EGR effects on hydrogen engines performance and emissions

Miqdam Tariq Chaichan

Abstract— Hydrogen as a fuel in the internal combustion engine can be a useful alternative to the use of fossil fuels. The depletion of oil reservoirs and the environment pollution are the major obstacles related to using conventional fuels whether gasoline or diesel. In this article, the hydrogen engine fundamentals are described, and the hydrogen abnormal combustion phenomenon is discussed like back-fire, pre-ignition, and knock, and emissions of NOx.
EGR is widely used to reduce NOx, and pumping loss. It also increases thermal efficiency in automobile engines. However, it does have some disadvantages such as its detrimental effects on combustion stability. In this article two separate EGR (hot and cooled) systems were tested, using Ricardo spark ignition single cylinder engine, constant velocity and HUCR for each case were used to compare between produced engine performance and emitted emissions.

Keywords-Hydrogen, gasoline, equivalence ratio, compression ratio, spark timing, speed, brake power, specific fuel consumption, indicated thermal efficiency, exhaust gas temperature.

1 Introduction

Today, new technologies are introduced every day in transportation's sector which is increasing rapidly. Fossil fuel mainly petroleum fuel is the major contributor to energy production and the primary fuel for transportation. The rapid depleting of oil reserves and the air quality reduction due to pollution raise many Urgent inquiries about the fossil fuels future. The search to find alternatives to conventional fuels becomes a necessity, especially with the increasing awareness about environment protection [1], [2].
The twin crises have very badly hit internal combustion engines which form the backbone of the available transportation and power sector. Under such circumstances, it has become essential that a renewable clean-burning alternative fuel must be expeditiously sought. In this respect, hydrogen since it is an infinite source potential and clean-burning characteristics possess the unique capability of providing the lasting solution to the twin crises [3], [4].
If hydrogen is produced by using regenerative energy (sun, wind, geothermic), it will be the only fuel that can be produced without using fossil energy sources and that also burns without emitting CO_2. The product of complete combustion is hot water vapor in the same quantity of the water required to produce hydrogen so that a closed circuit results. Therefore, many leading companies in the field of the mineral oil industry, car manufacturers and legal authorities assess H_2 as the most promising of all fuels in the long term [5].
The properties of hydrogen are detailed in Table 1. The properties that contribute to its use as a combustible fuel are its:
• A wide range of flammability: The broad flammability range characterizes hydrogen compared to other fuels. As a result, hydrogen can be combusted in an internal combustion engine over a wide range of fuel-air mixtures. Therefore, hydrogen can run on a lean mixture [6].
• Low ignition energy: Hydrogen has very low ignition energy. This property enables hydrogen to ignite at very lean mixtures and ensures prompt ignition. Unfortunately, this low ignition energy causes that hot residual gases and hot spots in the combustion chamber turn to ignition sources. These sources create the premature ignition and flashback phenomenon [7], [8].
• Small quenching distance: Hydrogen has a lower quenching distance than gasoline. Hydrogen flames cruise closer to the cylinder wall than other fuels before they extinguish. The less quenching distance increases the backfire tendency as the hydrogen-air mixture passes closer to intake valve than hydrocarbon flames [9].
• High auto-ignition temperature: Hydrogen has a relatively high autoignition temperature, which enables increasing compression ratio, which results in increasing the brake power and the thermal efficiency of the system [10].
• High diffusivity: Hydrogen characterized by its high diffusivity, which enables it to disperse in the air in a greater manner compared to gasoline. This high diffusivity is advantageous for two main reasons; firstly, It facilitates the formation of fuel and air mixture. Second, in the case of a hydrogen leak, it dispersed quickly. However, this can avoid unsafe conditions or reduced it when the engine fueled with hydrogen [11].
•Very low density: Hydrogen has very low density, which results in two main problems when it is powered the internal combustion engines. Firstly, the storage of enough hydrogen demands an enormous volume to supply the vehicle with adequate driving range. Secondly, the energy density of a hydrogen-air mixture, and hence the power output, is reduced [12].
•High Flame Speed: The hydrogen flame speed is high at the stoichiometric ratio. The hydrogen engines can approach the ideal engine thermodynamic cycle. However, at leaner mixtures, the flame velocity decreases significantly [13], [14].
Hydrogen fueled engines suffered a major setback. In fact, several abnormal combustion features are likely to crop up with hydrogen fueled the engines designed to run on petrole-

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um-based fuel. Using hydrogen as a fuel in spark-ignition engines appears three types of abnormal combustion. The knock, which is the fuel auto-ignition at the end gas region, differs from the pre-ignition which is the uncontrolled ignition prompts by hot spots and occurs before the spark ignition. The backfire, which is also called as back flash, flashback and induction ignition differs from knock and pre-ignition. The above mentioned conditions precede the ignition during the intake stroke [15]. Backfire has been a particularly tough obstacle to the development of hydrogen engines.

TABLE 1

PROPERTIES OF HYDROGEN, METHANE AND GASOLINE

Property	H_2	CH_4	gasoline
Density (kg/m^3)	0.0824	0.72	730
Diffusion coefficient in air (cm2/s)	0.61	0.16	0.05
Normal boiling point (K)	20.3	116.6	310-487
Flammability limits (volume % in air)	4-75	4.3-15	1.4-7.6
Flammability limits (Ø)	0.1-7.1	0.4-1.6	0.7-4
Autoignition temperature in air (K)	858	723	550
Minimum ignition energy (mJ)	0.02	0.28	0.24
Flame velocity (m/s)	1.85	0.34	0.37-0.43
Laminar burning velocity in air (m/s)	2-2.3	0.37-0.43	0.37-0.43
Adiabatic flame temperature (K)	2480	2214	2580
Quenching distance (mm)	0.64	2.1	2
Stoichiometric fuel/air mass ratio	0.029	0.069	0.068
Stoichiometric air/fuel ratio (Ø=1)	34.3	17.23	14.6
Stoichiometric volume fraction %	29.53	9.48	2
Lower heating value (MJ/kg)	119.7	45.8	44.79
Heat of combustion (MJ/kg$_{air}$)	3.37	2.9	2.83
Research octane No.	>120	140	91-99

The causes cited for backfire are:

- Hot spots in the combustion chamber.
- The spark plug.
- Residual gas.
- Exhaust valves.
- Residual energy in the ignition circuit.
- Induction in the ignition cable.
- The burning at the top of the piston reaches the inlet valve on the time of entry of the consignment and ignited the new charge.
- Pre-ignition.

All the reasons mentioned above cause the backfire Ignition for the charge, and the hydrogen engine should be designed in a way to prevent these phenomena. This hydrogen engine design can be regarded as an essential condition for the possibility of the spread of hydrogen engines [16].

Control of pollutant emissions from fuel burning engines is of major environmental concern worldwide, especially for engineers who design engine components with the aim of minimizing the emission of nitrogen oxides (NOx). NOx is a very undesirable emission and play a major role in the formation of acid rain, greenhouse effect and the global warming issue and hence accelerates the process of ice cap melting in north and south poles. Hydrogen engines distinguished by emitting higher levels of NOx concentrations compared with hydrocarbon fuels [17].

EGR has been one of the key technologies due to its ability to reduce exhaust NOx emission. EGR acts as additional diluents in the unburned gas mixture, thereby reducing the peak burned gas temperatures and NOx formation rates [18]. This NOx reduction mechanism of EGR is essentially accomplished by the following thermodynamic factors:

- The heat capacity of the exhaust gas is higher than the air entering the engine, and this reduces the temperature inside the combustion chamber where the amount of heat released during combustion stays equal.
- Reduction of O2 partial pressure results in a reduced oxygen mass in the cylinder since part of the available air for combustion is replaced by exhaust gas with lower oxygen content.
- Reduction of combustion speed results in an additional temperature reduction [19]-[20]-[21].

Exhaust Gas Recirculation (EGR) system is usually used to reduce NOx emissions and to enhance the fuel economy. EGR system can suppress knock since it offers the benefits of charge dilution. However, the occurrence of excessive cyclic variation with high EGR rates makes the engine combustion undesirable leading to the deterioration of both engine performance and emitted emissions. The optimum EGR rate must be supplied as a function of the operating conditions to avoid the reduction of thermal efficiency and to improve fuel economy [22]-[23].

The aim of this paper was to investigate the impact of EGR in several volumetric fractions on hydrogen engine performance and emissions. In this study two methods of EGR (hot and cold) were examined. In continuous of the previous studies [24]-[45] in the Energy and Renewable Energy Technology Center, we are focusing on the greater use of alternative fuels as it is green fuels, available, and suitable for Iraq conditions.

2 EXPERMINTAL TECHNIQUES

The investigation was carried out on a single cylinder, 4-stroke spark ignition Ricardo E6 engine with variable compression ratio, spark timing, and equivalence ratio, fueled with gaseous hydrogen. Hydrogen supplying system was consist of a gas cylinder, pressure regulator, chocked nozzles system as gas flow rate measuring device and as a flame trap. Alcock air flow meter was used to measure air flow rate; a tachometer was used to measure speed. All the measuring devices were calibrated several times during the work.

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

- A magnetic oxygen analyzer for O2.
- A chemiluminescence analyzer for NO and NO2.

The engine was operated with pure hydrogen produced by Al-Mansur Company with 99.99% purity.

2.1 EGR SYSTEM

The exhaust gas was recirculated by extracting it after a flange connecting the exhaust gas manifold and the exhaust pipe. This flange is 35 cm downstream from concourse point. By this arrangement, the EGR driving force was the pressure difference between the exhaust and the intake manifold pressure.

In the case of hot EGR, the EGR desired amount controlled by a control valve placed 50 cm from the copper tube from the extraction point. The feedback point of the EGR located at the end of gas mixer adapter 3 cm downstream of the mixer to avoid the interaction between the recycled exhaust gas and residual gasses at valve overlap period as efficiently as possible.

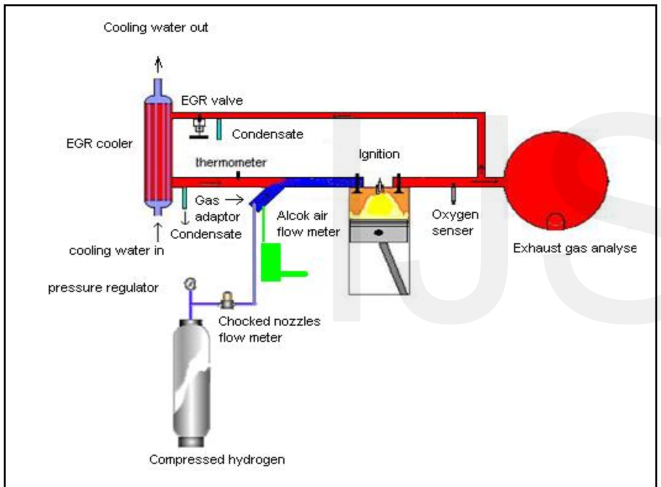

Fig. 1. Experimental layout of the tested engine with cooled EGR

The following equations were used to calculate important engine variables:

1. Brake power (bp):

$$b_p = \frac{W_b \times N}{348.067} \quad \ldots\ldots\ldots\ldots\ldots (1)$$

Where: Wb- acting load.

N – Engine speed in revolution per second (rps).

2. Equivalence ratio:

$$Ø = \frac{(\frac{A}{F})_{stoichiometric}}{(\frac{A}{F})_{actual}} \quad \ldots\ldots\ldots\ldots (2)$$

3. Brake specific fuel consumption:

$$bsfc = \frac{\mathring{m}_f \times 3600}{b_p} \ldots\ldots\ldots\ldots\ldots (3)$$

Where: $m^o{}_f$- hydrogen fuel consumption rate.

4. Indicated thermal efficiency:

$$\eta_{ith} = \frac{b_p + f_p}{\mathring{m}_{hydrogen} \times (LHV)_{hydrogen}} \ldots\ldots\ldots\ldots (4)$$

Where: - $m^o{}_{hydrogen}$: hydrogen flow rate.

- $(LHV)_{hydrogen}$: hydrogen lower heating value.

Tests procedure

Variable tests conducted during this investigation. In the first set, the engine was run at speed (1500 rpm) with hydrogen, and the higher useful compression ratio (HUCR) for it determined at optimum spark timing (OST) and full load for each point tested. The tests started from CR=8, which is a common engines compression ratio. At this CR the engine performance was studied in detailed to find the effect of spark timing and equivalence ratio on engine performance and emissions. For all the experiments, the exhaust gas delivered to gas analyzer to determine emissions concentrations. In the second set of tests, all the experiments were examined using hot EGR, starting from pure hydrogen compression ratio resulted from the first set of tests. In the third set cooled EGR was used, and finally there were comparison between all the results at HUCR for each system to evaluate the engine performance and emitted emissions.

3 RESULTS AND DISCUSSION

Fig. 2 shows the changes in brake power with variable compression ratios. The tests were conducted to determine the HUCR. From the figure appears that HUCR for pure hydrogen is 11:1. At this rate, the maximum brake power achieved while at compression ratios above this ratio the pre-ignition phenomenon limited the engine operation for certain equivalence ratios on the lean side. The engine operated at Ø=0.85 at CR=12.5:1, and ran at CR=13:1 to equivalence ratio about Ø=0.54. The engine operation at higher compression ratio is possible to attain higher brake power (bp) by using special techniques to prevent the air-fuel charge pre-ignition. From these techniques, injecting water into combustion chamber to cool the mixture; using cold spark igniter, and cooling the valves with keeping combustion chamber clean from combustion residuals.

Fig. 2 clarifies that bp increased with equivalence ratio increase, from the lean side to rich side. The maximum bp attained at Ø=0.95, with CR increase the equivalence ratio at which the maximum bp occurred reduced to reach Ø= 0.9. Oxygen availability and increased flame propagation speed are the reasons for the high bp. The increasing load after this ratio caused abnormal combustion phenomenon, resulting in bp reduction. The upper flammability limit of hydrogen at stoichiometric equivalence ratios is the reason for this phenomenon.

Using hydrogen help working at very lean equivalence ratios cannot be reached with any other fuel, and much more lean equivalence ratio can be achieved by increasing CR. The min-

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imum lean limit obtained in these tests was Ø=0.32. It is larger from those reached by some researchers; the reasons are the incomplete mixing of hydrogen and air, where fuel supplied to engine near intake valve. Besides, the engine has no design parameters to increase mixture turbulence. Nevertheless, the low lean equivalence ratio at which the engine was run cannot be reached with any other known fuel.

Hydrogen is well suited for internal combustion engines due to its combustion characteristics. Its broad flammability limits permit very lean operation. High hydrogen engine efficiency is possible due to the high flame speeds of hydrogen-air mixtures near stoichiometric. These characteristics, however, introduce the particular challenges of NOx emission control. The high combustion rate and preignition during high load operation may limit the specific power output of hydrogen engines.

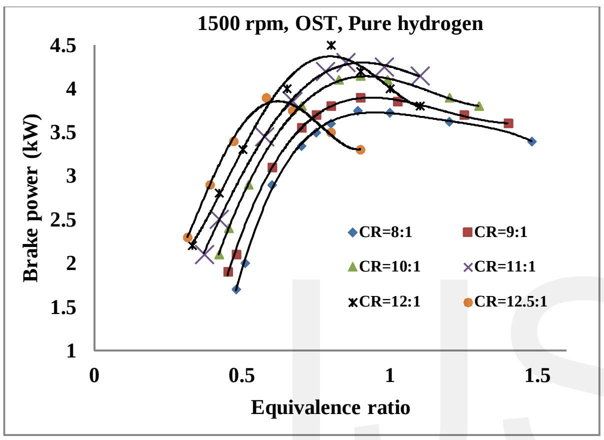
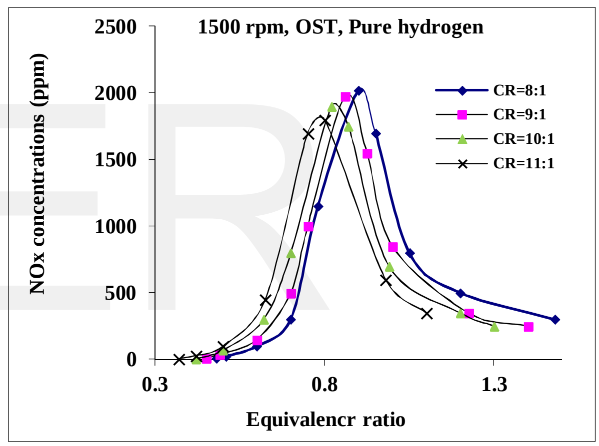


Fig. 2. Compression ratio effect on bp for wide range of equivalence ratios using pure hydrogen

Hydrogen is a clean fuel because of the absence of carbon atoms. Apart from the minimal amount of CO, CO_2 and HC's in the exhaust gas because of the burning of lube oil. The only pollutant emitted from hydrogen combustion is NO and NO2, thus NOx. The NOx concentration is primarily a function of the air-fuel ratio and reaches its maximum values at about Ø=0.8. This peak is even higher than when operating the engine with gasoline. However, these NOx concentrations decrease extremely rapidly when the air-fuel mixture gets leaner. Fig. 3 represents the impact of increasing CR on emitted NOx levels. On the lean side, NOx concentrations increased because the combustion temperature increased, accompanied by oxygen availability for NOx formation. NOx levels reduced on the rich side with CR increase, because of the lack of the oxygen needed for the reaction. Also, the dissociation reactions freeze at expansion stroke. Increasing CR reduces the maximum value of NOx in exhaust gas, because of interfere of another parameter with CR; it is spark timing retardation, which reduced the available time for higher NOx concentration formation. NOx levels with hydrogen fuel are high, because of its high combustion speed, which produce higher pressure and temperatures especially in the lean equivalence ratio (Ø=0.8). Hydrogen fuelled SI engines emitted only NOx and limited engine operation at medium to the high loads. Several strategies were developed to reduce NOx emissions. Engine operation at equivalence ratios less than 0.5 manages the engine to emit lower NOx concentrations less than 100ppm. For leaner equivalence ratios less than 0.5, NOx emissions are close to zero. However, some special action is required to recover the power lost due to the lean operation.

NOx showed a declining trend in equivalence ratios higher than 0.8. The patterns are understood concerning the relative effects of combustion temperature and oxygen concentration which together determine the production of NO.

Fig. 4 shows the CR effect on optimum spark timing (OST) for wide range of equivalence ratios. It appears that increasing CR causes the OST to retard, because of increase in compressed mixture temperatures in combustion chamber, which results in increasing its combustion. The growth of equivalence ratio from very lean equivalence ratios causes the OST to retard because the mixture combustion speed increased. When the hydrogen-air mixture reaches the equivalence ratio at which the engine has the maximum bp, after that OST advanced due to the lack of oxygen in the combustion chamber that reducing the burning velocity.



F.ig 3. Compression ratio effect on NOx concentrations for wide range of equivalence ratios using pure hydrogen

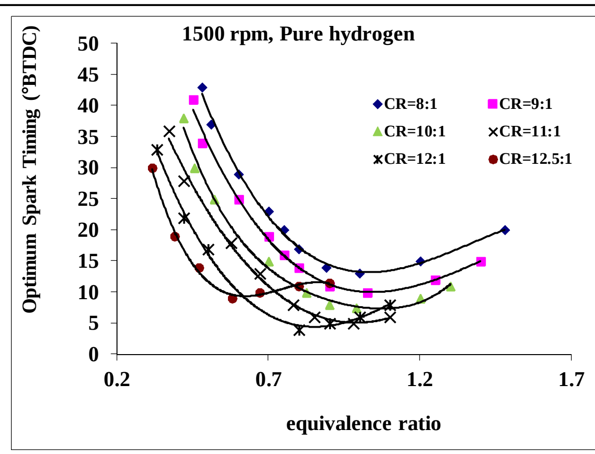


Fig. 4. Compression ratio effect OST for wide range of equivalence ratios using pure hydrogen

Fig. 5 illustrates the effect of CR on indicated thermal efficiency. Indicated thermal efficiency increases with CR increase, which means improvement in released thermal efficiency.

The maximum value of indicated thermal efficiency located at very low equivalence ratios, it increased with CR increase and decreased quickly with equivalence ratio increase. Nevertheless, it still higher than any indicated thermal efficiency recorded for any other fuels, indicating better reactions of hydrogen.

This efficiency enhancement appears when studying CR effect on exhaust gas temperature in Fig. 6. These temperatures reduced with CR increase; the high flammability limits of hydrogen which burn rapidly in the power stroke, so, when expansion stroke begins, most of the fuel, if not all, will be burned. Therefore, at exhaust stroke, the exhaust gas temperatures reduced. The maximum value of exhaust gas temperature was at equivalence ratio that gave maximum bp, and when the mixture was leaned or enriched the exhaust gas temperature reduced highly.

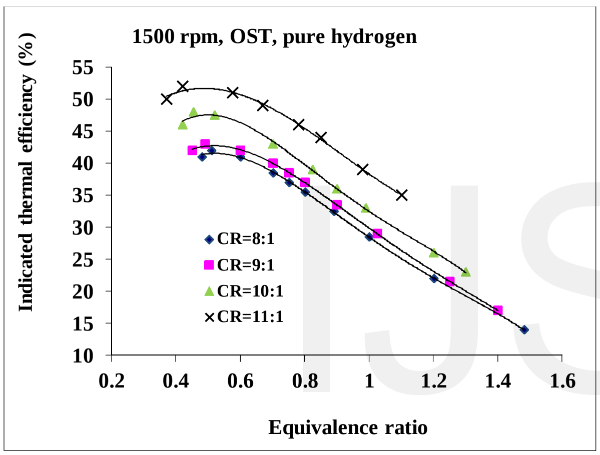


Fig. 5. Compression ratio effect on indicated thermal efficiency for wide range of equivalence ratios using pure hydrogen

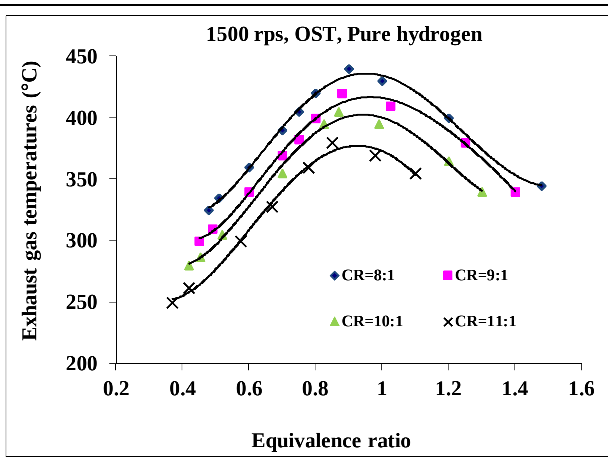


Fig. 6. Compression ratio effect on exhaust gas temperatures for wide range of equivalence ratios using pure hydrogen

Fig. 7 represents the effect of CR on brake specific fuel consumption (bsfc). Te bsfc reduced with CR increase, which means higher energy reactions inside the combustion chamber. When Ø increased the bsfc reduced until it reached its minimum value at Ø=0.55, after this ratio it began to rise highly with mixture enrichment.

Engine volumetric efficiency decreased rapidly with equivalence ratio increase because hydrogen displaced a significant volume when it enters the combustion chamber. This volumetric efficiency reduction effects produced bp as fig 8 shows. The mechanical efficiency increased until reached its maximum value then reduced a little after this point, still as a function for engine bp.

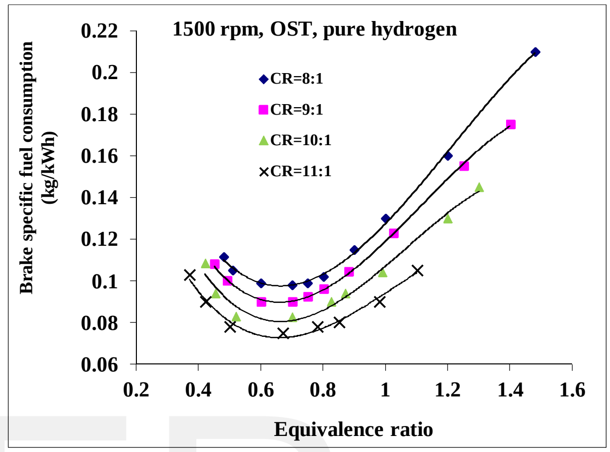


Fig. 7. Compression ratio effect on bsfc for wide range of equivalence ratios using pure hydrogen

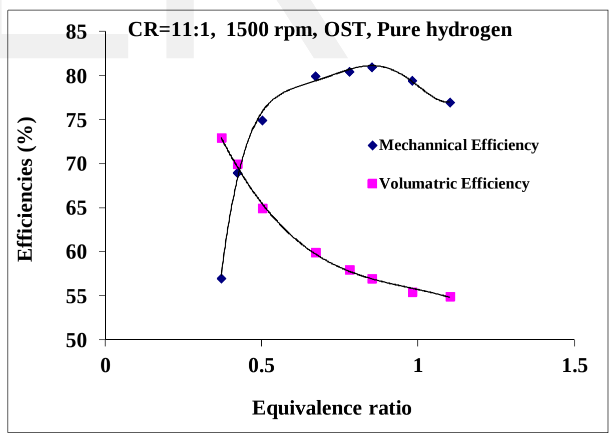


Fig. 8. Compression ratio effect on mechanical and volumetric efficiencies for wide range of equivalence ratios using pure hydrogen

Ignition timing considered most affecting factor on hydrogen engine operation, where hydrogen ignition and burning velocities are fast. The high burning speed of hydrogen makes the mixture burn time very short. Also, makes the spark timing changing range very limited, especially when the engine operates near stoichiometric equivalence ratios. Where the highest mixture burning velocity occurred, As Fig 9 declares.

The OST for the equivalence ratio (Ø=0.4) can be advanced away from OST about 10 degrees, and retarded to about 15 degrees, without occurrence of any abnormal combustion phenomenon (like high pressure rate before top dead centre, pre-ignition and backfire). However, when engine operated with equivalence ratio (Ø=0.63), the ignition timing operated degrees range reduced. At this equivalence ratio, the spark timing cannot be advanced more than 7.5 degrees from OST, or retarded more than 10 degrees. The pre-ignition phenomenon happened if the spark timing was increased or reduced from these regions. By hydrogen enrichment, the available spark timing range reduced. The hardest working part was near stoichiometric equivalence ratio, where advancing and retarding ST without abnormal combustion occurrence was about 2.5 degrees from OST. It is obvious that any hydrogen engine operation must be accompanied by ST amendment to prevent abnormal combustion.

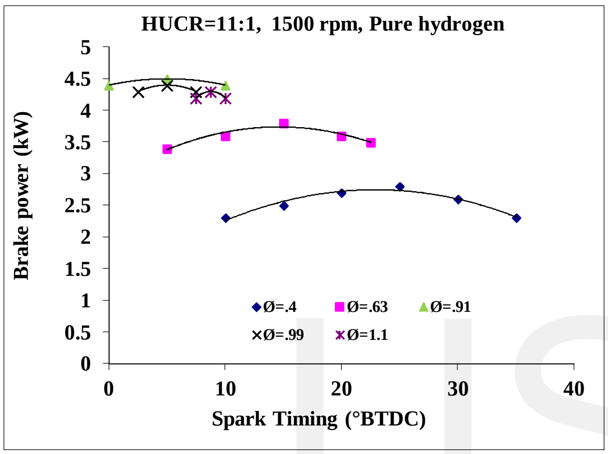


Fig. 9. Spark timing effect on bp for specific equivalence ratios using pure hydrogen

Fig. 10 shows a photo of the oscilloscope for highest pressure occurrence before top dead centre (with ST advanced about 5 degrees from the OST). The knock like phenomenon shown in the single cycle was observed with increasing burn rate at higher load and equivalence ratio approaching Ø=1. The higher effective octane number of hydrogen indicates that auto-ignition of the end gas may not explain this phenomenon. It may be attributed in part to the sustained high burn rate of the primary combustion event causing a resonance in the chamber; the engine operation at equivalence ratio approaching Ø=1 and the ignition advances achievable at higher loads were limited by this phenomenon.

Fig. 11 represents pre-ignition phenomenon when engine operated at Ø=0.99 (with ST retarded 5 degrees from the OST). Pre-ignition, as indicated by premature cylinder pressure increase, was noted to occur at higher loads and with equivalence ratio close to Ø=1, in association with the highest rates of pressure rise. This pressure increase represents a high risk of engine damage.

Backfire observed as an audible event and abnormal pressure increase in the intake system occurring when the intake valve open and with some time before intake valve closure. The mixture self-ignites before the intake valve has closed causing a flame to flash back into the intake port. In this work, backfire was eliminated with lean mixture operation, and by preventing pre-ignition.

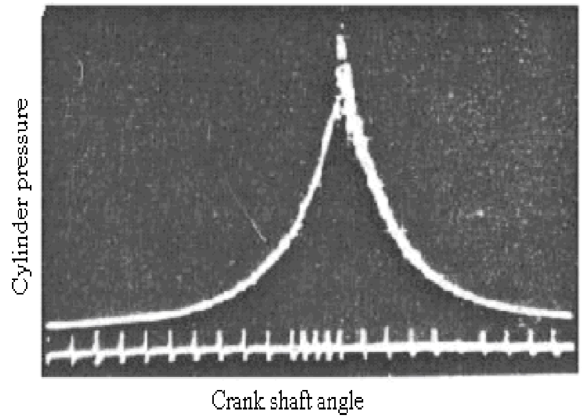


Fig. 10. oscilloscope picture indicates high-pressure rates before TDC

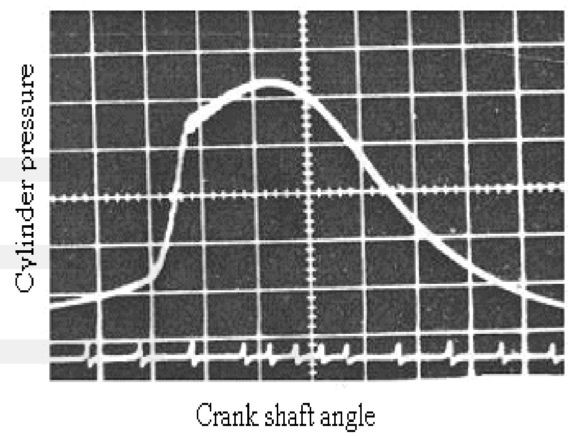


Fig. 11. oscilloscope picture indicates pre-ignition phenomenon.

EGR is widely used to reduce NOx, the pumping loss, and to increase thermal efficiency in engines. However, it does have some disadvantages such as its detrimental effects on combustion stability. EGR acts as additional diluents in the unburned gas mixture and thereby reducing the peak burned gas temperatures and NOx formation rates.

When the engine was run with 10% volumetric fraction hot EGR, its compression ratio could be increased more than the HUCR for neat hydrogen (Ø=11). It was possible to improve CR and extend the equivalence ratio range. The HUCR for this case as Fig. 12 shows was 13:1. Also, the engine managed to operate with higher rich mixtures reached Ø=1.33 while working with the pure hydrogen the equivalence ratio stopped at Ø=1.1, due to heavy pre-ignition.

A more homogeneous mixing of air-hydrogen and recirculated exhaust gas slow down the high burning velocity of hydrogen due to the EGR dilution. Adding EGR assisted for stable combustion.

Using cooled EGR gives the same results with more improvements, as Fig. 13 represents. The HUCR, in this case, was higher; it reached 14:1, and the equivalence ratio extended to Ø=1.35, and the more important parameter was the elimination of abnormal combustion phenomenon especially at equivalence ratios near stoichiometric.

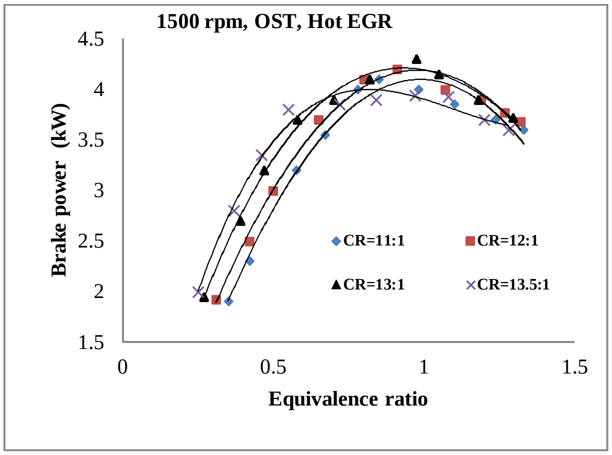


Fig. 12. Compression ratio effect on bp for wide range of equivalence ratios using hot EGR

For comparison aim, the resulted brake power at CR=11 which is the HUCR of hydrogen fuel was studied in Fig. 13. Brake power reduced with hot EGR about 17%, while it reduced about 9% with cooled EGR. The hot EGR displace volume larger than what cooled EGR move away, reducing the volumetric efficiency, hence the resulted bp less than cooled EGR's bp.

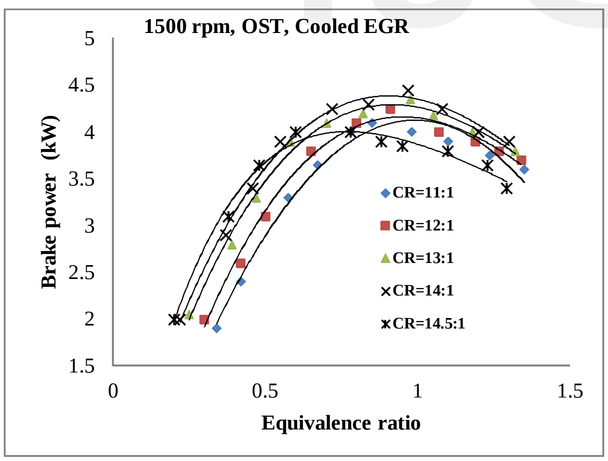


Fig. 13. Compression ratio effect on bp for wide range of equivalence ratios using cooled EGR

When the engine runs at HUCR for each system, the resulted bp differs in Fig. 14 from the last one. The hot EGR's bp increased but still less than pure hydrogen's bp, but the cooled EGR's bp passed them, it was higher than pure hydrogen's bp with about 3.5%. Compression ratio increments improved the combustion, and raised bp; this improvement was limited in hot EGR because of the reduction of volumetric efficiency.
Diluting the hydrogen – air mixture with 10% EGR resulted in OST advanced with about 9 °BTDC for hot EGR and about 15°BTDC for cooled EGR at equivalence ratios between Ø=0.8-1.05. Optimum spark timing was advanced about 20°BTDC for hot EGR and about 25°BTDC for cooled EGR at equivalence ratios within Ø=1.05-1.3, as fig. 15 reveals. Hydrogen distinguished by its high burning velocity, diluting the air-fuel mixture with uncombusted gas affected the burning speed and imposing it to reduce. Then, to compensate this reduction and to improve engine performance ST advanced, from the figure it was evident that cooled EGR needed more advance from the hot one, because advancing spark timing increased the combustion temperatures inside the combustion chamber, and improved the resulted bp, and these advanced degrees recompensed inner mixture temperatures difference.

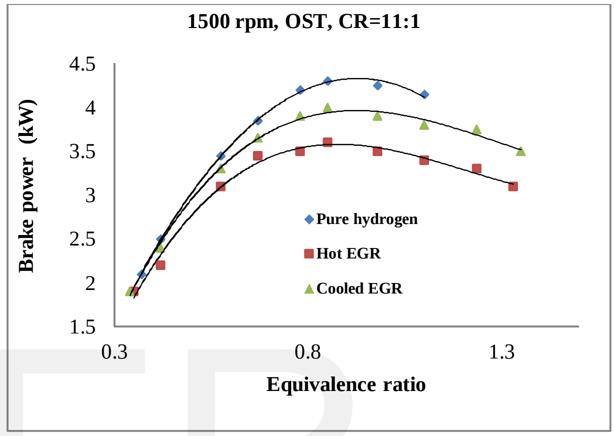


Fig. 14. Comparison between the three systems bp produced with wide range of equivalence ratios at CR=11:1 for the three systems

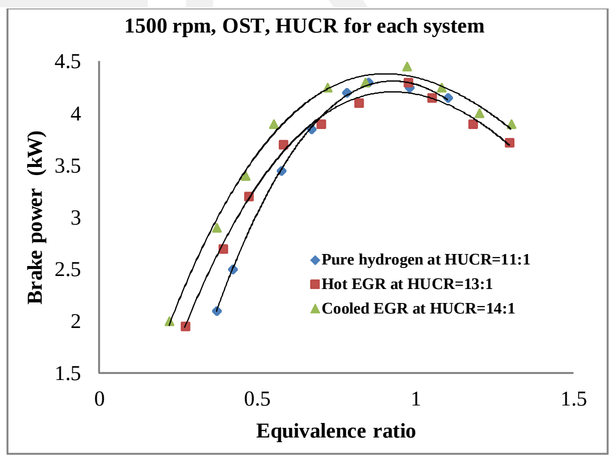


Fig. 15. Comparison between the three systems bp produced with wide range of equivalence ratios at HUCR for each system

Figure 16 shows a reduction in exhaust gas temperatures when EGR systems were used, and it was diminished more for cooled EGR. This behavior can be explained in light of the reduction in combustion temperature. Taking into account the fact that the reduced combustion temperature results in lower NOx emissions, those could also be verified since it is well-known that NOx is very sensitive to flame temperature. Exhaust temperature is higher for the uncooled EGR due mainly to the elevated inlet temperature.

Volumetric efficiency reduced when EGR systems were used, as fig 17 clarifies, it was reduced with hot EGR more than the cooled one, owing to the volume occupied with hot diluting gas larger than that for cooled gas. This abatement in volumetric efficiency is the cause of bp reduction; its effect was apparent on the engine performance.

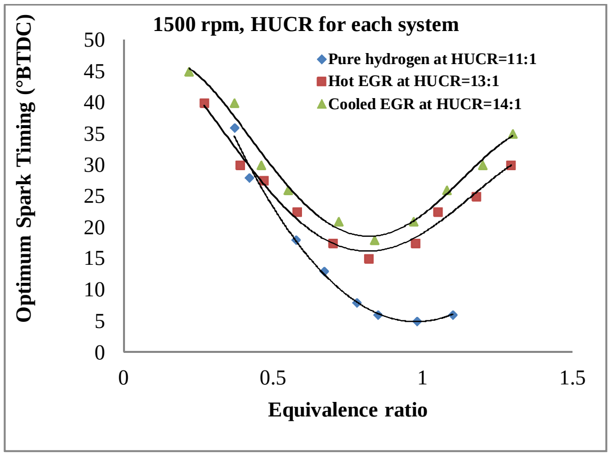


Fig. 16. Comparison between the three systems OST used for wide range of equivalence ratios at HUCR for each system

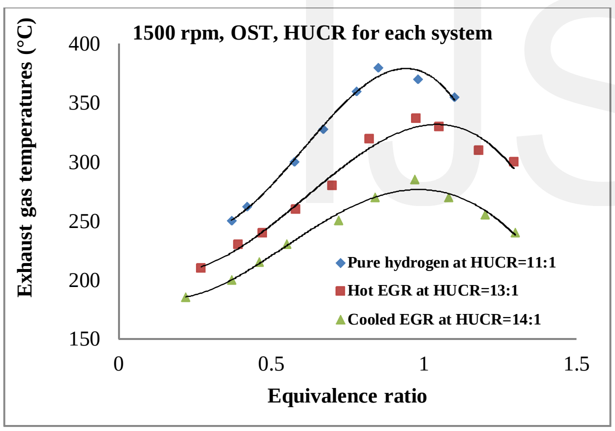


Fig. 17. Comparison between the three systems exhaust gas temperatures resulted for wide range of equivalence ratios at HUCR for each system

Spark timing is the most affecting factor of hydrogen engine operation as mentioned above, advancing or retarding spark timing is very limited with pure hydrogen, but with EGR systems, the condition changed. Figs. 18 to 21 review the spark timing influences. In Fig. 18 its effects were studied for very lean mixtures Ø=0.4, the results indicate higher bp for cooled EGR system at OST compared with hot one, and a significant reduction in it when advancing or retarding ST away from the optimum position. The same thing happened to the other systems with one variation, with cooled EGR the spark timing range extent wider than the others. A similar result recorded in Fig. 19 with equivalence ratios Ø=0.6 for the three systems, with expansion in the hot EGR spark timing range. At very lean mixtures, the operation was straitened due to low fuel quantity, but with these equivalence ratios the combustion became better, and ignition limits extended.

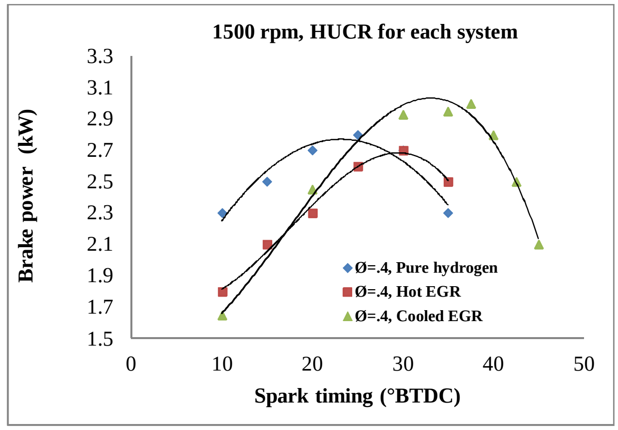


Fig. 18. Comparison between bp of the three systems resulted for wide range of spark timing at HUCR for each system and Ø=0.4

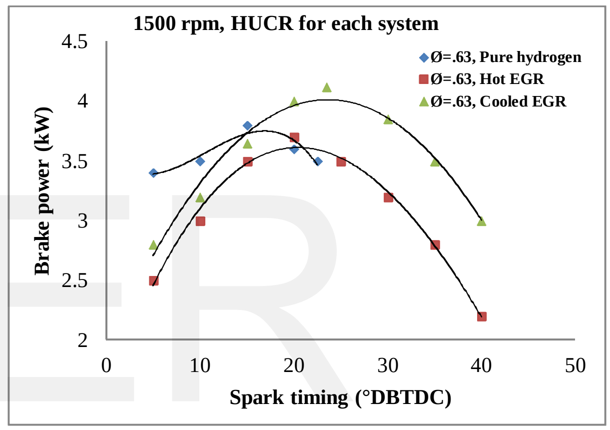


Fig. 19. Comparison between bp of the three systems resulted for wide range of spark timing at HUCR for each system and Ø=0.63

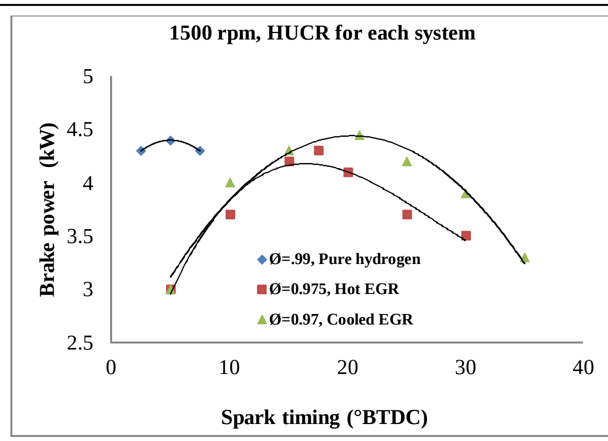


Fig. 20. Comparison between bp of the three systems resulted for wide range of spark timing at HUCR for each system and Ø near stoichiometric

At rich mixtures as Fig. 21 reveals, hydrogen engine refused to operate at equivalence ratios more than Ø=1.1, due to the pre-ignition occurrence. The high combustion chamber temperatures had ignited the mixture before the flame front reached it.

So, the spark timing range was very limited not exceed 2 to 3 degrees and its location near the top dead centre. In EGR systems, they were advanced away from the top dead centre, and expand for more degrees, although the resulted bp's were small. We must remember that the studied equivalence ratios can't be reached with pure hydrogen.

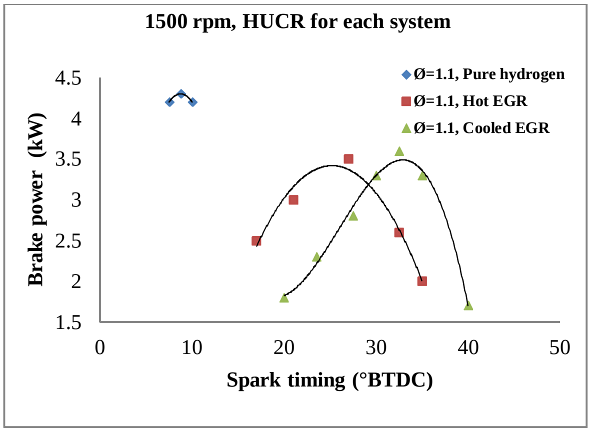


Fig. 21. Comparison between bp of the three systems resulted for wide range of spark timing at HUCR for each system and highly rich mixtures

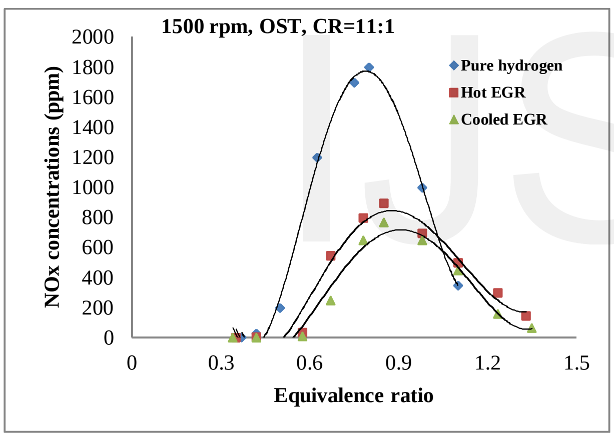


Fig. 22. Comparison between NOx concentrations of the three systems resulted for wide range of equivalence ratios at CR=11:1

Figs. 22 to 24 represent the EGR effects on engine emitted emissions. In Fig. 22, a comparison between the emitted NOx concentrations for the three fuel systems is illustrated. When the engine ran at CR=11:1 (which is the HUCR for pure hydrogen), the engine operation at equivalence ratios less than Ø=0.6 resulted in very low NOx concentrations with EGR systems. The engine needs to operate at equivalence ratios less than Ø=0.45 with pure hydrogen to confirm these levels. In light of improving fuel economy, the lean burn system is more efficient than EGR system because the specific heat ratio of excess air is higher than that of recycled exhaust gas. At equivalence ratios between Ø=0.7- 1.0 where the maximum NOx concentrations exist, the resulted concentrations with hot EGR reduced by about 45% compared with pure hydrogen, and reduced by about 55% with cooled EGR. Diluting effects on burning velocities caused this reduction in NOx concentrations.

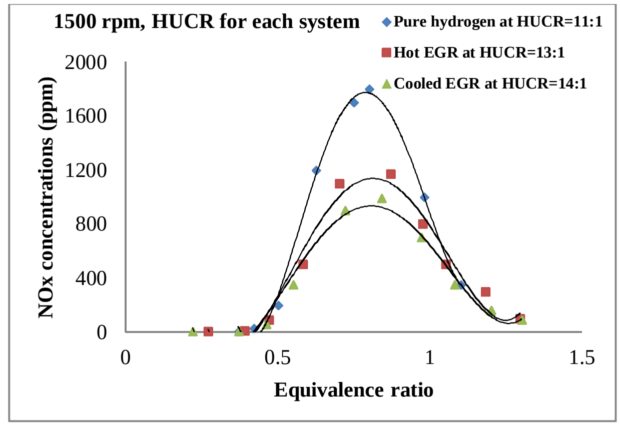


Fig. 23. Comparison between NOx concentrations of the three systems resulted for wide range of equivalence ratios at HUCR for each system

In Fig. 23, the comparison was made when each system operated at its HUCR, the resulted NOx concentration from hot EGR reduced by about 39% while the cooled EGR emitted NOx reduced by 45%. The specific heat capacity of the exhaust gases, which consist primarily of water and nitrogen gas, increased dilution in the induced charge and reduced the hydrogen burning velocity and the flame temperatures. Increasing CR increased combustion chamber temperatures, causing higher emitted NOx concentrations, so when working at HUCR, the engine needs another control on NOx in spite of EGR addition.

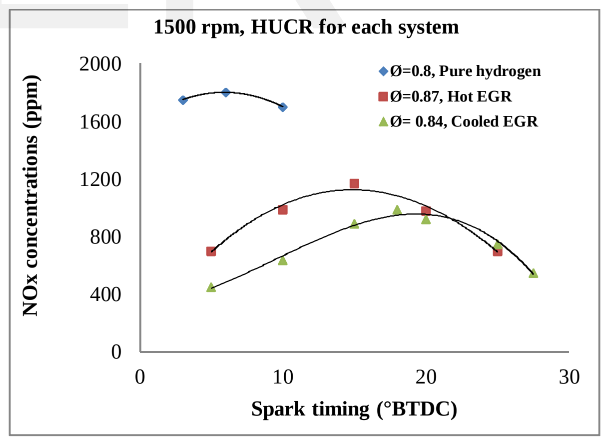


Fig. 24. Comparison between NOx concentrations of the three systems resulted for wide range of spark timings at HUCR for each system

Owing to more control on NOx concentrations, the ST effect on the equivalence ratios that resulted in the maximum NOx concentration was studied for each case as Fig. 24 declares. The engine operation with pure hydrogen means working at OST while when EGR systems employed, the ST ranges extended. For hot EGR engine with retarding ST 10°BTDC from the OST, NOx concentrations reduced by 61% compared with pure hydrogen, and 40% reduction achieved lower than its levels at OST. For cooled EGR retarding ST with 13°BTDC away from OST, NOx concentration dropped 75% compared with pure hydrogen, and 54.5% reduction of its levels at OST was achieved. Retarding ST considered as an active factor to control NOx concentrations, because it delayed the spark ignition, reducing the burning temperatures, and the heat produced inside the chamber, this will reduce the resulted bp also, so a compromise must be taken to prevent extreme retarding which affects engine performance.

4 CONCULOSIONS

The internal combustion engine, particularly the spark ignition engine, is very suitable for the use of hydrogen as a fuel. Some of the hydrogen properties are advantageous like the wide flammability range (omitting throttle), high burning velocity (efficiency), high auto-ignition temperature (compression ratio) and high diffusivity (mixture formation and safety). Other properties involve some difficulties like low ignition energy (pre-ignition and backfire), small quenching distance and density (power loss and storage problems). The purpose is to make the most of the 'good' properties and to conquer the drawbacks caused by the undesirable characteristics of hydrogen.

In this work, hot and cooled recirculated exhaust gas added to the hydrogen-air mixture in the suction manifold. EGR proved to be an efficient method to eliminate the emitted NOx. The engine power output can be preserved by altering the amount of recycled exhaust gas, instead of throttling, thus avoiding engine efficiency penalties.

The exhaust gas temperature decreased according to the decrements in the combustion temperature. The reduction of combustion temperature results in lower NOx emissions, as NOx is very sensitive to flame temperature.

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