ratio that achieved the maximum bp. The mechanical efficiency reduced at the rest equivalence ratios, and be minimal value was at too lean ratios. From the figure, the gasoline mechanical efficiency was greater than those of NG and LPG. The LPG mechanical efficiency was higher than NG ones due to the gasoline higher bp for all tested compression ratios and equivalence ratios compared to the alternatives.

The indicated thermal efficiency increased from the lean side to reach the maximum value at an equivalence ratio of about Ø = 0.9, and it was decreased after this ratio, as Fig. 13 represents. The equivalent ratio that gave the maximum indicated thermal efficiencies are the same proportions that gave the lowest fuel consumption. It appears from the figure that the thermal efficiency of the NG was higher than those produced using gasoline or LPG due to the reduction of the NG specific fuel consumption with increased bp.

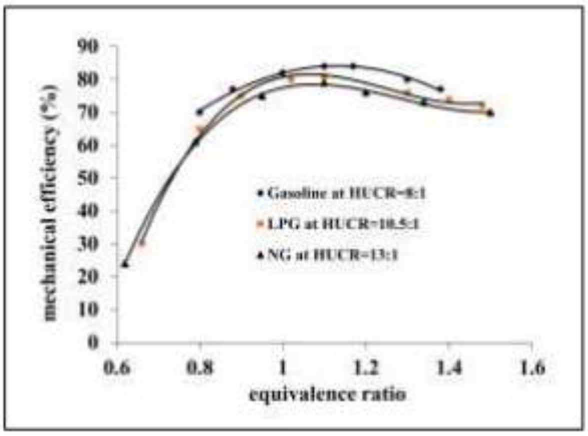


Fig-12: The effect of variable equivalence ratios on the mechanical efficiency of the tested fuels at HUCR for each

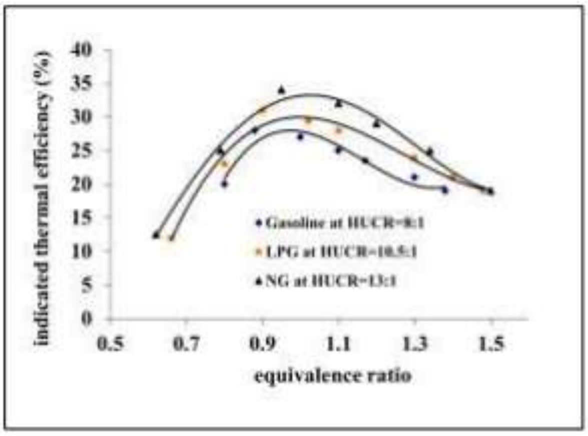


Fig-13: The effect of variable equivalence ratios on the volumetric efficiency of the tested fuels at HUCR for each

CONCLUSIONS

This study aimed to examine the possibilities of using two types of gaseous fuels in a laboratory engine to find the performance characteristics of these fuels in SIE. The experimental tests were conducted using Iraqi gasoline, LPG and NG. The study results reveal that the HUCR for the tested gasoline was 8: 1, for LPG, was 10.5: 1, and for NG was 13: 1. The gasoline bp was higher than that of NG and LPG when the engine was run at a compression ratio of 8: 1. The bp for the three fuels was converged when the engine was run at the HUCR for each fuel. Therefore, the engine preparation to operate with any of the tested gasses requires some design changes as increasing the engine compression ratio to get better bp and performance which is asymptotic to the gasoline engine performance. The gas engine can work at leaner equivalence ratios with the compression ratio increase. Also, the equivalent ratios range became wider with natural gas compared to the others. The engine operation at lean equivalence ratios increased the indicated thermal efficiency and produced fewer exhaust pollutants. The OST of the NG engine must be advanced significantly compared with the other fuels operation, due to it's the small flame speed. This NG's property must be taken into consideration when designing engines run on natural gas. The NG engine's bsfc was less than that for LPG which in turn was less than that caused by gasoline.

Nomenclature

BTDC :Before top dead center
ºCA :Crank angle degrees
CR :Compression ratio
OIT:Optimum injection timing
Bmep:Brake mean effective pressure
Bp: Brake power
Bsfc: Brake specific fuel consumption
HUCR:Higher useful compression ratio
Ø: Equivalence ratio

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