iJET EXPLORE International Journal of Engineering & Technology
CENTRE OF PROFESSIONAL RESEARCH PUBLICATIONS
ISSN 2049-3444

International Journal of Engineering and Technology Volume 4 No. 12, December, 2014

Performance Characteristics of Lean Burn Engine Fuelled with Hydrogen Rich Gas from Plasma Fuel Reformer

K. Kumaravel, C.G. Saravanan, B. Prem Anand, K. Anusha
Department of Mechanical Engineering, Faculty of Engineering Technology, Annamalai University,
Annamalainagar – 608 002, Tamilnadu, India.

ABSTRACT

In order to analyze the effect of hydrogen rich gas addition (HRG) on lean burn engine's thermal efficiency and emission, an experimental research was conducted on a gasoline direct injection (GDI) engine using definite fraction of HRG. The results showed that HRG addition could significantly extend the lean operation limit, improve the engine's lean burn ability, and decrease burn duration. HRG addition 10%, 18%, and 30% extended the lean limit equivalence ratio 0.62, 0.58, and 0.52 respectively. Combustion duration decreased as increasing fraction of HRG due to speed up flame propagation. HC emission decreased with the increasing fraction of HRG upto equivalence ratio 0.6 due to higher cylinder temperatures making it easier to burn lean mixtures. But thereafter HC emissions steadily increased despite of HRG addition as amount of fuel in regions was very lean to burn during primary combustion process. The same trend was observed in CO emission. However, nitrogen oxides (NOx) were found to increase with HRG addition because of hydrogen's high burn speed. Thermal efficiency increased 3% with addition of HRG at equivalence ratio 0.7. The experimental results showed that the maximum brake torque timing (MBT), air fuel ratio and HRG supplementation were implied significant influence on lean limit operation, burn durations, thermal efficiency and exhaust emission. It was concluded that taking advantage of hydrogen's high burn speed, obviously spark timing has to be retarded to MBT for substantial reduction of NOx emissions and further increase of engine's thermal efficiency.

Keywords: *Combustion duration, Equivalence ratio, GDI engine, Hydrogen rich gas supplementation, Lean burn engine, On-board generation, Plasma fuel reformer, Performance and Emission.*

1. 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 NOx 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.

2. EXPERIMENATL CONFIGURATON

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.1. 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.

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.

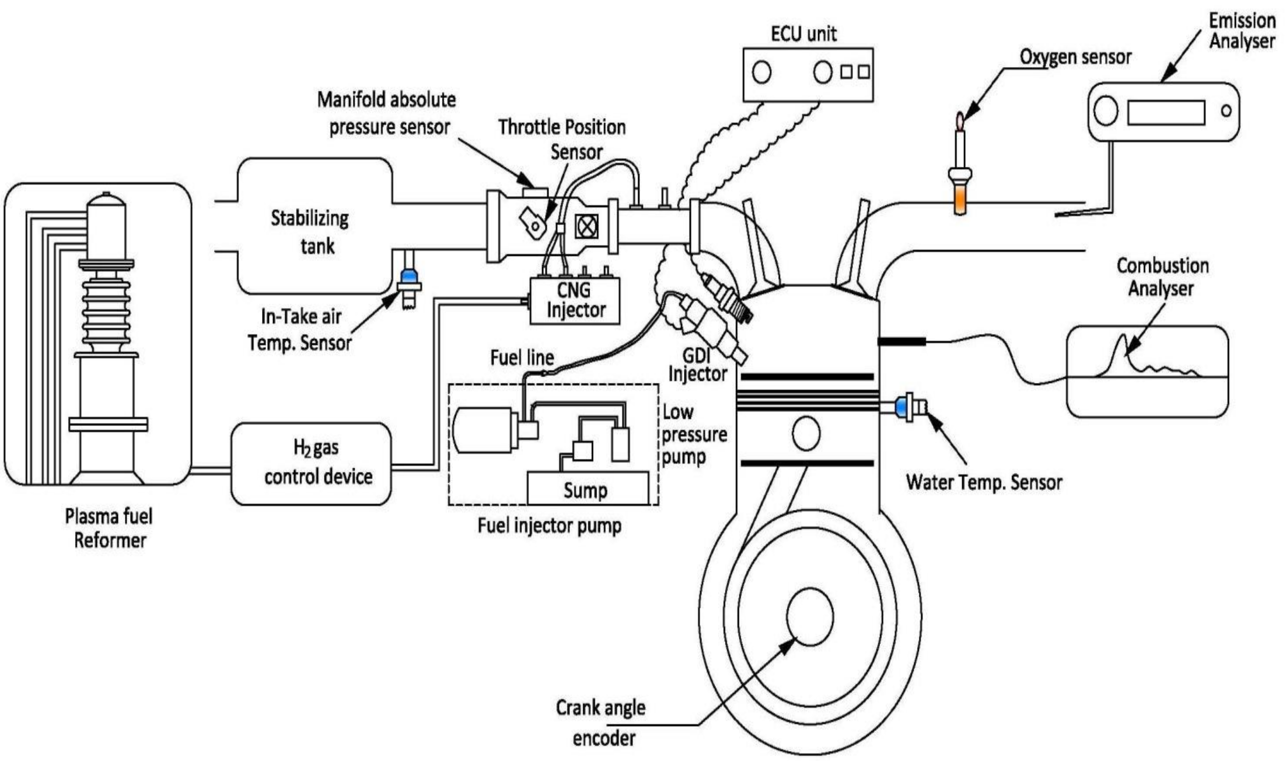

Fig.1 Experimental configuration of Lean Burn Engine

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	23.55m

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.

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.

Experimental Procedure

The engine was coupled to an eddycurrent 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, NOx, and CO, and were measured by AVL Di gas analyzer. For ensuring precise measurement a glass fiber 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 phase was to examine lean extension limit while the third one was to investigate engine's thermal efficiency and emission characteristics with effect of addition of HRG in definite fraction.

3. RESULTS AND DISCUSSIONS

Setting maximum brake toque (MBT) timing

Variations of spark timing relative to top centre affected the pressure development in the SI engine cylinder. If combustion starts too early in the cycle, the work transfer from the piston to the gases in the cylinder at the end of the compression stroke is too large; if combustion starts too late, the peak cylinder pressure is reduced and the expansion stroke work transfer from the gas to the piston decreases. There exists a particular spark timing which gives maximum engine torque at fixed speed and mixture composition and flow rate. The optimum timing which gives the maximum brake torque called maximum brake torque, or MBT timing - occurs when the magnitudes of these two opposing trends just offset each other. Timing which is advanced or retarded from this optimum gives lower torque [12]

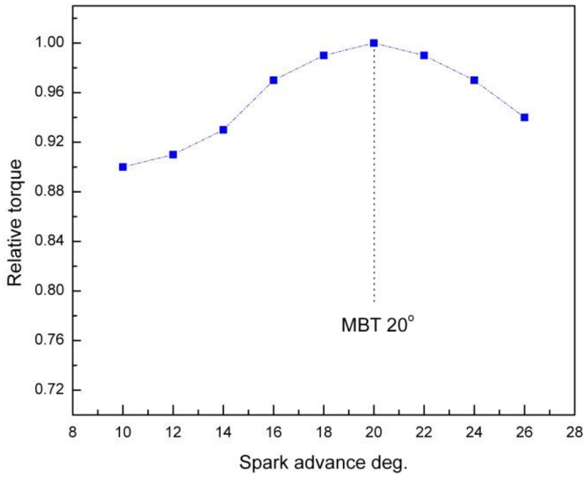

Fig.2 Relative torque Vs spark advance

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 Fig 2.

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 [13]. 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 [12].

Fig. 3 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. 2, lean limit got significantly ex-tended 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.

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.

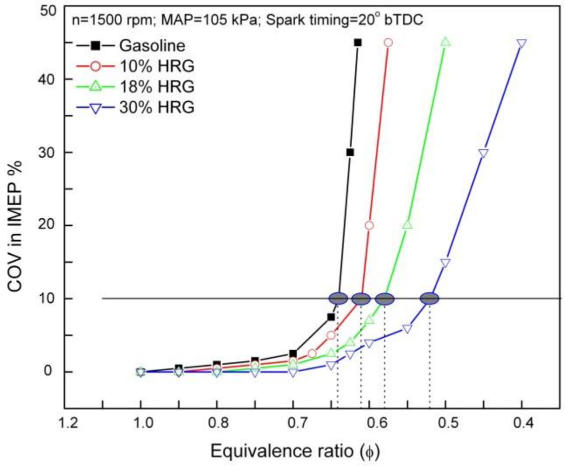

Fig.3 COV in IMEP Vs Equivalence ratio

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

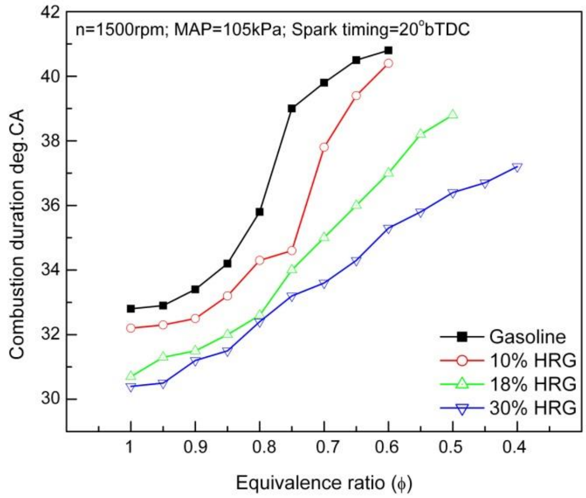

Fig.4 Combustion duration Vs Equivalence ratio

Fig. 4 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 MBT

Figs. 5–8 show the emissions of NOx, HC, CO and thermal efficiency as a function of equivalence ratio. From Fig. 5 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 NOx 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 NOx.

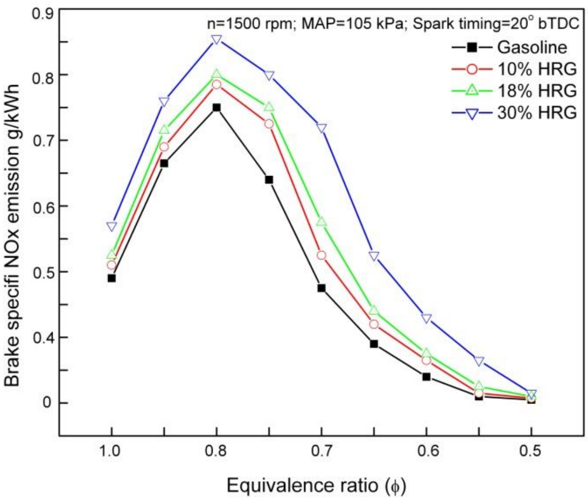

Fig.5 Brake specific NOx Vs Equivalence ratio

Trend of HC emission is shown in Fig. 6. 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. 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.

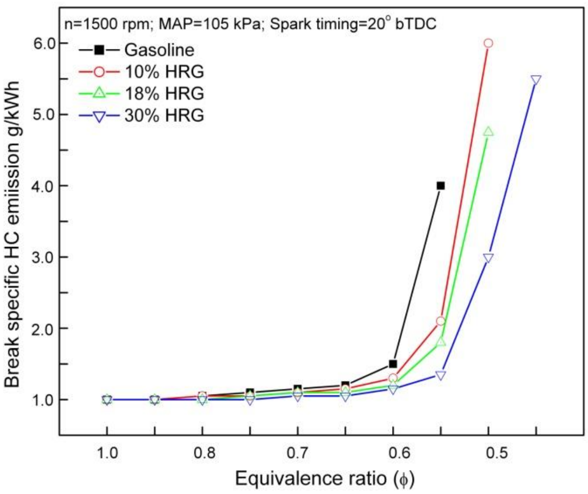

Fig.6 Brake specific HC Vs Equivalence ratio

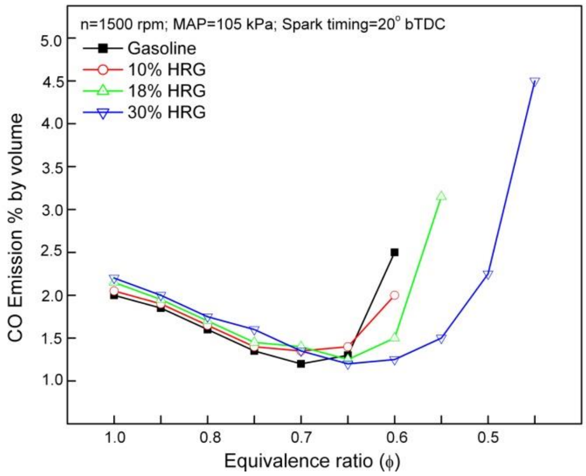

Fig.7 Brake specific CO Vs Equivalence ratio

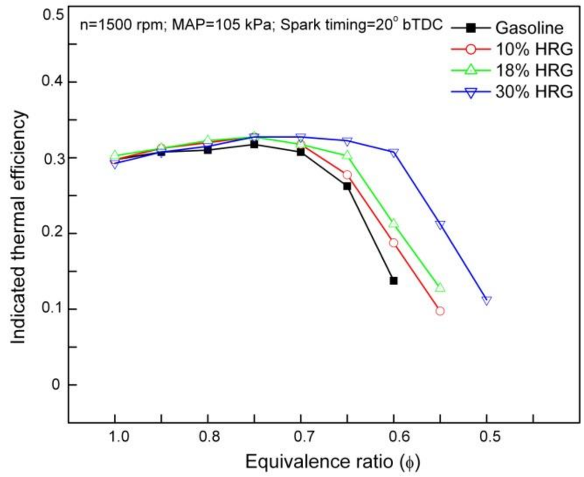

Fig.8 Indicated thermal efficiency Vs Equivalence ratio

The formation of CO is mainly due to incomplete combustion. As can be seen in Fig. 7, 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.

The engine's indicated thermal efficiency can see from Fig. 8 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 [7–11] have showed that the fast burn speed of hydrogen could improve thermal efficiency. Furthermore, 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.

4. CONCLUSION

The experimental results showed that the maximum brake torque timing (MBT), air fuel ratio and HRG supplementation were implied significant influence on lean limit operation, fuel economy, thermal efficiency and exhaust emission reductions. 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 NOx 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. Thermal efficiency increased 3% with addition of 30 % HRG at equivalence ratio 0.7 but this was considered as marginal improvement on engine's performance. From post process analysis it was concluded that, if spark timing was retarded to MBT, NOx emissions expected to be decreased and engine thermal efficiency further increased with the increase of HRG fraction.

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
NOx	–	oxides of nitrogen
PC	–	personal computer
SI	–	spark ignition
SOI	–	start of injection
TDC	–	top dead centre

Acknowledgement

The authors extend their heartfelt thanks to the authorities of the Ministry of New and Renewable Energy (MNRE), Hydrogen Energy and Fuel Cell Division, Government of India, New Delhi, for permitted them to utilize the facility of engine test system to this research work.

REFERENCES

[1] Y. Jamal and M.I. Wyszynski., "On-board generation of hydrogen-rich gaseous fuels-a review" International Journal of Hydrogen Energy, Volume 19, Issue 7, July 1994, PP. 557-572.

[2] D.R Cohn, A. Rabinovich, C.H. Titus and L.Bromberg., "Near-term possibilities for extremely low emission vehicles using onboard plasmatron generation of hydrogen" International Journal of Hydrogen Energy, Volume 22, Issue 7, July 1997, PP. 715-723.

[3] Bromberg, L., Cohn, D.R., and Rabinovich, A., "Plasma Reformer/Fuel cell systems for Decentralized power applications", International Journal of Hydrogen Energy, 1997, PP. 22-83

[4] Bromberg, L., Cohn, D.R., and Rabinovich. A., "Plasma Reforming of Methane", International Journal of Energy and Fuels" 1998, PP. 11-18.

[5] L. Bromberg and N. Alexeev., "Plasma Assisted Reforming of Methane: Two Stage Perfectly Stirred Reactor (PSR) Simulation" A release of Massachusetts Institute of Technology Plasma Science and Fusion Center. August 25, 2005.

[6] L. Bromberg, A. Rabinovich, N. Alexeev, and D.R. Cohn, "Plasma Reforming of Diesel fuel" Article presented at the American Chemical Society Meeting, Annaheim CA, March 1999.

[7] J.B. Green, Jr., N. Domingo, J.M.E. Storey, R.M. Wagner and J.S. Armfield, "Experimental Evaluation of SI Engine operation supplemented by Hydrogen rich gas from a compact plasma boosted reformer" SAE paper 2000-01-2206.

[8] Rong-Fang Horng, Yuh-Ping Chang and Shang-Chen Wu, "Investigation on the production of hydrogen rich gas in a plasma converter for motorcycle applications" International Journal of Energy conversion and Management, volume 47, Issue 15-16, September 2006, PP. 2155-2166.

[9] Rong-Fang Horng, Yuh-Ping Chang and Ming-Pin Lai, "A study of the hydrogen production from a small plasma converter" International journal of Fuel. Volume 86, Issues 1-2, January 2007, PP. 81-89.

[10] Varde KS. Combustion characteristics of small spark ignition engines using hydrogen supplemented fuel mixtures. SAE Paper No. 810921, 1981.

[11] Yamin J.A.A., Gupta H.N., Bansal B.B., Srivastava O.N., "Effect of combustion duration on the performance and emission characteristics of a spark ignition engine using hydrogen as fuel". International Journal of Hydrogen Energy, Volume 25, Issue 6, 1 June 2000, pp. 581 - 589.

[12] Heywood J., Internal Combustion Engine Fundamentals, 1st Edition, Tata McGraw-Hill, USA, 1988.

[13] Hoekstra R.L, Blarigan P.V, Mulligan N. NOx emission and efficiency of hydrogen, natural gas, and hydrogen/ natural gas blended fuels. SAE paper no. 961103; 1996.

[14] Quader A.A. Lean combustion and the misfire limit in spark ignition engines. SAE paper no. 741055; 1974.

