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Influence of Hydrogen Rich Gas Addition on Performance Features of a Gasoline Direct Injection (GDI) Engine Operation

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Abstract: In a practical sense, lean burning engines are limited by onset of engine misfiring as lean flammability limit of any fuel is approached. Hydrogen may be used to extend lean limit of conventional fuel in order to stabilize lean combustion. No nationwide distribution system exists for hydrogen and its storage as a high-pressure gas or cryogenic liquid. These potential difficulties can be avoided by generating hydrogen in an on board gas generator using partial oxidation method from hydrocarbon fuel. Plasma fuel reformer provides (PFR) electrical discharges at high temperature boosts partial oxidation reactions between gasoline and air, producing hydrogen-rich gas (H_2).

The aim of the research was to extent the lean operation limit of GDI engine with supplementation of hydrogen rich gas from on board plasma fuel reformer (PFR). Performance tests were conducted on GDI engine under varied operating parameters such as air fuel ratio, spark timing and injection timings with definite fraction of hydrogen rich gas. Engine lean burn limit could be extended by hydrogen addition because hydrogen fuel has broader burn limit and fast burn speed. 10%, 18%, and 30% hydrogen rich gas fraction extended the lean limit equivalence ratio 0.68, 0.62, and 0.52, respectively, where 0.58 is the lower limit equivalence ratio of Gasoline. Unburned HC emission decreased with increasing fraction of hydrogen addition irrespective of spark timing alterations. After optimizing spark timing to MBT, engine efficiency increased with increase of hydrogen fraction. NO_x emission for different hydrogen fraction showed little difference at the MBT spark timing.

Keywords: GDI engine, equivalence ratio, Hydrogen rich gas, lean limit, Plasma fuel reformer.

Introduction

Decreasing emissions from automobiles and increasing engine efficiency are necessary steps towards improving air quality and decreasing green house gases. Transportation vehicles are the largest consumer of imported oil and a major source of pollutants that affect urban areas. A variety of potential improvements are currently being investigated for engine power output, fuel consumption and exhaust emissions: Spark-ignited direct – injection engines, new catalyst formulations, close coupled catalysts, new types of exhaust after treatment, electric and fuel-cell powered vehicle and alternative fuel.

An alternative and more basic approach to the emissions problem is to modify initial combustion process in the engine by using lean mixtures. The primary advantage of lean burn is that it increasingly reduces NO_x and CO, but problem is misfiring as the lean flammability limit of any fuel is approached. Hydrogen may be used to extend the lean limit of conventional fuel in order to achieve higher efficiency and lower pollutant emissions.

The introduction of hydrogen as a supplemental automotive fuel could be hindered by serious logistic problems. No nationwide distribution system exists for hydrogen and its storage as a high-pressure gas or cryogenic liquid requires vehicle capabilities which do not exist commercially [1].

In response to the storage problematic of hydrogen, one possible way to ensure the feed of hydrogen on the vehicle would be to store it in liquid fuels and then to produce hydrogen out of the fuel. The potential difficulties of storage problem can be solved by generating hydrogen in an on-board gas generator by fuel reforming technique [2].

Conventional catalytic technology used for fuel reforming, essentially in the case of small and moderate scale portable applications has certain problems make the reformers commercially not viable. Substantial improvements in fuel reformulation facilitate thermal plasma technology in the production of hydrogen and hydrogen rich-gas from methane and variety of fuels [3, 4, 5].

The main disadvantage of thermal plasma reforming is the dependence of an electrical energy. The non-thermal low current plasma converter developed at Massachusetts Institute of Technology (MIT) cambridge makes it possible to overcome this difficulty and drastically decreases energy consumption [6].

The use of low current high voltage non thermal plasma greatly reduces the specific electrical energy consumption and the electrode wear relative to thermal arc plasma reformers. The newly developed non-thermal low current plasma fuel converter technology is attractive for a variety of stationary application including distributed, low pollution electricity generation from fuel cells; hydrogen-refueling gas stations for fuel cell powered cars, decentralized hydrogen for industrial processes and onboard automotive applications [7, 8, 9]

The idea of adding hydrogen into conventional vehicle fuels to improve thermal efficiency and inhibit cyclic variation could date back to several decades ago. A relatively early research was investigated on combustion characteristics of a spark ignition engine using hydrogen enriched gasoline. Researchers concluded that small amount hydrogen addition could extend the lean limit and improve the engine's thermal efficiency as well as combustion stability [10, 11]

The present research was focused to develop a plasma fuel reformer (PFR) for on-board hydrogen-rich gas generation by non thermal plasma reforming from gasoline fuel. The plasma reformed gas was supplemented to a gasoline direct injection (GDI) engine and to study the effect on the combustion, performance and exhaust emission characteristics.

An Overview of Plasma Fuel Reformer (PFR)

In general fuel reformer is an electrical device that takes advantage of the finite conductivity of gases at very elevated temperatures for fuel conversion. The electricity required by the fuel reformer is provided by a low current high voltage transformer.

A novel device named Plasma fuel reformer (PFR) provides electrical discharges at high temperature boosts partial oxidation reaction in flowing gases of hydrocarbon fuel and air. The resulting generation of reactive species in the flowing gases along with increased mixing accelerates reformation of hydrocarbon fuels into hydrogen rich gas (H_2).

The device operates at atmospheric pressure, with air as the plasma forming gas. Air and fuel are continuously injected in a plasma region provided by a discharge established across an electrode gap. Most of the heating is provided by the exothermicity of the partial oxidation reaction. In the case of liquid fuels, approximately 15% of the heating value of the fuel is released in the partial oxidation reaction.

The hydrocarbon fuel reforming is partial oxidation reaction in which oxygen in air play the role of the oxidant:

$$C_mH_n + \frac{m}{2}(O_2 + 3.773N_2) \rightarrow mCO + \frac{n}{2}H_2 + \frac{m}{2}3.773N_2$$

The reformed gas contains substantial amount of hydrogen rich gas (H_2) and other composition of combustible gases. However for real time automotive application, reforming reactions are possible within narrow band of operating parameters. The hydrogen rich gas would be supplemented to GDI engine to achieve significant gains in efficiency, emissions and combustion stability.

Equivalent Percent Hydrogen Rich Gas (HRG) Fraction Definition

Before any hydrogen addition experiments could be performed an appropriate definition of the equivalent percent hydrogen had to be formed so that a direct comparison between the hydrogen addition and hydrogen rich gas (HRG) addition experiments is valid.

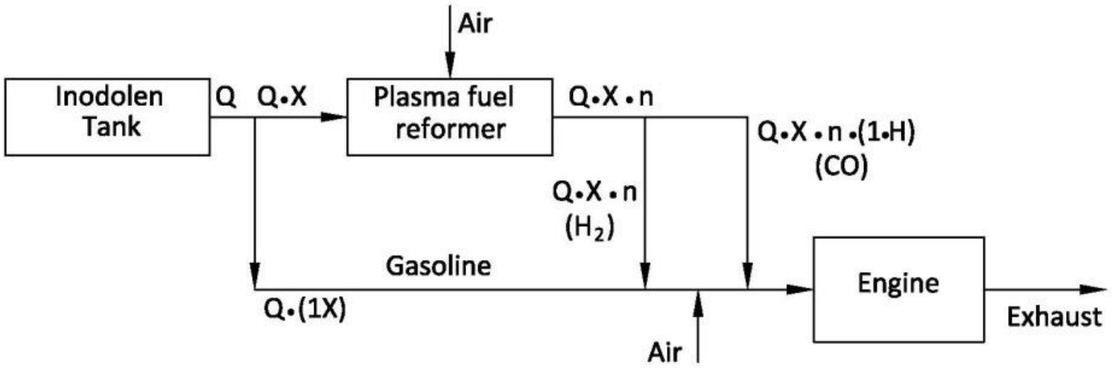

Figure 1. Energy balance with a plasma fuel reformer

Figure 1 shows an energy balance on the system. Equation 2 forms a ratio of the energy provided by hydrogen to the total energy entering the engine:

$$\frac{Q_{H_2}}{Q_{engine}} = \frac{Q_{ind} \cdot X \cdot \eta_{PLAS} \cdot H}{Q_{ind} \cdot X \cdot \eta_{PLAS} H + Q_{ind} \times (1 - X)} = \frac{X \cdot \eta_{PLAS} \cdot H}{1 + (\eta_{PLAS} - 1)} \quad \rightarrow (1)$$

Where, Q = Chemical Energy

η_{PLAS} = Plasmatron Efficiency

H = Fraction of the HRG gas energy supplied by hydrogen

X = Reformed Fraction (Equivalent Percent HRG gas)

$$H = \frac{\eta_{H_2} \cdot MW_{H_2} \cdot LHV_{H_2}}{\eta_{CO} \cdot MW_{CO} \cdot LHV_{CO} + \eta_{H_2} \cdot MW_{H_2} \cdot LHV_{H_2}} \quad \rightarrow (2)$$

Where, LHV = Lower Heating Value

n_x = Moles of component x

All of the variables on the right hand side of equation (2) are known for the ideal plasma fuel converter case. Inserting these values into (2) gives $H = 0.4521$. Finally, inserting numerical values for H and the plasmatron efficiency into equation (1) gives:

$$\frac{Q_{H_2}}{Q_{engine}} = \frac{0.3876 \cdot X}{1 - 0.1426 \cdot X} \quad \rightarrow (3)$$

Equation (3) can be used to calculate the fraction of the total energy entering the engine that should be provided by hydrogen to match a certain ideal plasma fuel converter experiment. For example, $X = 10\%$ gives 3.93% as the fraction of the total energy provided by hydrogen. In other words, setting the hydrogen energy fraction equal to 3.93% in a hydrogen addition experiments will provide an equivalent percent HRG gas of 10%.

Experimental Configuration

The overall experimental structure consists of GDI engine, Gasoline Direct Injection (GDI) system, on board PFR and Eddy Current Dynamometer as power measurement device as shown in fig.2. The PFR operates as an auxiliary device for performance enhancement of engine. The onboard PFR consists of three compartments namely fuel injection zone, plasma generation - fuel reaction zone, and heat exchanger zone configured in single stack.

The fuel injection system is an integral part of test engine and mainly comprises three parts: fuel supply system, electronic control unit (ECU) and injector assembly. The fuel supply system provides a constant pressure resource for the injector. The ECU controls the injection quantity and injection timing of the injector by special programs according to calculation and analysis of analog and digital inputs of various sensors.

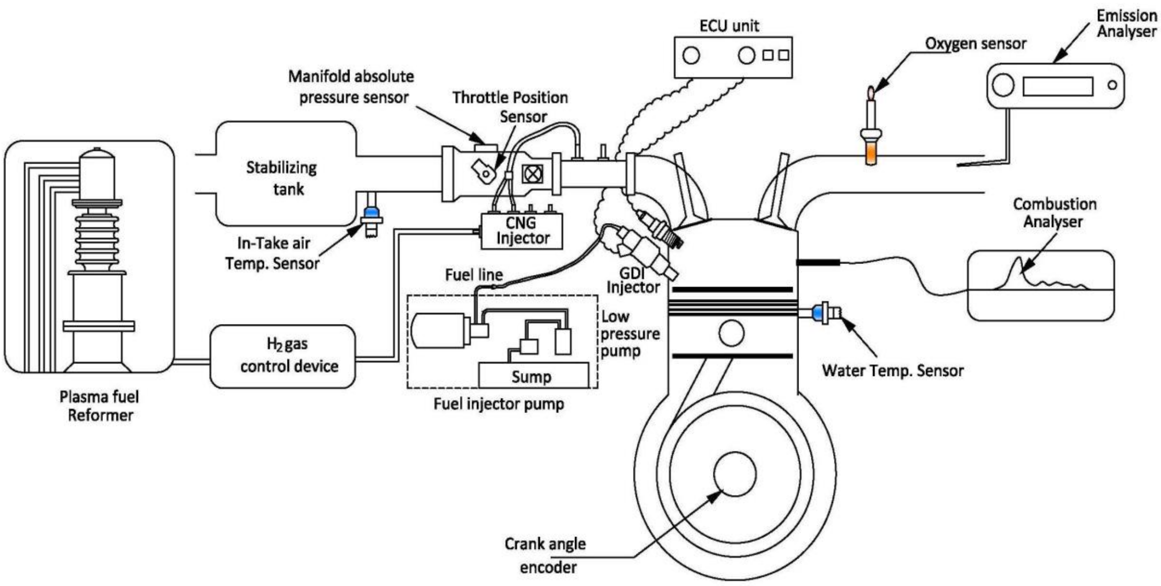

Figure 2. Experimental configuration of Lean Burn Engine

Engine Modification

A single cylinder four stroke 5 HP diesel engine was modified to operate a GDI engine for the proposed task. The preset compression ratio (CR) 17:1 of the diesel engine was modified to CR 9:1 by increasing the clearance volume in the engine head. Diesel injection system was removed from the engine and in the place of diesel pump a dummy flange was mounted to stop the oil spillage from the engine crankcase. Engine head was sectioned and examined to identify the location for mounting fuel injector, spark plug and combustion pressure sensor as the space is limited by the structure of the cylinder head for GDI development. The engine head was drilled with two holes of size M14 for mounting gasoline injector and M10 size for combustion pressure sensor.

Table 1. Specification of GDI Engine

Engine type	Single cylinder, Water cooled
Bore x Stroke	80 x110 mm
Displacement	484 CC
Compression ratio	9:1
Injection angle	Through 145˚.2' bTDC
Combustion chamber	Hemisphere open type
Piston	Flat - bowl piston
Ignition type	Coil on plug
Injector	Inward swirl
Nominal power/speed	3.7 kW/1500 ± 100 rpm
Max. Torque	5m

A spark plug was fitted in the place of diesel injector and an ignition coil named coil on plug was mounted on the cylinder body. A GDI injector was side-mounted on the cylinder head with an attachment. The spark plug and injector were connected with NI 9474 digital output module for ignition and injection timing of the engine. Other peripherals such as sensors, crank angle encoder were also connected with ECU system. The test engine was also provided with accessibility to combustion and emission measurement.

Experimental Procedure

The engine was coupled to an eddy current dynamometer for load measurement. Engine control management was carried out with custom built ECU control system which provided access to all calibration parameters. The ECU system allows the user to set a desired equivalence ratio and spark advance.

The exhaust concentration of HC, NO_x, and CO, and were measured by AVL Di gas analyzer. For ensuring precise measurement a glass fibre filter paper provided at the entry point of pickup probe was changed on par schedule. The emission pickup probe was mounted 60 cm from the exhaust manifold. In addition, air fuel ratio measurement was performed by a HORIBA wide-range lambda analyzer.

The in cylinder pressure data was measured using an AVL make piezoelectric 250 bar pressure transducer with the sensitivity of 16 pC/bar. An AVL 3057 charge amplifier converts charge yield by the pressure transducer into proportional electric signal. A personal computer was interfaced with an AVL 619 Indimeter hardware and Indwin software version 2.2 data acquisition system to collect combustion parameters. Crankshaft position was measured by a Kubler make crank angle encoder with resolution of 0.1°CA. Crank angle encoder is mounted on a base plate keyed to the engine frame. A toothed belt running over the pulley between extended camshaft and encoder shaft drives the crank angle encoder

The test was conducted in three phases: the first phase was to examine the lean operation limit of gasoline fuel. The second and third phase was to investigate engine’s thermal efficiency and emission characteristics with effect of addition of HRG in definite fraction at fixed spark timing and MBT respectively.

Results and discussions

Setting maximum brake toque (MBT) timing

Tests were conducted at different spark advance angle from 12° to 26° bTDC with an increment of 2° at each of start of injection angle (SOI). The tests report revealed that the engine produced maximum brake torque at specific spark advance as one such SOI angle 18° is shown in Figure 3.

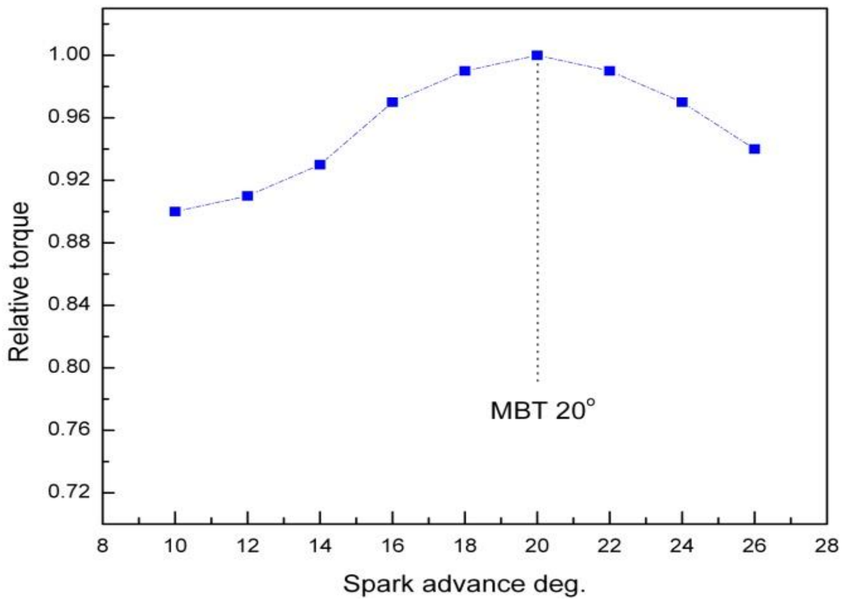

Figure 3. Relative Torque versus Spark Advance

It was observed that a maximum 10% loss of torque was measured against the reduction of 9.8% of average speed before reaching MBT at each of SOI. A minimum 1% loss of torque was recorded against the reduction 1% of speed when the spark angle approached nearer the region of MBT at each of SOI. Maximum torque produced when the engine was operated at 16°18° and 20° spark advance for 70, 140, and 210 degree SOI respectively. The engine started gaining speed as spark advances and attained the observed torque 23.55 N – m at MBT with specific air fuel ratio at wide open throttle. The observed torque begin to declined when the engine was operated beyond MBT at each of SOI. It was clearly evident from the trial that the maximum power condition of engine strongly responded to specific engine operating points only. Maximum power development was observed at MBT 20° bTDC with SOI 210 CA for the experimental configuration.

Effect of hydrogen addition on lean operation limit and combustion duration

Lean operation limit is an important parameter to represent the fuel's lean burn ability. It is generally accepted that a COV_{IMEP} above 10% will be perceived by a driver as a poor running condition [12]. Therefore, in this study, the lean operation limit was defined as the excess air ratio (reciprocal of equivalence ratio) at which COV_{IMEP} reaches 10%. COV_{IMEP} in this study was defined as the standard deviation in IMEP divided by the mean IMEP [13].

Fig. 4 shows the variation of COV_{IMEP} versus equivalence ratio and hydrogen rich gas (HRG) fraction, hydrogen fraction varied from 10%, 18% and 30% in volume. As can be seen in Fig. 4, lean limit got significantly extended as hydrogen fraction increased. Lean limit equivalence for gasoline is 0.58, but engine operating conditions forced to attained lean limit at 0.68. The fuel blends containing 10%, 18 %, and 30% HRG fraction extended lean limit 0.62, 0.58, and 0.52, respectively.

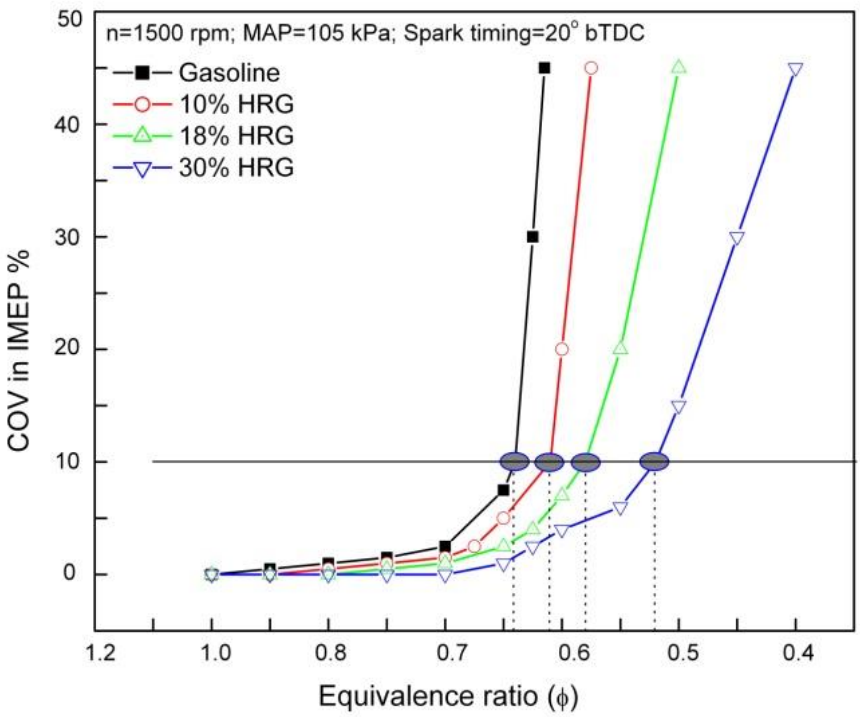

Figure 4. COV in IMEP versus Equivalence Ratio

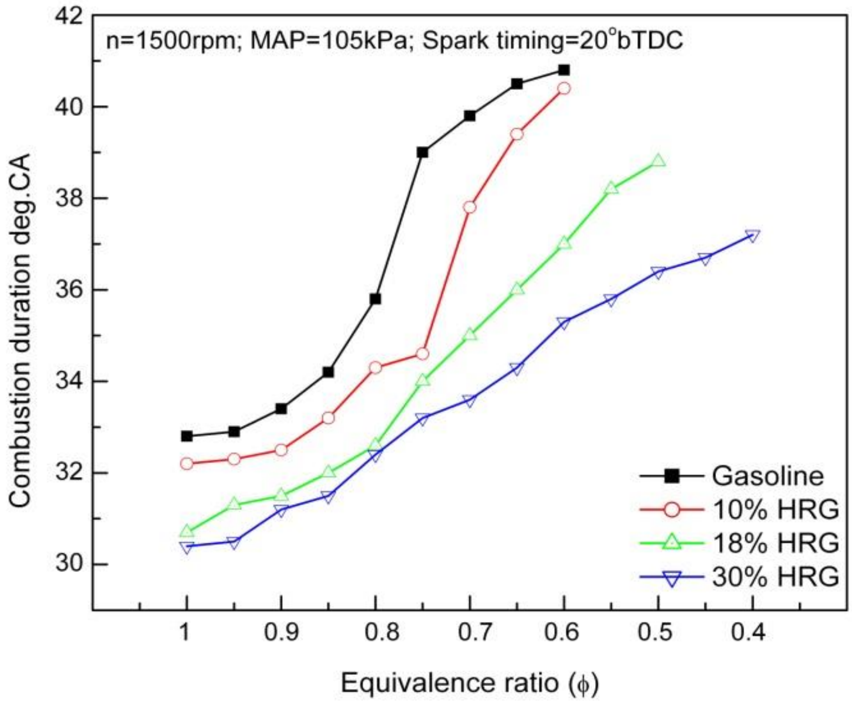

Figure 5. Combustion Duration versus Equivalence Ratio

Quader's re-searches have showed that the combustion duration is nearly the same when the engine reaches its lean limit no matter what type of fuel used [14]. This is to say although combustion duration will be prolonged as the engine is gradually leaned out, it has an upper limit which is independent on fuel type and once the combustion duration exceeds this upper limit, the engine would become unstable due to combustion instability. Therefore, a certain type of fuel will have greater lean operation ability if it provides shorter combustion duration at a given equivalence ratio, because it may require leaner fuel air mixtures to make the combustion duration reach the upper limit.

The above analysis makes it clear that examining the effect of HRG addition on combustion duration is essential to the analysis of hydrogen's ability to extend lean limit.

Figure 5 gives the variation of combustion duration versus equivalence ratio for different HRG fractions. At a given equivalence ratio, combustion duration shortened as hydrogen fraction increased. This illustrated that hydrogen addition could indeed speed up flame propagation.

Engine thermal efficiency and emission characteristics at fixed spark timing

Figures 6–8 show the emissions of NO_x, HC, CO and thermal efficiency as a function of equivalence ratio. From figure 6 it was observed that as the engine was gradually leaned out, NO_x emission increased rapidly, reached a peak at equivalence ratio 0.8, and then decreased gradually to a relatively small value, and this trend was independent on hydrogen fraction. It is also noted that NO_x emission could be very low after equivalence ratio reached 0.5. However, more HRG added would result in more NO_x emission at a given equivalence ratio, this is thought to be caused by the elevated combustion temperature due to hydrogen fraction since high temperature was a catalyst for the formation of NO_x.

Trend of HC emission is shown in figure 7. HC emission reached its minimum value when equivalence ratio was unity and slightly less than unity. At this region, there was extra air to ensure combustion completeness and on the other the fuel air mixture was not too lean, so the exhaust temperature could keep at a high level which was beneficial to the further oxidation of HC formed through crevice and flame quenching. In addition reduced HC emission by HRG addition was observed in this study could be explained by the fact that hydrogen could speed up flame propagation and reduce quenching distance, thus depressing the possibilities of incomplete combustion [15]. Moreover, the fact that carbon concentration of the fuel blends decreased due to hydrogen addition was also accounted for. But above equivalence ration 0.6 HC emissions steadily increased despite of HRG addition as amount of fuel in regions was very lean to burn during primary combustion process.

The formation of CO is mainly due to incomplete combustion. As can be seen in Figure 8, CO concentration first dropped gradually, reached a minimum value at equivalence ratio 0.6, and then started climbing rapidly due to poor combustion conditions as the engine was further leaned out. In the region where equivalence ratio was less than 0.6, adding hydrogen not showed significant difference on CO emission, but once equivalence ratio exceeded 0.6, more HRG addition resulted in much less exhaust CO. This was also attributed to hydrogen's ability to strengthen combustion, especially for lean fuel air mixtures.

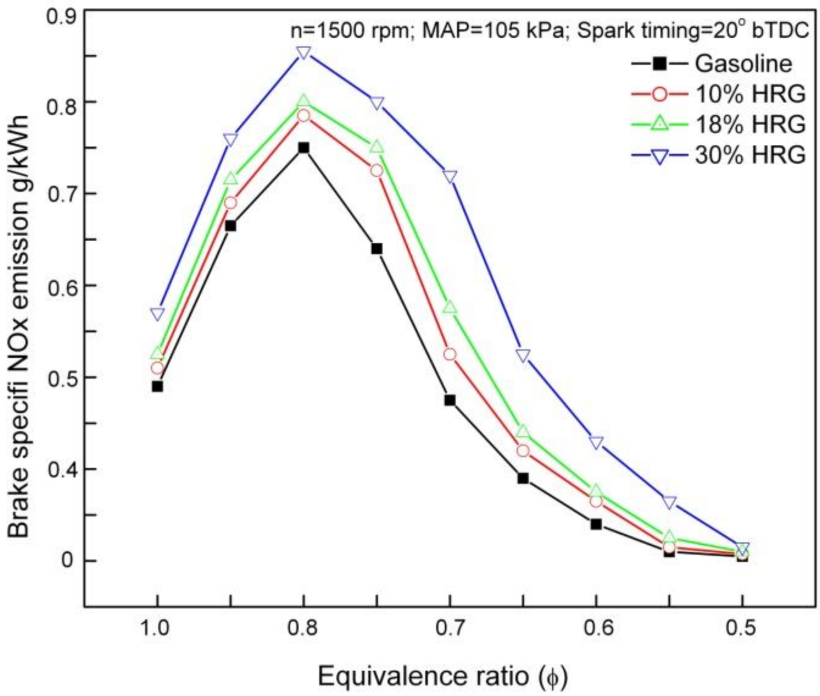

Figure 6. Brake Specific NO_x versus Equivalence Ratio at Fixed Spark Timing

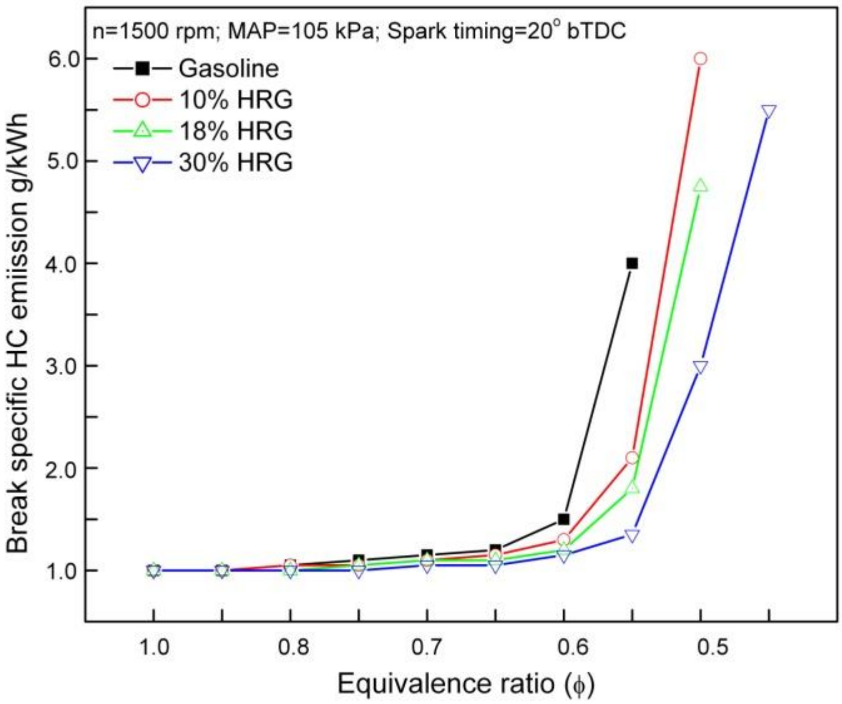

Figure 7. Brake Specific HC versus Equivalence Ratio at Fixed Spark Timing

The engine's indicated thermal efficiency can see from figure 9 that when equivalence ratio was under 0.7, HRG addition was not beneficial to efficiency improvement. Beyond equivalence ratio 0.7 engines's thermal efficiency exhibited small difference with addition of 10 % and 18 % HRG fraction from gasoline fuel operation. Marginal improvement in thermal efficiency 2 % was observed with 30% HRG addition. But the results were not as expected. Many researches [16–18] have showed that the fast burn speed of hydrogen could improve thermal efficiency. According to the emission analysis presented above, hydrogen addition could lower the emission of HC which meant better combustion efficiency. The reason for this was hydrogen's fast burn speed needs retarded spark timing to get best torque.

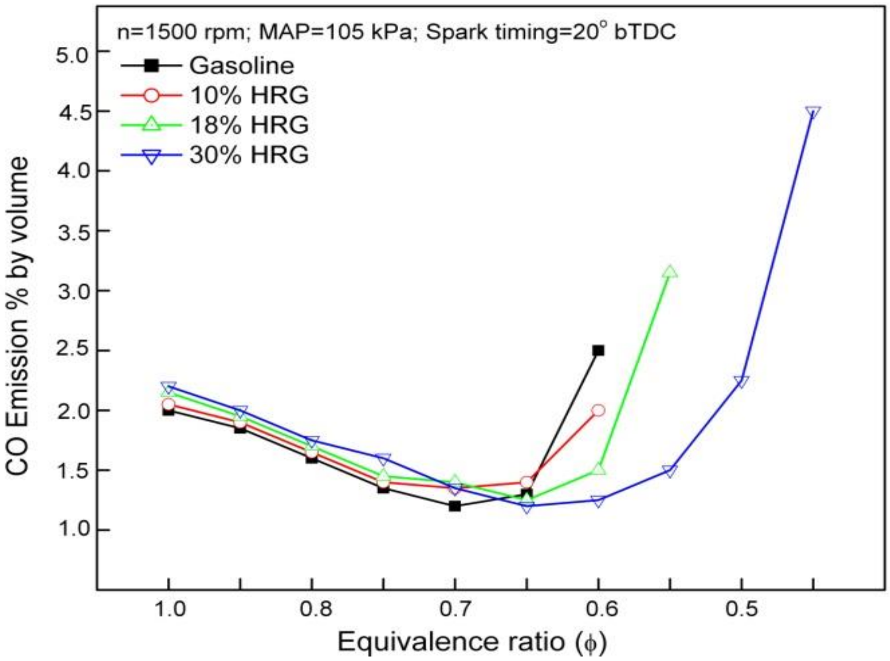

Figure 8. Brake Specific CO versus Equivalence Ratio at Fixed Spark Timing

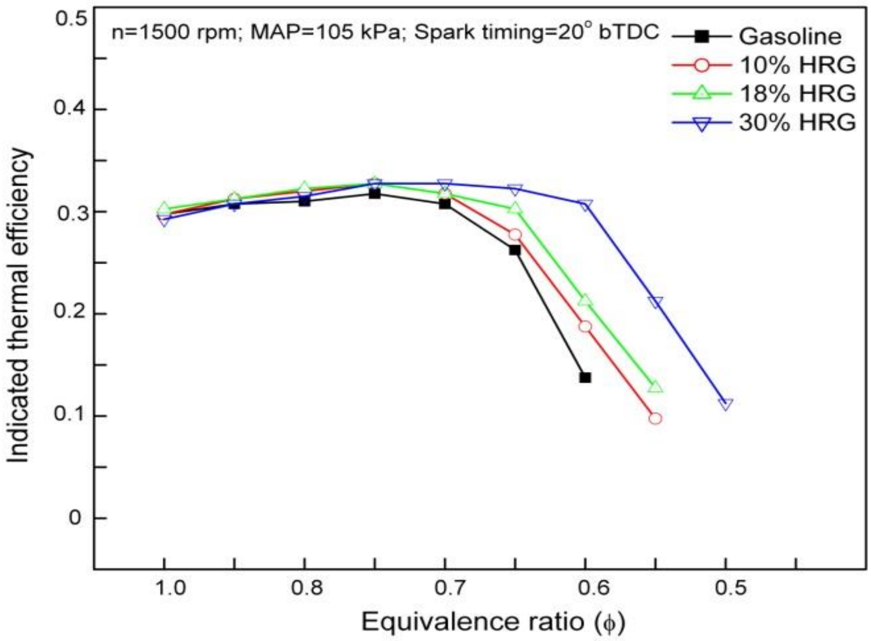

Figure 9. Indicated Thermal Efficiency versus Equivalence Ratio at Fixed Spark Timing

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Engine thermal efficiency and emission characteristics at MBT

Generally, the engine can reach its highest efficiency when maximum pressure occurs at 10–15° aTDC. So, spark timing should be retarded as HRG fraction increases. Retarding spark timing can not only reduce minus work in compression stroke but also decrease combustion temperature which is good for reducing NO_x emission.

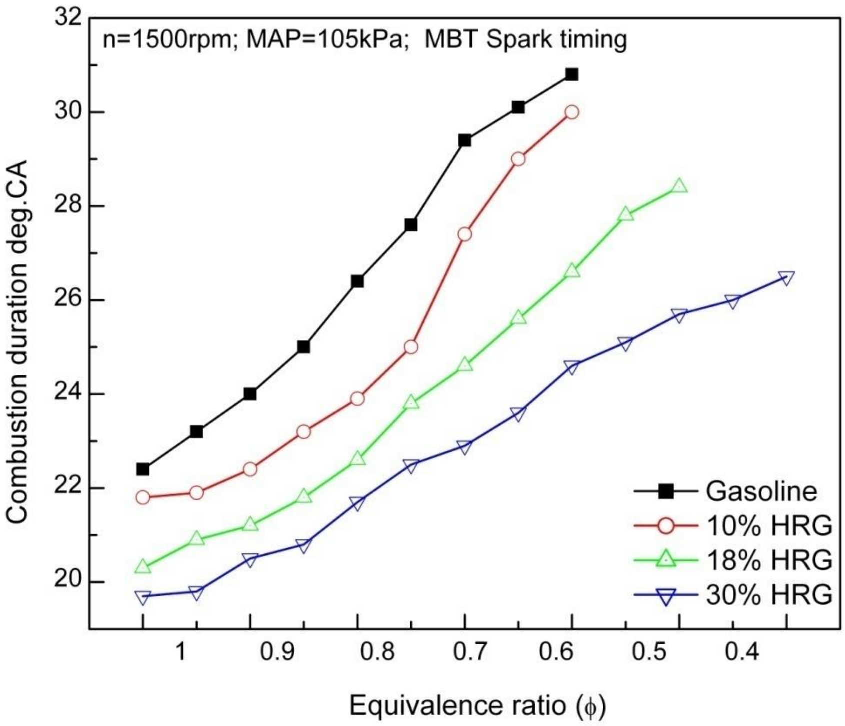

Figure 10. Combustion Duration versus Equivalence Ratio at MBT Spark Timing

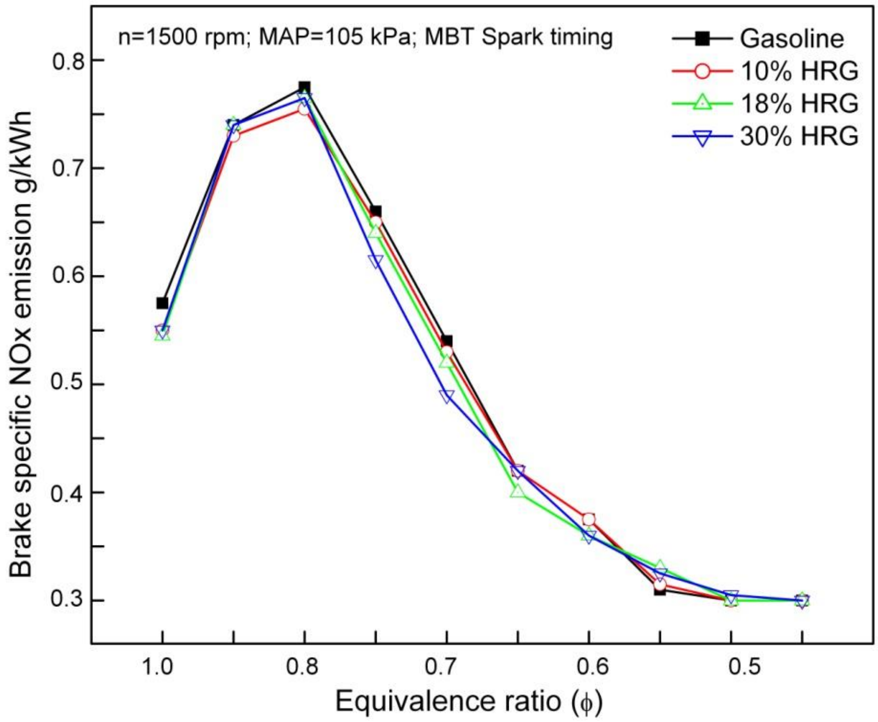

Figure 11. Brake Specific NO_x Emission versus Equivalence Ratio at MBT Spark Timing

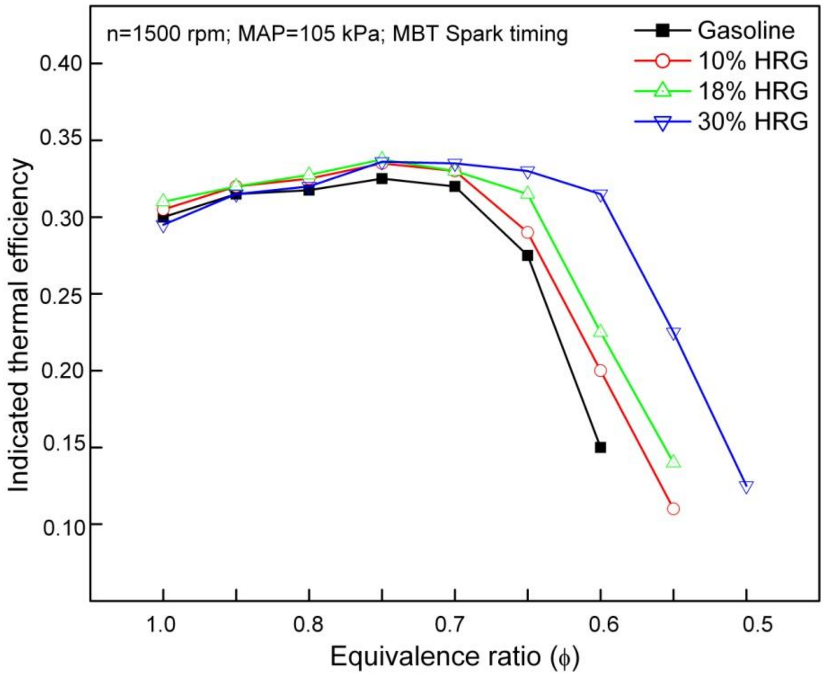

Figure 12. Indicated Thermal Efficiency versus Equivalence Ratio at MBT Spark timing

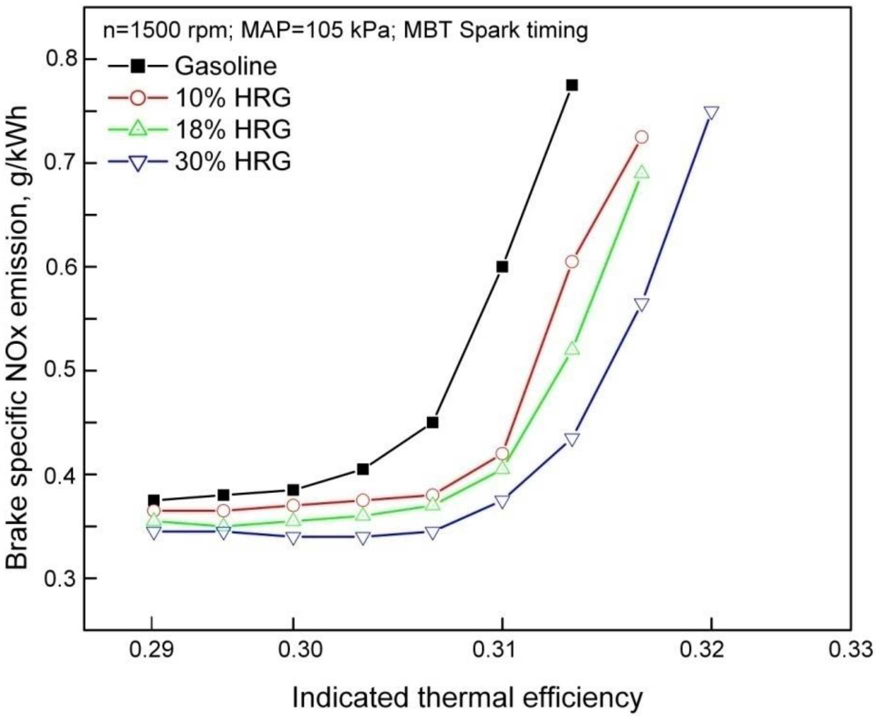

Figure 13. Brake Specific NO_x Emission versus Indicated Thermal Efficiency at MBT Spark Timing

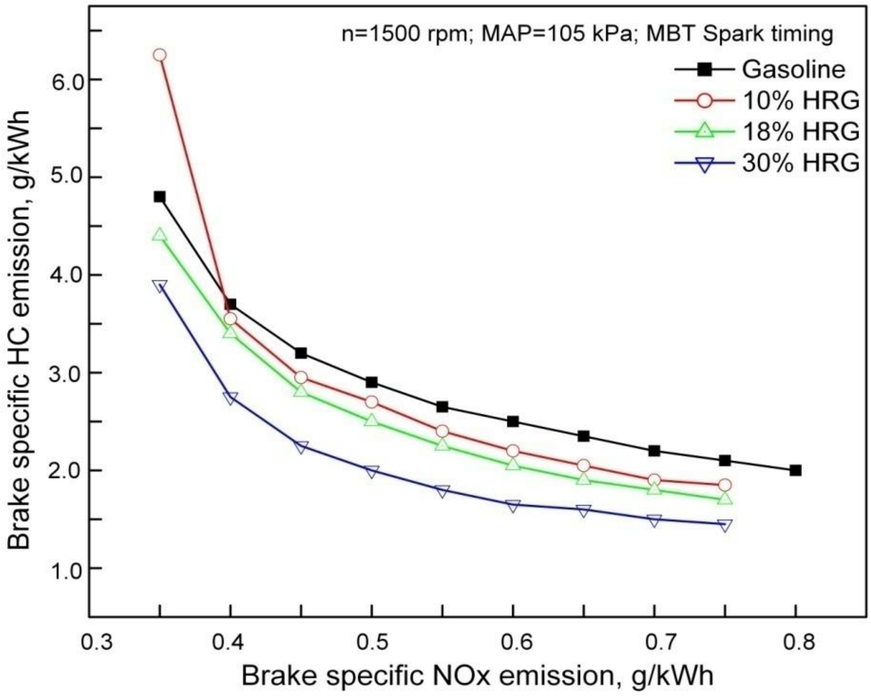

Figure 14. Brake Specific HC Emission versus Brake Specific NO_x Emission at MBT Spark Timing

It was decided to adjust spark timing to MBT based on experiments of a sweep of spark timing. The variation of MBT spark timing versus HRG fraction and equivalence ratio was plotted in figure 10. Obviously, MBT spark timing is dependent on flame speed, namely faster flame speed will result in a decrease in the crank angle before TDC at which the spark for maximum torque is applied. The flame speed could reach its maximum value at stoichiometric air fuel and will decrease if extra air is added, and this was confirmed in Figure 10 by the fact that minimum MBT spark timing was always located at equivalence ratio 1. Figure 10 also shows that MBT was retarded after HRG addition at a given lambda reflecting hydrogen's ability to speed up flame propagation.

Figure 11 shows that when spark timing was adjusted to MBT, NO_x emission after HRG addition was no longer obviously higher than that at gasoline operation. It was observed from figure 12 that HRG addition could improve thermal efficiency after spark timing optimization and learnt that more HRG fraction will yield more efficiency. An increasing thermal efficiency of 3 % was observed at 30 % HRG fraction.

Based on the results it was concluded that adding HRG supplementation with gasoline was good not only for reducing the exhaust NO_x emission but also for improving engine thermal efficiency. This effect was proved by referring the figure 13 show relations between indicated thermal efficiency (ITE) and NO_x emission. At given ITE, NO_x emission decreased as the increase of HRG fraction and at given NO_x emission, ITE increased as more HRG was added.

Among the various engine out emission, NO_x and HC have tradeoffs' relation. According to post process analysis, HRG addition should alleviate this contradiction which is confirmed in figure 14. Obviously, the curve moves further to the left bottom as more HRG was added which revealed the trade off between NO_x and HC emission was indeed alleviated by HRG addition.

Conclusions

An experimental study on thermal efficiency and emission of a GDI engine operating with HRG 10%, 18% and 30% in mass basis was conducted.

The main results are summarized as follows.

- Engine lean burn limit could be extended by HRG addition because hydrogen fuel has broader burn limit and fast burn speed. 10%, 18%, and 30% hydrogen rich gas fraction extended the lean limit equivalence ratio 0.62, 0.58, and 0.52, respectively, where 0.58 is the lower limit equivalence ratio of Gasoline.

- At a given equivalence ratio, combustion duration shortened as hydrogen fraction increased. The test results revealed that HRG addition would lead to higher NO_x emission at equivalence ratio 0.8 due to increasing fraction of cylinder contents being burnt gases close to stoichiometric during combustion.
- HC emission decreased with the increasing fraction of HRG addition upto equivalence ratio 0.6 due to higher cylinder temperatures making it easier to burn lean mixtures.
- CO emission decreased with increase of HRG fraction and approached lower value at equivalence ratio 0.6. Marginal improvement in thermal efficiency was observed with 30% HRG addition.
- After optimizing spark timing to MBT, engine efficiency rose with increase of hydrogen fraction. NO_x emission for fuel blends with different hydrogen fraction showed little difference at this MBT spark timing.
- Hydrogen addition was beneficial to the alleviation of the tradeoff relation between HC and NO_x emission.

Appendix

AFR – air fuel ratio
bTDC – before top dead centre
BDC – bottom dead center
CA – crank angle
CO – carbon monoxide
CR – compression ratio
GDI – gasoline direct injection
HC – hydrocarbon
HRG – hydrogen rich gas
MBT – maximum brake torque
NO_x – oxides of nitrogen
PC – personal computer
SI – spark ignition
SOI – start of injection
TDC – top dead centre

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