Evaluation of the effect of cooled EGR on knock of SI engine fueled with alternative gaseous fuels

Miqdam Tariq Chaichan-Lecturer
Machines & Equipment Eng. Dept., UOT, Baghdad.

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Abstract

The warming of the climate system is unequivocal, as it is now evident from many observations. Most of the climatic warming caused by human activities; include in particular the burning of fossil fuels that generate the 'greenhouse' gases. The transport sector is one of the most important causes of the high levels of air pollution. 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. Exhaust gas recirculation (EGR) is one of the principal techniques used to control spark ignition NOx. However, EGR has different effect on performance, combustion, and emissions production that are difficult to distinguish.

The knock characteristics in a single cylinder SI engine were investigated under high load operation and higher useful compression ratio with cooled EGR addition. Liquefied petroleum gases (LPG), natural gas (NG) and hydrogen gas were used as fuels. In this paper, practical analysis has been done to predict the influence of cooled EGR on engine knock by changing some operational and design parameters such as equivalence ratio, spark timing and compression ratio.

Keywords: *spark ignition engine, cooled EGR, knock, gasoline, LPG, NG, hydrogen, compression ratio, equivalence ratio, spark timing.*

1.Introduction

To meet the customer's requirement for a high-performance vehicle and the social needs (such as stringent exhaust gas legislation, improving fuel economy), increasing the compression ratio in a spark ignition engine has long been considered a reasonable method to achieve them. Although increasing the compression ratio of an engine is effective in improving its thermal efficiency, power output, and fuel consumption. The highest compression ratio is limited because of a knock phenomenon (Szwaja, 2007 & Hani et al., 2000).

To meet the twin problems of fuel scarcity and air pollution caused by growing use of petroleum fuels. Alternative renewable clean burning fuels as LPG, NG and hydrogen have been explored, as a most prominent eco-friendly fuel for use in SI engine vehicles. They are a promising alternative fuels to meet strict engine regulation in many countries (Chaichan, 2010 & Amrouche, 2010).

LPG and NG powered engine offer certain advantages e.g. clean combustion (no lead or sulphur compound) due to high H/C ratio, does not contaminate and dilute the engine oil which forms no deposits on spark plug (Soylu, 2005). It also offer advantage over gasoline in terms of its low density and ready miscibility with air, high knock resistance (octane number 110 for LPG and 130 to 140 for NG), lean burning capability. However, due to lack of modification in available engines including supply of conversion kits, it is difficult to fully utilize the above properties of LPG and NG powered engine to enhance the performance (Chaichan, 2008 & Tirkey, 2010).

The high autoignition temperature of hydrogen (858 K) means that hydrogen is most suitable as a fuel for spark ignition (SI) engines (Chaichan, 2008 & Killingswortha, 2010). Hydrogen is a very versatile fuel when it comes to load control (Mohammadi, 2007). The high flame speeds of hydrogen mixtures and its wide flammability limits permit very lean operation and substantial dilution (Kawahara, 2009 & Verhelst, 2009). Despite the high autoignition temperature, the energy liberated by hydrogen–air mixtures are approximately an order-of-magnitude lower than that of hydrocarbon–air mixtures. The low ignition energies of hydrogen–air mixtures mean that H_2 engines are predisposed towards the limiting effects of preignition (White, 2006 & Kanti Roy, 2010). Three main factors are limiting the usage of this gas as a fuel in internal combustion engines. These factors are the high prices of hydrogen production in addition to its large volume, and the need for high pressures to store it (Chaichan, 2008).

Knock is due to an unexpected combustion in Spark Ignition (SI) engines. It is a result of spontaneous ignition of a portion of end gas in the engine chamber, ahead of the propagating flame (Griffiths, 2002). Rapid heat release implied

by this abnormal combustion generates shock waves that can lead to the decrease in output, the increase in some pollutants and the destruction of the engine (Abu-Qudais, 1996). Although knock has been more or less overcome in gasoline engines by controlling the fuel quality, gas engines are not safe from knock (Naber et al, 2006). Combustion knock has long been one of the major barriers in the development of spark ignition engine for increased efficiency or improved fuel tolerance (Chun, 1989 & Moses, 1995).

Up to now, Combustion is affected by temperature, pressure, mixture composition and piston geometry. The mixture composition is essentially influenced by EGR and air to fuel ratio (AFR) (Eguz, 2009). One of the most important parameters for combustion in new engines is definitely EGR. EGR has more than just one effect on the engine cycle (Boot, 2009). First of all, EGR plays a crucial role in combustion phasing by changing mixture composition (Brecq, 2002). AFR and the amount of oxygen in the cylinder are reduced by increasing the EGR fraction in the mixture. O_2 % is evaluated via air, fuel and EGR flow rates (Bade-Shrestha, 2001). Second, EGR drops the temperature and pressure increase during engine cycle by increasing the constant specific heats (Landry, 2007). However, if exhaust gas is recirculated to the intake without cooling, the intake temperature is increased which is another crucial effect of EGR (Boot, 2009).

In this study, cooled EGR is used. EGR ratio was fixed at 15% during the experiments. The effect of EGR on engine knock was studied when the engine was fueled with LPG, NG and gydrogen. The results were compared with gasoline engine used the same EGR percentage at the same operational conditions.

2.Experimental setup

2.1.Materials

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. 84. The LPG fuel produced from Al-Taji Gas Company; consist of ethane 0.8%, 18.47 isobutane, 49.8% propane and 30.45% butane. NG used was produced from Iraqi Northern Gas Company; consist of 86.23% methane, 11.21% 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. These constitutions were taken from the materials sources.

The very wide diversity in the composition of the gaseous fuels commonly available and their equally wide variety of their associated physical, chemical and combustion characteristics make the prediction and optimization of their combustion behavior in engines a more formidable task compared to conventional liquid fuels. Continued research is needed to provide more light on their suitability as engine fuels and understand better the roles of the many factors that control their behavior.

2.2.Internal combustion engine and its accessories

The engine used in these investigations was 4 stroke single cylinder, with variable compression ratio, spark timing, a/f ratio and speed Ricardo E6. The engine specifications are listed in **Table 1**. The engine connected to electrical dynamometer, and lubricated by gear pump operated separately from it. The cooling water circulated by centrifugal pump, **Fig.1** represents the engine used in this study and its accessories.

Gasoline supply system: This system consists of major tank (6 liter capacity), minor tank (1liter capacity), and gasoline carburetor.

LPG supply systems: This system consist of LPG tank, fuel drier, solenoid valve, LPG carburetor, gaseous fuel flow measuring device (orifice plate), damping box.

Hydrogen and NG supply system: Hydrogen and natural gas 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.

Air flow measurement: Air interring the engine was measured by Alock viscous flow meter connected to flame trap.

Speed measurement: Engine speed was measured by tachometer.

Power measurement: The electric dynamometer was used to measures indicated power, brake mean effective pressure and friction lost power. In addition of it was used to measure power; it is used as electric motor also, to rotate the engine in the starting.

Exhaust gas temperatures measurement: Exhaust gas temperatures were measured by nickel chrome/ nickel aliumel thermocouple, which was calibrated before it was used.

The in-cylinder pressure was measured using a piezoelectric pressure transducer (Kistler, 6052b). The piezo electric pickup fitted into the combustion chamber, along with a charge amplifier, oscilloscope and Iwatsu Signal Analyzer enable the measurement of cylinder pressure.

EGR system: In order to furnish the tested engine with EGR, a supply system was fitted to the engine. The exhaust gas was extracted immediately above an intermediate flange connecting between the exhaust gas manifold and exhaust pipe, which is 35cm downstream from confluence point. By this arrangement, the driving force for the EGR was the pressure difference between the exhaust and the intake manifold pressure. EGR cooler was used to generate cooled exhaust gas.

In order to assure sufficient cooling capacity of this cooler, which used the cooling water taken from the municipal water supply, cooled EGR line used a 2m copper tube. EGR cooler was used as a heat exchanger. The cooler is a two-stream cross-counter flow type heat exchanger of a tubular shape, by which the recirculated exhaust gas flows through the outer tube. The entire EGR cooler was constructed of steel. Fig. 2 represents schematic diagram for the tests system

3.Analysis

The following equations were used in calculating engine performance parameters (Keeting, 2007):
The equivalence ratio which was determined from the measured air and fuel flow rates to the engine, defied as:

$$\emptyset = \frac{\text{actual fuel/air ratio}}{\text{stoichiometric fuel/air ratio}} \quad (1)$$

Brake power

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

Fuel mass flow rate

$$\dot{m}_f = \frac{v_f \times 10^{-6}}{1000} \times \frac{\rho_f}{time} \quad kg/_{sec} \quad (3)$$

Air mass flow rate

$$\dot{m}_{a,act.} = \frac{12\sqrt{h_o * 0.85}}{3600} \times \rho_{air} \quad \frac{kg}{sec} \quad (4)$$

$$\dot{m}_{a_{theo.}} = V_{s.n} \times \frac{N}{60*2} \times \rho_{air} \quad \frac{kg}{sec} \quad (5)$$

Brake specific fuel consumpotion

$$bsfc = \frac{\dot{m}_f}{bp} \times 3600 \quad \frac{kg}{kW.hr} \quad (6)$$

Total fuel heat

$$Q_t = \dot{m}_f \times LCV \quad kW \quad (7)$$

Brake thermal efficiency

$$\eta_{bth.} = \frac{bp}{Q_t} \times 100 \quad \% \quad (8)$$

EGR rate is calculated as follows (Hountalas, 2008):

$$EGR\,(\%) = \frac{m_{\dot{E}GR}}{\dot{m}_{EGR} + \dot{m}_a} \times 100 \quad (9)$$

Where merge is the mass flow rate of EGR and meager is the mass flow rate of fresh air. In order to determine how far the EGR valve should be opened to achieve a desirable EGR mass ratio, different EGR rates were extracted from a simple computer code based on the equation of gas state and the method of trial and error.

4.Test program

The first tests were conducted to determine the higher useful compression ratio (HUCR) for each fuel. Engine performance was tested for wide range of equivalence ratios, optimum spark timing and 1500 rpm engine speed. The same tests were repeated again but with adding cooled EGR.

The single cylinder was used while operating unthrottled under atmospheric pressure conditions. The engine compression ratio was varied (from 7:1 to 17:1), while spark timing was varied from 5 to 50°BTDC. The induction system permits the simultaneous admission of fuels that were supplied separately from a set of high-pressure cylinders and metered individually using a series of calibrated choked nozzles or orifice plate.

To determine the knock limited equivalence ratio, the fuel-air mixture composition was fixed and the engine load was varied gradually from no load until the onset of knock was first encountered, while all other operating parameters kept constant.

To determine the knock limited compression ratio, it was varied gradually until the onset of knock was first encountered, while other parameters were kept constant. The spark timing advance was reduced incrementally with raising compression ratio for operations of the gaseous fuels and follows a similar approach of (Karim and Klat, 1966).

The onset of knock was established by a combination of the appearance of the characteristic oscillations on the cylinder pressure-time trace together with the detection of the specific level of the occasional audible noise related to knocking. The “knock limit” in this contribution corresponds to borderline knock and it is defined in terms of the equivalence ratio, (i.e. fuel-air ratio relative to the stoichiometric value), at a specific compression ratio and a set of operating conditions. All tests were conducted without throttling the engine at constant intake pressure of 0.98 bar. The engine was pre-heated by burning gasoline fuel, bringing the coolant temperature to 75°C.

5.Results and Discussions

The knock free operating mixture range is bounded by the lean and rich operational ignition limits as shown for gasoline (**Fig. 3**), LPG (**Fig.4**), NG (**Fig. 5**) and hydrogen (**Fig. 6**) and air. This range is seen to limit compression ratios due to knock occurance, but after cooled EGR addition

this range extended to 9.5:1 for gasoline, 13:1 for LPG, 16:1 for NG and 14:1 for hydrogen before any knocking is encountered first around lean to stoichiometric mixture region. The figures show the effect of compression ratio on engine brake power. The increase in compression ratio shows the increase in engine brake power, which is due to the higher cylinder gas pressure and improvement of combustion with higher peak pressure and temperature and due to more scope of expansion work.

The knock free region at high compression ratios becomes increasingly confined to a rapidly narrowing mixture region that will much reduce the available power output that can be achieved and indicates reduced flexibility in lean mixture control. It can be seen also that for yet higher compression ratios autoignition of the whole homogeneous charge can take place, even in the absence of electric spark with much intense knocking. Increasing compression ratio accompanied with increasing combustion chamber temperatures. These tempresures rose to an extend that they will be able to ignite the fuel-air charge without electric spark.

Methane the main part of NG is well known to be a highly superior fuel in resisting the tendency to knocking than virtually all other common fuels. This feature comes about largely from its relatively slow rates of autoignition reactions, which make it, require high temperatures to initiate autoignition and knock.

It can be seen that hydrogen despite its very high flame propagation rates is the most prone to knocking permitting only very lean mixtures to be used under knock-free conditions. This would seriously reduce the power output of engines operating on hydrogen. It can be seen also that LPG is much more resistant to knocking than gasoline but is significantly less than for NG. Thus, NG remains the superior fuel.

Fig. 7 shows the variation of cooled EGR effect on teseted fuels when they operated at knock limits equivalence ratio. It can be seen that the addition of EGR to fuel increase the behavior of knock resistance. Differences can be noted between the used fuels showing that gasoline has the narrower range of equivalence ratios which has the higher brake power compared to gasous fuels.

With increase in compression ratio, the clearance volume decreases and a greater portion of spent up gases are exhausted. The temperature of the charge is high at higher compression ratio. Since the heat transfer increases with greater charge density and increased temperature, the flame speed increases. The combustion duration decreases with increase of compression ratio, because the maximum cycle gas temperature increases resulting in higher apparent flame speed. These factors greatly affect engine brake specific fuel consumpsion as **Fig. 8** represents. Hydrogen appeared to have the minimum bsfc values, while gasoline has the maximum values.

Fig. 9 shows the effect of knock limit operation and cooled EGR addition on volumetric efficiency for wide range of equivalence ratios. Volumetric efficiency determines the maximum power that can be developed by the four stroke cycle engines due to their distinct induction process. Volumetric efficiencies of gaseous fuels are worse than gasoline. Low density of gaseous fuels caused approximately 4 % power loss for LPG, 7.9% for NG and 12% for hydrogen compared to gasoline powered engine. Introduction of gaseous fuels in the intake manifold decreased the air partial pressure notably compared to gasoline. This reduction in power is inherent the structure of gaseous fuels.

The second reason which causes power loss is related to the intake manifold air density. The heat of vaporization of gasoline helps to decrease the temperature of mixture, producing the dense mixtures. Although LPG and NG have higher heat of vaporization value, they are already in gaseous state when inducted into the intake manifold and they do not provide this cooling effect. Development of liquid fuel injection systems for LPG, NG and hydrogen engines should provide better performance and efficiency. Also, the injection of liquid fuel provides better control of A/F ratio.

Another loss off volumetric efficiency and power is related to adding EGR. Most engines converted to burn with EGR suffer an additional 10-20 % power loss due to obstruction of air flow. This explains the difference between theoretical power loss and actual power loss. The improved volumetric efficiency and higher combustion energy increase the output of the engine. The resistance to pre-ignition and knock increased due to cooling effect of cooled EGR. This leads to higher compression ratios which mean higher power output.

The effect of compression ratio on engine thermal efficiency is very well known. Higher compression ratio improves thermal efficiency and provides more power that can be produced by the engine. Higher octane rating of gaseous fuels compared to gasoline allows higher compression ratio for the engine. If the initial mixture temperature is decreased using a cooled EGR as in these tests, higher compression ratios can be attained, as **Fig.10** illustrates. Autoignition is observed with cooled EGR addition at higher compression ratios compared to the case of engine operation without EGR. Hydrogen recorded the maximum brake thermal efficiency at altra lean equivalence ratios, while gasoline recorded the minimum values.

Further, parameters such as spark timing can be optimized under knock limitation for the best performance, as **Fig. 11** shows. Spark timing shows how much this parameter can be adjusted to reduce the engine autoignition tendency. The end gas temperature is strongly dependent on timing, and therefore, any retardation in timing decreases the end gas temperature. Onset knock limit depends on spark advance, more advance

sparking increases the peak pressure of the cycle and therefore increases the pressure and temperature of the end charge resulting shortens delay period (induction/auto ignition time) and increases the tendency to knock. Knock control systems generally retard the spark timing to prevent heavy knock. Hydrogen is distinguished by its high burning velocity. That's why it has the minimum optimum spark timing compared to other fuels. NG is distinguished by it low burning velocity, that's why it has the maximum optimum spark timing of the tested fuels. In this context, the influence of cooled EGR during the period by the end of compression is considered to be dominant compared with that during combustion because of the short duration of the combustion stroke. We can therefore consider the decrease in gas temperature at the end of the compression stroke as an index which effectively represents the effect of engine cooling optimization. It is well known that rapid auto-ignition occurs at elevated temperatures advancing the combustion, whereas a drop in O_2% in the mixture composition retards the combustion which is one of the main reasons for EGR utilization. The application of cooled EGR affects both temperature and O_2 concentration.

Fig. 12 shows the effect of cooled EGR on exhaust temperature according to knock conditions. The exhaust temperature increases at knock conditions or near it, which may be due to the increase in combustion temperature. As can be seen in the figure, exhaust gas temperature decreased with the EGR rate. Accordingly, a cooled EGR rate resulted in lower exhaust temperature, accompanying the importance of thermal efficiency since EGR reduces the combustion temperature and prevents a late combustion. At an EGR rate of 15%, the exhaust temperature decreased by approximately 50°C at maximum under the engine operation of 1500 rpm and full load.

The more homogeneous mixing between air, fuel, recirculated exhaust gas and residual gas appears to offset the effect of slower burning rate due to the charge dilution of recycled gas by dint of the reduction of EGR temperature. As a result, this behavior might have affected the decrease in exhaust gas temperatures. The exhaust gas temperature decreased according to the decreasing EGR temperature.

6.Conclusions

The major findings of present investigation for the effects of cooled EGR on the knocking combustion process in a spark ignition engine can be summarized as follows:

- It is clearly indicated that the knock intensity can be dramatically reduced by the application of the cooled EGR. The reduction rate in knock intensity is strongly dependent upon the recirculated gas mass fraction at the onset of the end-gas autoignition. The amount of the cooled EGR at the onset of autoignition is found to play an important role in the knocking combustion.
- The onset of knock may be avoided under any set of operating conditions for all the fuels tested by reducing the equivalence ratio for lean mixture operation, lowering the compression ratio and/or retarding the spark timing.
- NG has superior knock resistance qualities to those of gasoline, LPG and hydrogen.
- Hydrogen fueled SI engine is most efficient when running on altra lean (Ø=0.25-0.45) mixture at moderate engine speed (≈ 1500 rpm) and optimum spark timing. NG is most efficient when running at stoichiometric or near it from the lean side (Ø=0.8-1.0).
- Brake power, indicated thermal efficiency and exhaust gas temperature increase with the increase of compression ratio. Brake specific fuel consumption reduces with increase of compression ratio.
- Onset knock limit depends on spark advance. An increase in spark advance from the optimized timing increases the tendency to knock, thus retarding saprk timing has high effect on engine tendency to knock. It is observed that retarding timing helps to reduce knock.
- Adding cooled EGR reduces the cylinder temperature at the end of the compression cycle, this effect is considered to be significant for suppressing knock. The in-cylinder gas mean temperature has perhaps the greatest influence on knock. Since the knock phenomenon is caused by a complex combination of many factors, this method should be used for evaluating the occurrence of knock in engines.

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تقويم تأثير تدوير غاز عادم مبرد على الطرق في محرك اشتعال بالشرارة يعمل بأنواع من الوقود الغازي البديل

د . مقدام طارق جيجان – مدرس
قسم هندسة المكائن والمعدات
الجامعة التكنولوجية ، بغداد

المستخلص

أن ارتفاع حرارة نظام الطقس واضح وجلي اليوم عبر عدة مشاهدات ، فمعظم السخونة الموجودة ناجمة عن فعاليات بشرية ، من ضمنها احراق الوقود الأحفوري المولدة لغازات البيت الزجاجي . وفي المناطق المدنية يمكن اعتبار قطاع النقل أحد أهم مسببات المستويات العالية لتلوث الهواء . يرتبط الوقود الغازي البديل بمجال واسع من أنواع الوقود تكون بالحالة الغازية في ظروف الجو الطبيعية . ويعتبر تدوير الغاز العادم أحد التقنيات الأساسية المستخدمة للتحكم بملوث NOx الناتج من محرك اشتعال بالشرارة ، ولتدوير الغاز العادم تأثيرات مختلفة على الأداء والاحتراق وانتاج الملوثات التي من الصعب تميزها .

تمت دراسة مواصفات الطرق للمحرك لظروف تشغيل بحمل عالي لمحرك اشتعال بالشرارة احادي الاسطوانة عند أعلى نسبة انضغاط مفيدة باضافة تدوير للغاز العادم المبرد . تم استخدام الغاز النفطي المسال والغاز الطبيعي والهيدروجين كوقود . لقد تم في هذة الدراسة تحليل عملي لتأثير تدوير غاز عادم مبرد على طرق محرك مع تغيير بعض العوامل التصميمية والتشغيلية مثل النسبة المكافئة وتوقيت الشرر ونسبة الانضغاط .

Table 1: Ricardo E6 engine geometry and operating parameters (Gould, 1976).

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

Table 2 Fuel properties at 25°C and 1 atm (Chaichan, 2010)

Property	Gasoline	LPG	NG	Hydrogen
Density (kg/m^3)	730	1.85/505	0.72	0.0824
Flammablity limits (volume % in air)	1.4-7.6	4.1-74.5	4.3-15	4-75
Flammablity limits (Ø)	0.7-4	0.7-1.7	0.4-1.6	0.1-7.1
Autoignition temperature in air (K)	550	588	723	858
Minimum ignition energy (mJ)	0.24		0.28	0.02
Flame velocity (m/s)	0.37-0.43	0.48	0.38	1.85
Adiabatic flame temperature (K)	2580	2263	2215	2480
Quinching distance (mm)	2		2.1	0.64
Stoichiometric fuel/air mass ratio	0.068	0.064	0.069	0.029
Stoichiometric volume fraction (%)	2		9,48	29.53
Lower heating value (MJ/kg)	44.79	42.79	45.8	119.7
Heat of combustion (MJ/kg air)	2.83	3	2.9	3.37

Fig. 1, The Recardo engine and its accessories used in this study.

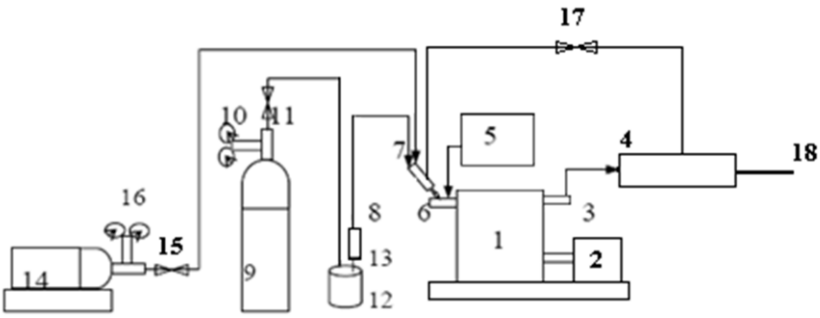

Fig.2: a schematic diagram for the tests system.

1. Single cylinder engine
2. Dynamometer
3. Engine exhausts manifold
4. Exhaust gas cooler
5. Air drum
6. Engine intake manifold
7. Solenoid valve
8. Gas carburetor
9. Hydrogen or NG cylinder
10. Pressure gauge
11. Non return valve
12. Flame trap
13. Choked nozzeles system
14. LPG cylinder
15. LPG flow meter (orifice plate)
16. Pressure gauge and pressure regulator
17. EGR flow meter
18. Exhaust gas

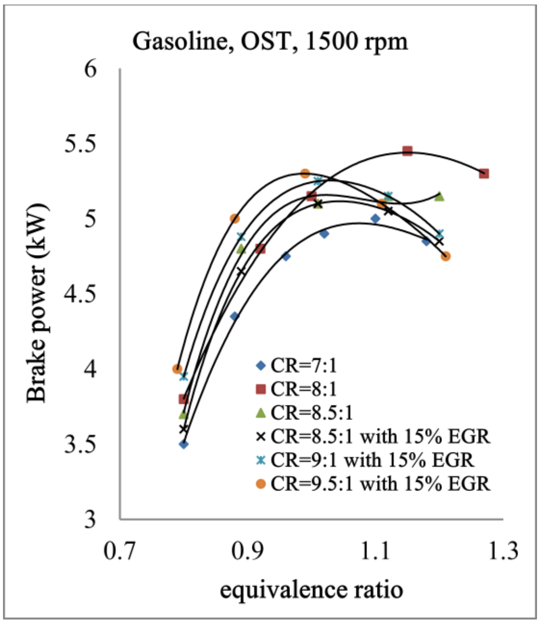

Fig. 3, the effect of compression ratio and EGR on brake power for wide range of equivalence ratios for gasoline fuel

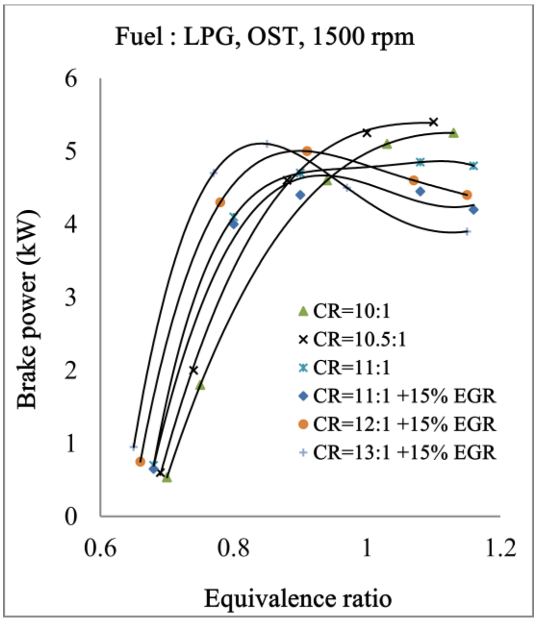

Fig. 4, the effect of compression ratio and EGR on brake power for wide range of equivalence ratios for LPG fuel

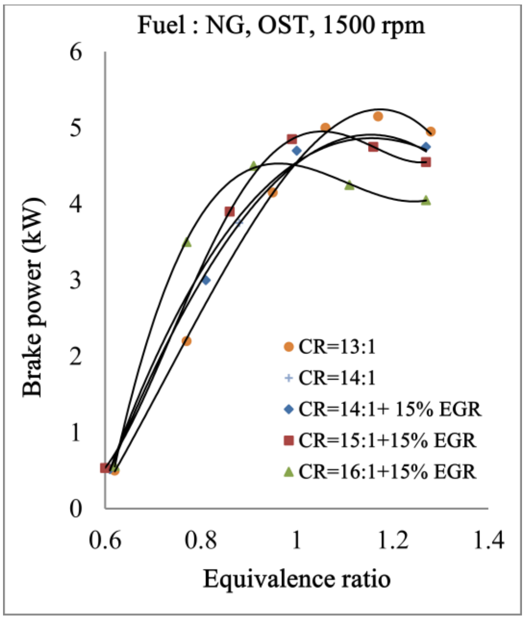

Fig. 5, the effect of equivalence ratio and EGR on brake power for wide range of variable compression ratios for NG fuel

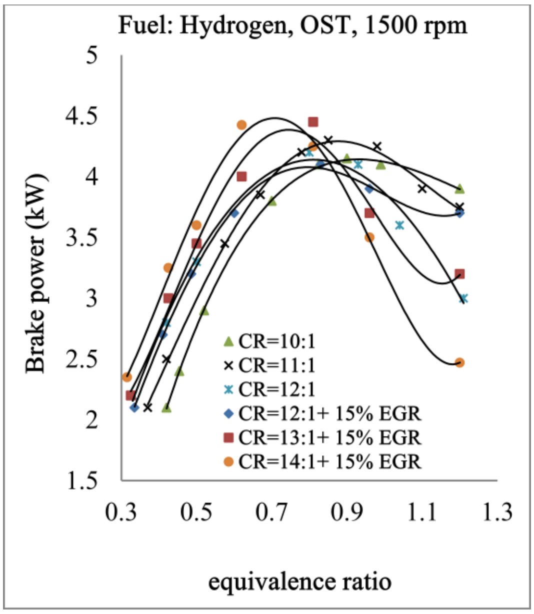

Fig. 6, the effect of wide range of equivalence ratios and EGR on brake power for variable compression ratio for hydrogen fuel

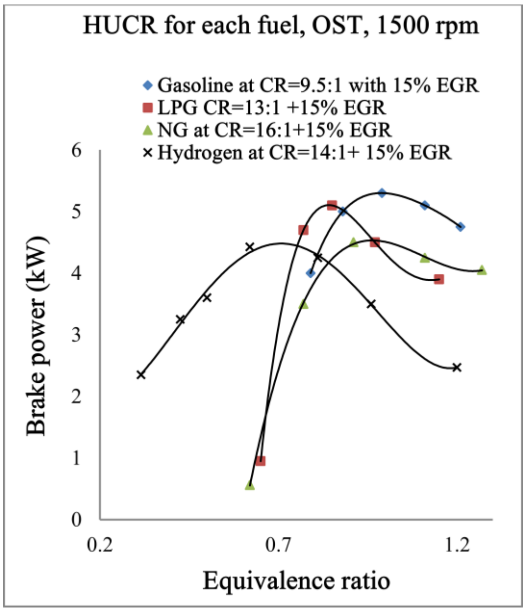

Fig. 7, Compareson between the tested fuels at HUCR of each fuel with EGR addition for wide range of equivalence ratios

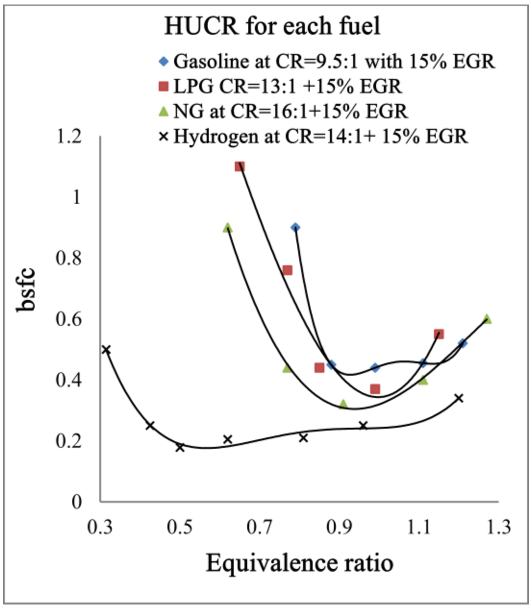

Fig. 8, Compareson of bsfc for the tested fuels at HUCR of each fuel with EGR addition for wide range of equivalence ratios

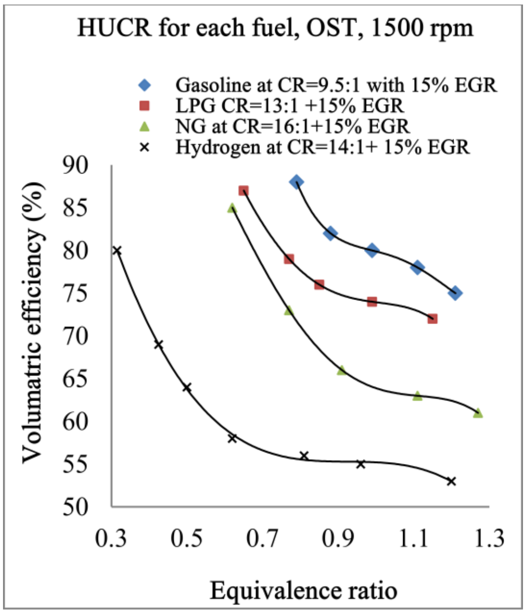

Fig. 9, Compareson of volumetric efficiency for the tested fuels at HUCR of each fuel with EGR addition for wide range of equivalence ratios

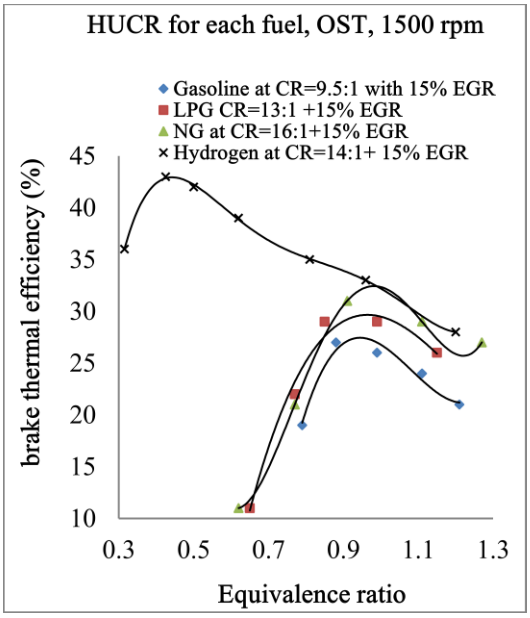

Fig. 10, Compareson of brake thermal efficiency for the tested fuels at HUCR of each fuel with EGR addition for wide range of equivalence ratios

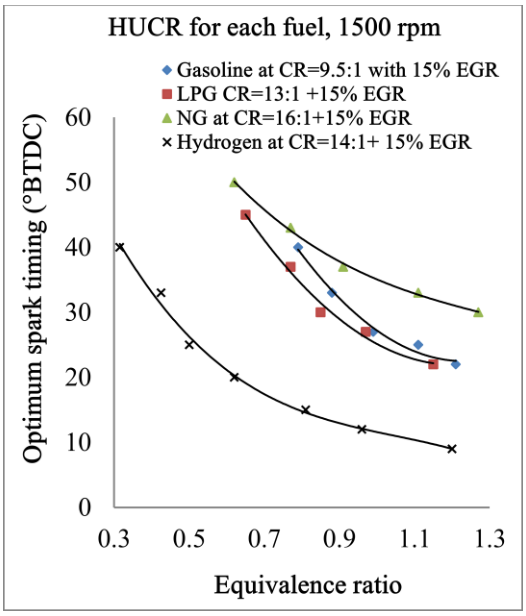

Fig. 11, Compareson of optimum spark timing for the tested fuels at HUCR of each fuel with EGR addition and wide range of equivalence ratios

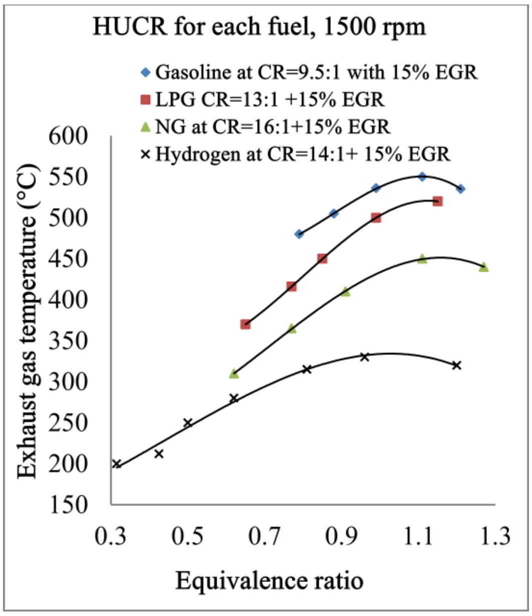

Fig. 12, Compareson of optimum spark timing for the tested fuels at HUCR of each fuel with EGR addition and wide range of equivalence ratios