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AN EXPERIMENTAL STUDY ON PERFORMANCE AND CYCLIC VARIATIONS IN A SPARK IGNITION ENGINE FUELLED WITH HYDROGEN AND GASOLINE

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Abstract: This paper focuses on the effect of fuel-air ratio on the performance and cyclic variations of single cylinder hydrogen and gasoline fueled spark ignition engine. The hydrogen fueled engine was operated at the wide-open throttle (WOT) condition, while the engine fuelled with gasoline was operated with conventional carburetor mode by varying the throttle opening to vary equivalence ratio. All the experiments were conducted at a constant engine speed of 1600 rpm. The spark timing was set to minimum advance for best torque. As a result, it was found that there is about 29% reduction in the maximum power output of the engine when operating with hydrogen. The equivalence ratio could have been varied in the range of $0.3 < \phi < 0.9$ for hydrogen and of $0.8 < \phi < 1.3$ for gasoline. In all combustion periods, cyclic variations for hydrogen engine were much lower than that of gasoline engine. As a consequence, it was found that the engine gave similar output at equivalence ratio of $\phi = 1.1$ for gasoline and $\phi = 0.6$ for hydrogen. However the engine was operated more smoothly when fuelled with hydrogen.
Keywords: Hydrogen, Gasoline, Performance, Cyclic variations.

HİDROJEN VE BENZİN İLE ÇALIŞTIRILAN BUJİ ATEŞLEMELİ BİR MOTORDA PERFORMANS VE ÇEVRİMLER ARASI DEĞİŞİM ÜZERİNE BİR DENEYSEL ÇALIŞMA

Özet: Bu çalışmada yakıt olarak hidrojen ve benzinin kullanıldığı buji ateşlemeli bir motorda yakıt-hava karışım oranının motor performansı ve çevrimler arası değişime etkisi incelenmiştir. Motor hidrojen ile maksimum gaz kelebeği açıklığında çalıştırılmıştır. Motorun benzin ile çalıştırılması durumunda ise konvansiyonel karbüratör ve gaz kelebeği açıklık miktarı kullanılarak farkı yakıt-hava karışım oranları sağlanmıştır. Tüm deneyler motorun 1600 d/d sabit devrinde gerçekleştirilmiştir. Buji ateşlemesi maksimum motor torkunun üretildiği minimum avans değerinde gerçekleştirilmiştir. Motorun hidrojen ile çalıştırılması durumunda maksimum motor gücünde, benzine oranla yaklaşık % 29 oranında azalma tespit edilmiştir. Yakıt-hava karışım oranı, hidrojen için $0.3 < \phi < 0.9$ aralığında, benzin için ise $0.8 < \phi < 1.3$ aralığında değiştirilebilmiştir. Hidrojen motorunda tüm yanma periyotları için çevrimler arası fark benzin motorundan daha düşük olmuştur. Sonuç olarak benzin motorunun $\phi = 1.1$ yakıt-hava karışım oranı ile hidrojen motorunun $\phi = 0.6$ yakıt-hava karışım oranında yaklaşık olarak eşit motor performansı elde edilmiştir. Fakat motorun hidrojenle daha stabil çalıştığı görülmüştür.
Anahtar Kelimeler: Hidrojen, Benzin, Performans, Çevrimler arası değişim.

INTRODUCTION

The great rise in pollution levels in the atmosphere and increased concern for energy independency are the two major driving causes for investigating alternative fuels (Ganesh et al., 2008). With increasing concern about energy shortage and environmental protection, research on improving engine fuel economy and reducing exhaust emissions has been among the most important research topic in combustion and engine development. Alternative fuels commonly useful to clean fuels compared to diesel and gasoline in the engine combustion process (Rousseau et al., 1999; Ben et al. 1999). Among the viable options, hydrogen is the only non-carbonaceous fuel obtainable on the earth (Ganesh et al., 2008). Different from traditional fossil fuels, the combustion of hydrogen generates no carbon-related emissions such as HC, CO and CO_2 (Heywood, 1988). Hydrogen has been regarded as an alternative fuel for power systems because of CO_2 and hydrocarbon free operation. Recent strong increase in the price of petroleum, fast increase in emission of green house gases and very strict environmental legislations are important motivating factors for usage of hydrogen in fuel cells and internal combustion engines (Ganesh et al., 2008). Hydrogen can be obtained from sources as natural gas, coal, biomass and water. The potential role of hydrogen in global warming is insignificant in comparison to hydrocarbon-based fuels (Salimi et al., 2009). Hydrogen has incomparable combustion characteristics that are different from those of hydrocarbon fuels and needs some special considerations where it is employed to an engine. High flame speed and wide flammability limit of hydrogen–air mixture procures running engine on wide open throttle while controlling load with changing equivalence ratio. Fast burning characteristics of hydrogen also permit high speed engine operation (Kahraman et al., 2007). This would increase engine efficiency due to minimization of pumping losses, but the challenge is that for mid to high loads of engine operation near-stoichiometric mixtures lead to strong increase of NO_x emission. Although the auto-ignition temperature of hydrogen–air mixture is very high compared to hydrocarbon–air mixtures, ignition energies of hydrogen–air mixtures are high than that of hydrocarbon–air mixtures. Therefore pre-ignition is one of the important restrictions for engine running at near-stoichiometric mixtures. In gasoline SI engines the power is limited due to only knock occurrence, but in hydrogen-fueled SI engines the pre-ignition could also restrict power (Tang et al., 2002). The high diffusion speed of hydrogen symbolizes the capability of hydrogen on improving the mixing of fossil fuels and air. However, due to the high adiabatic flame temperature and low energy density of hydrogen on volume basis, compared with gasoline engines, hydrogen-fueled engines always encounter the increased NO_x level and reduced engine power (Al-Janabi and Al-Baghdadi, 1999; Ganesh et al., 2008). Hydrogen is very light, the energy density of hydrogen on volume basis is only 10.8 MJ/m^3, which may lead to a reduced power output for hydrogen engines at stoichiometric mixture, compared with gasoline engines (Ji and Wang, 2009). At the same time, the lack of hydrogen fueling infrastructure and the high cost of hydrogen on production and storage are also restricts for the wide application of hydrogen-fueled engines in the near ahead (Ji et al., 2010). The properties of hydrogen along with gasoline are listed in Table 1.

Hydrogen induction techniques play a very important role in determining the performance characteristics of the hydrogen fueled internal combustion engine (Suwanchotchoung, 2003). Research on the hydrogen-fueled SI engine has generally been focused on pre-mixed charge and most of these studies have been maintained by the automotive industry (Tsujimura et al., 2003; White et al., 2006). Hydrogen can be taken in the intake manifold either by continuous or timed injection. The former method produces unexpected combustion problems, less controllable (Das, 2002). But the latter method, timed port fuel injection is a good candidate and extensive studies indicated the ability of its adoption (Das, 1990; Das et al., 2000).

Table 1. Properties H_2 and gasoline (Ji and Wang, 2009).

Properties	H_2	Gasoline
Molecular weight (g/mol)	2.015	110
Stoichiometric fuel to air ratio (F/A)	34.3	14.6
Minimum ignition energy (mJ)	0.02	0.24
Ignition temperature (K)	858	530
Adiabatic flame temperature (K)	2384	2270
Flame speed at 20 ^oC (cm/s)	237	41.5
Limits of flammability (vol % in air)	4.1-75	1.5-7.6
Quenching gap (cm)	0.06	0.2
Lower heating value (MJ/kg)	120	44
Diffusion coefficient at stoichiometric (cm^2/s)	0.61	0.05

External mixture formation through fuel injection has been demonstrated to result in higher engine performances, extended lean operation, lower cycle-to-cycle variation and lower NO_x emission (Yi et al., 2000; Rottengruber et al., 2004, Kim et al., 2006). This is the result of the higher mixture homogeneity because of longer mixing times for port fuel injection. External mixture formation provides a greater degree of freedom concerning storage methods (Verhelst et al., 2006). The most important problem with port fuel injection is the high possibility of pre-ignition and backfires, especially with rich mixtures (Kabat et al., 2002; Ganesh et al., 2008). A port fuel injection hydrogen engine operating stoichiometric mixture at WOT, has a theoretical power deficit of about 15% compared to a gasoline engine, because of the lower volumetric efficiency (Das, 1990).

Normally, cycle-to-cycle variations occur in the engines operating with very lean mixtures. But, with hydrogen, these variations are much less compared to that of engines powered by other hydrocarbon fuels (Kim at al., 1995). Hydrogen gas is characterized by a rapid combustion speed, wide combustible limit and low minimum ignition energy. Such characteristics play a role to decrease engine cycle variation for the safety of combustion (Varde et al; 1985; Kim at al., 1995).

EXPERIMENTAL SETUP AND METHOD

The engine was modified to be fueled with hydrogen injected into the intake port. Hydrogen was injected into the intake manifold at a pressure of about 5 bar. The engine fuelled with hydrogen was operated at wide-open throttle. Baseline tests were conducted with conventional carburetor mode by varying the throttle opening to vary equivalence ratio. The ignition timing was adjusted to a minimum advance for best torque (MBT) for all engine operating conditions. The engine was tested at different equivalence ratios for gasoline and hydrogen fuels. The hydrogen and gasoline was compared in terms of combustion, cyclic variation and engine performance. Figure 1 shows the schematic diagram of the experimental setup used in this work. The specifications of the engine are given in Table 2.

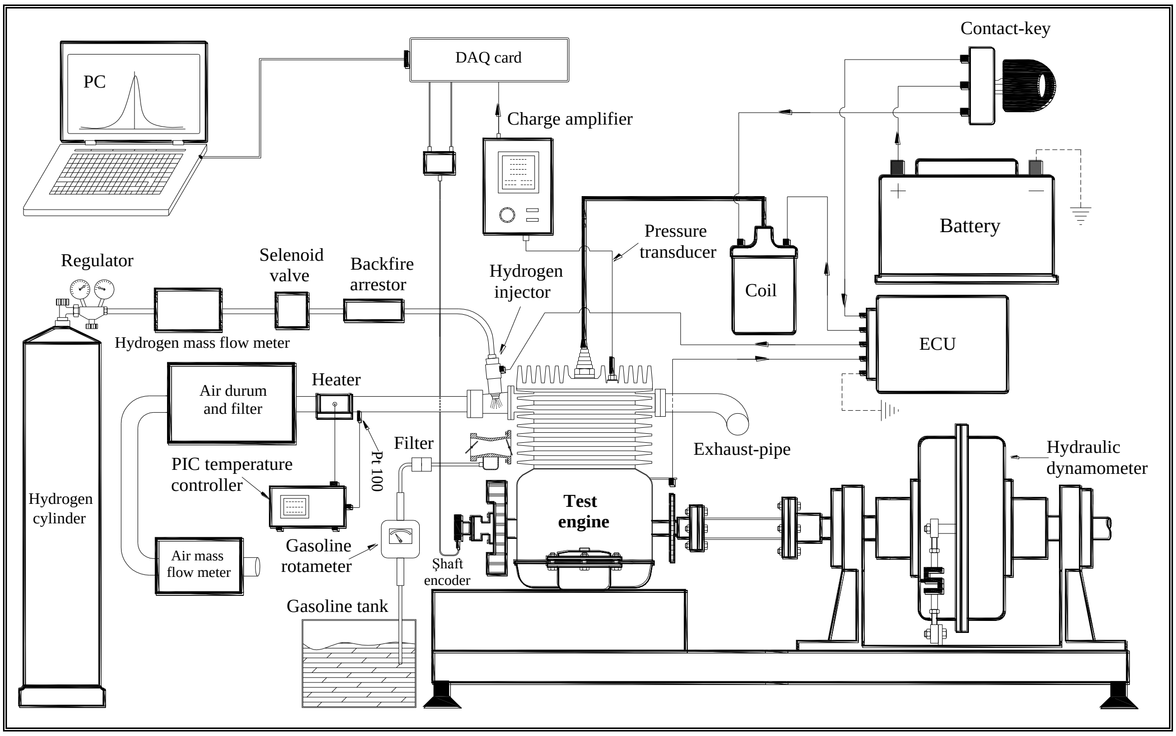

Figure 1. Schematic diagram of experimental setup.

Table2. Engine specifications.

Engine Specifications	
Engine type	Four stroke, air-cooled, single cylinder, L-head, Modified SI engine
Fuel	Hydrogen, gasoline
Max. rpm	3600 d/d
Rated power	12 hp
Max. torque	≈ 25 Nm
Bore x Stroke	85.7 x 82.6
Displacement volume	476.5 cm³
Compression rate	7.6

A shaft encoder (Kistler, 2618B) was fitted to the crankshaft to measure engine speed and crank position. The shaft encoder produces 1800 pulses per revolution (a signal at very 0.2^o CA) as well as top dead center (TDC) pulse. The cylinder pressure was measured using a piezoelectric pressure transducer (Kistler, 6052C). A Motec M4 engine control unit (ECU) was used to control spark timing and H_2 injection duration and timing. The use of a reference wheel and an inductive sensor fitted to the camshaft allowed the detection of both engine speed and cycle position. A Bosch ignition module (0227 100 124) was connected to a Bosch MEC 717 coil supplying energy to the spark plug. A conventional spark-plug was used as ignition source. Hydrogen flow rate was measured by a thermal mass flow meter (Aalborg, GFC67). The air flow rate was measured using a thermal mass flow meter (Aalborg, GFM77). The engine was fitted with an injection system allowing hydrogen to mix with the inlet air. Hydrogen was stored in a gas cylinder at a pressure of around 200 bar. A pressure regulator was used to feed the injector with hydrogen at a pressure of 5 bar. Hydrogen was fed into the system at an ambient temperature of approximately 20 ^{o}C. For manifold injection of hydrogen, a solenoid operated gas injector (Bosch NGI2) was used. It was operated with 12V DC power supply. Hydrogen was injected into the manifold after the start of the intake stroke (300^o CA before TDC). To stop flame travelling up the fuel line, a flashback arrester (Wittgas, RF53) was connected to the fuel supply line. When the engine is running, the hydrogen in the combustion chamber may leak through the gap between the piston and liner into the crankcase. Continuous accumulation of hydrogen in the crankcase may lead to crank case explosion. Therefore the crankcase was ventilated to the atmosphere. Hydrogen is a colorless and odorless gas, which makes it difficult to detect if it leaks. A portable hydrogen leak sensor was used to detect the leakage during the experiments. Stainless-steel pipes and leak proof fusion joints were used in the fuel supply line, which can withstand up to a pressure of about 150 bar. The air inlet temperature was controlled (30 ^{o}C) using an 800 W electric heater and a PIC temperature controller. The exhaust gas temperature and engine oil temperature were measured via K-type thermocouples coupled to a temperature indicator. A rotameter used to measure fuel volume flow rate of gasoline. All the experiments were conducted at a constant engine speed of 1600 rpm and a constant inlet air temperature of 30 °C. The engine was coupled to a hydraulic dynamometer for loading. A data acquisition system (Measurement Computing-USB-1616HS-4) with a 16-bit resolution and 1 MHz sampling frequency was used to acquire and process crank angle sensor and pressure transducer signals. The in-cylinder pressure data was recorded by PC controlled data acquisition system along with crank angles for 100 consecutive cycles. A FORTRAN program was used to process pressure data. The data captured was then used to calculate ‘‘real time’’ performance parameters such as engine indicated mean effective pressure(IMEP), indicated thermal efficiency, indicated power, indicated specific fuel consumption and mass fraction burned. The equations used for the calculations are given below.

Rassweiler and Withrow method (Rassweiler and Withrow, 1938) was used for estimating the mass fraction burned (mfb) profile from in-cylinder pressure and volume data. In this method, the mass fraction burned is given by;

$$\text{mfb} = \frac{\sum_{i=ign}^{i=0} \Delta P_C}{\sum_{i=ign}^{i=N} \Delta P_C} \tag{1}$$

In this study, Eq. (2) used to estimate mass fraction burned from only pressure rise due to combustion was obtained by subtracting pressure rise due to engine compression from pressure rise due to combustion (Buran,1998).

$$\text{mfb} = \frac{\sum_{i=ign}^{i=0} \Delta (P_C - P_m)}{\sum_{i=ign}^{i=N} \Delta (P_C - P_m)} \tag{2}$$

Where, N is the total number of crank angle intervals, ΔP_C is the pressure rise due to the combustion, ΔP_m is the pressure rise due to the motoring, i the integer crank angle location, ign the ignition crank angle location.

Cyclic variations can be evaluated from the pressure data. The most frequently used parameter in evaluating engine cyclic variations is the coefficient of variations of indicated mean effective pressure (COV_{IMEP}) (Heywood, 1988). COV_{IMEP} is defined as,

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{IMEP_{mean}} \times 100 \tag{3}$$

Where, COV_{IMEP} is the coefficient of variation of imep for N consecutive cycles, $IMEP_{mean}$ the mean value of IMEP for N cycles, σ_{IMEP} the standard deviation of imep for N cycles.

RESULTS AND DISCUSSION

Engine Performance Parameters

The variation of IMEP and indicated power output with the equivalence ratio is shown in Figure 2 for both hydrogen and gasoline engine. From Figure 2, it can be observed that hydrogen engine produces approximately 29% less maximum indicated power than that of gasoline engine. Hydrogen being a low-density gas displaces a corresponding amount of air by volume causing the mixture quantity to reduce. This resulted in a drop in the indicated power. Indicated power increases linearly when equivalence ratio increased from 0.3 to 0.75. After this equivalence ratio, indicated power increases gradually. It seems to be that the engine power will start to drop after an equivalence ratio of ϕ=0.9 or ϕ=1.0. This early fall may be related to the drop in volumetric efficiency of the engine with increasing equivalence ratio.

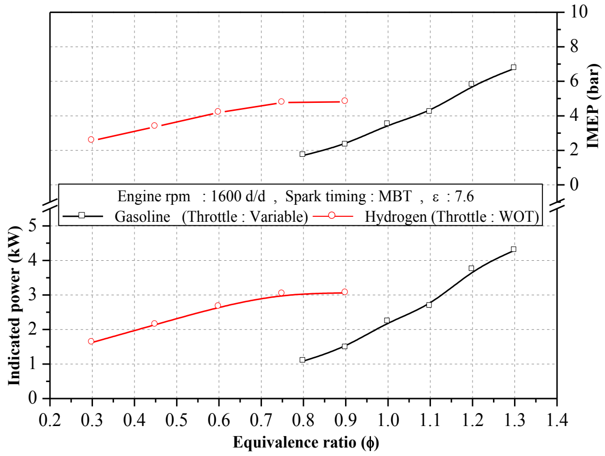

Figure 2. Variation of IMEP and indicated power with equivalence ratio for gasoline and hydrogen engine.

Figure 3 presents the variation of indicated thermal efficiency with equivalence ratio for both hydrogen and gasoline engine. It is clearly seen that operation on hydrogen is generally more efficient than operation on gasoline at given engine conditions. It can be observed that hydrogen engine produces approximately 10% more maximum indicated thermal efficiency than that of gasoline engine. This is due to high combustion speed of hydrogen compared to that of gasoline.

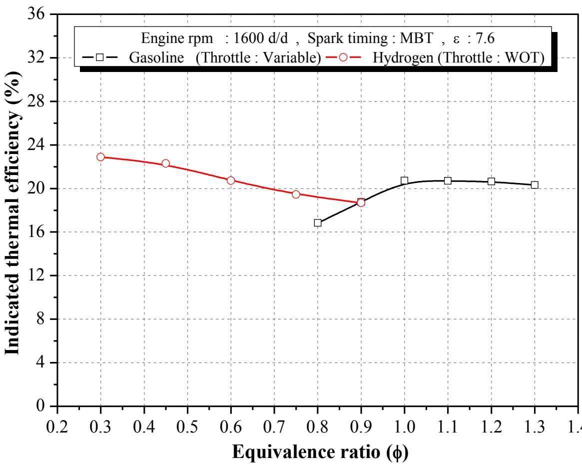

Figure 3. Variation of indicated thermal efficiency with equivalence ratio for gasoline and hydrogen engine.

Figure 4 illustrates the variation of indicated specific fuel consumption versus equivalence ratio for both hydrogen and gasoline engine. It is observed that indicated specific fuel consumption is lower for hydrogen engine than that of gasoline engine at all equivalence ratios. For the engines with similar output, specific fuel consumption is approximately 75% less for hydrogen engine compared to gasoline engine.

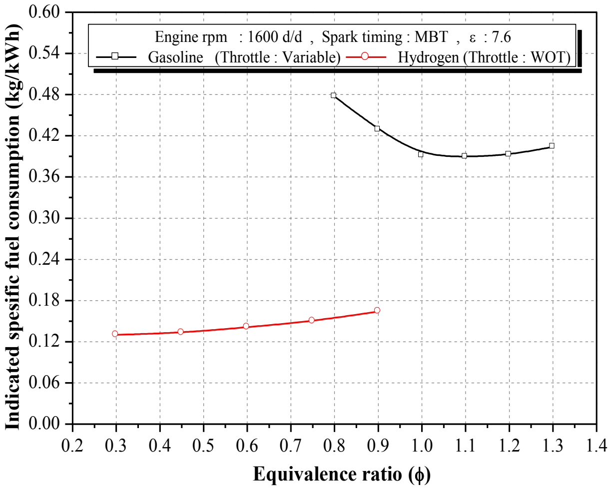

Figure 4. Variation of indicated specific fuel consumption with equivalence ratio for gasoline and hydrogen engine.

The variation of volumetric efficiency with equivalence ratio for both gasoline and hydrogen engine is shown in Figure 5. The volumetric efficiency dropped due to pumping losses in gasoline engine. However it tended to rise with increasing throttle angle. In the hydrogen engine, initially the volumetric efficiency was high due to WOT. Because hydrogen displaces air, the volumetric efficiency drops with increasing equivalence ratio. Similar results related to the degree of volumetric efficiency for hydrogen engine can be found in literature (Subramanian et al., 2007).

It could be concluded that hydrogen engine is able to produce volumetric efficiencies higher than conventional gasoline engine, when operating engine in WOT at lean mixtures.

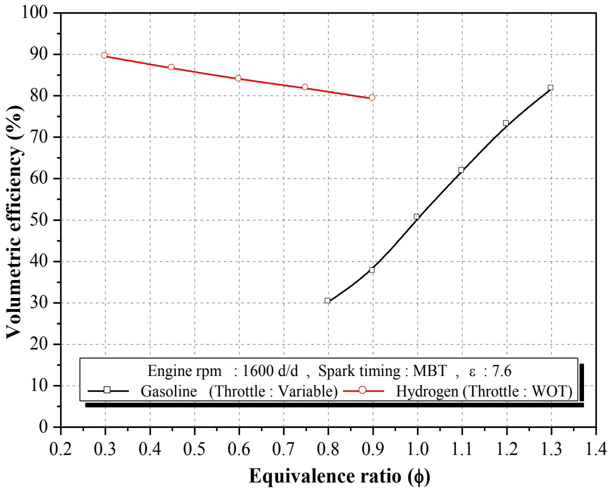

Figure 5. Volumetric efficiency versus equivalence ratio for gasoline and hydrogen engine.

Figure 6 shows the variation of maximum brake torque (MBT) ignition timing with equivalence ratio for gasoline and hydrogen engine. It is observed from Figure 6 that the MBT ignition timings for hydrogen at given equivalence ratios were more retarded than those of gasoline engine. This resulted in higher burning and heat release rates for hydrogen.

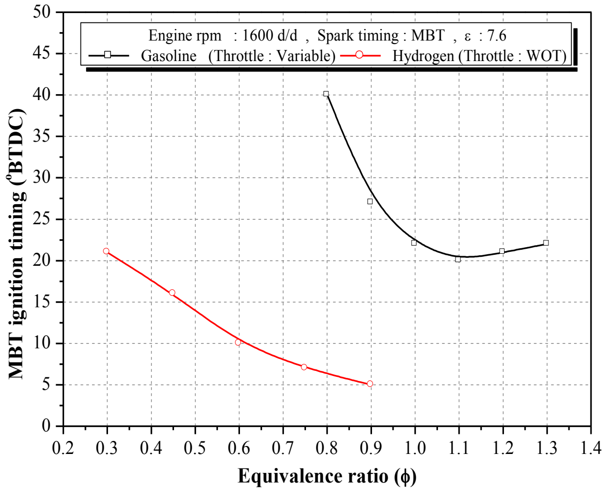

Figure 6. Variation of MBT ignition timing with equivalence ratio for gasoline and hydrogen engine.

Shown in Figure 7 is the variation of total combustion duration with equivalence ratio for gasoline and hydrogen engine. It is observed that the total combustion duration reduced with increasing equivalence ratio for both fuels. The total combustion duration for hydrogen engine is generally lower than that of gasoline engine at given engine conditions.

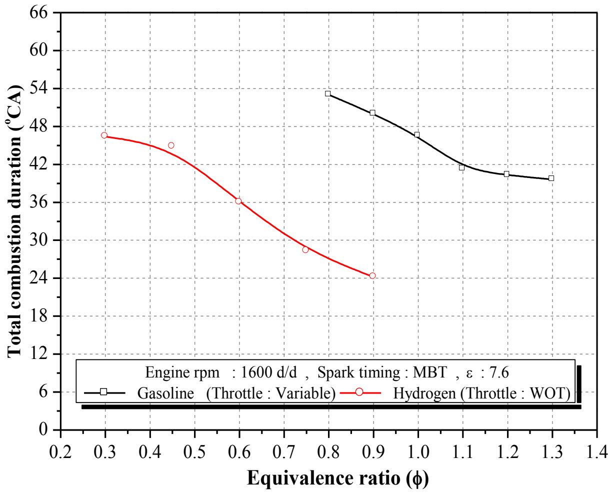

Figure 7. Variation of total combustion duration with equivalence ratio.

Figure 8 illustrates the variation of (a) oxygen concentration in the exhaust gas and (b) temperature of exhaust gas according to equivalence ratio for both gasoline and hydrogen. It is noticed that oxygen concentration in the exhaust gas is gradually reduced by increasing equivalence ratio for gasoline engine in contrast for hydrogen engine. The hydrogen engine was operated at WOT for given equivalence ratios. So, the hydrogen concentration increases with equivalence ratio resulting in corresponding amount of air displaces and hence oxygen concentration drops down rapidly. Figure 8(b) shows the temperature of exhaust gas increased with increasing equivalence ratio for both gasoline and hydrogen. The exhaust gas temperature for hydrogen engine is higher than that for gasoline engine due to high temperature of hydrogen flame.

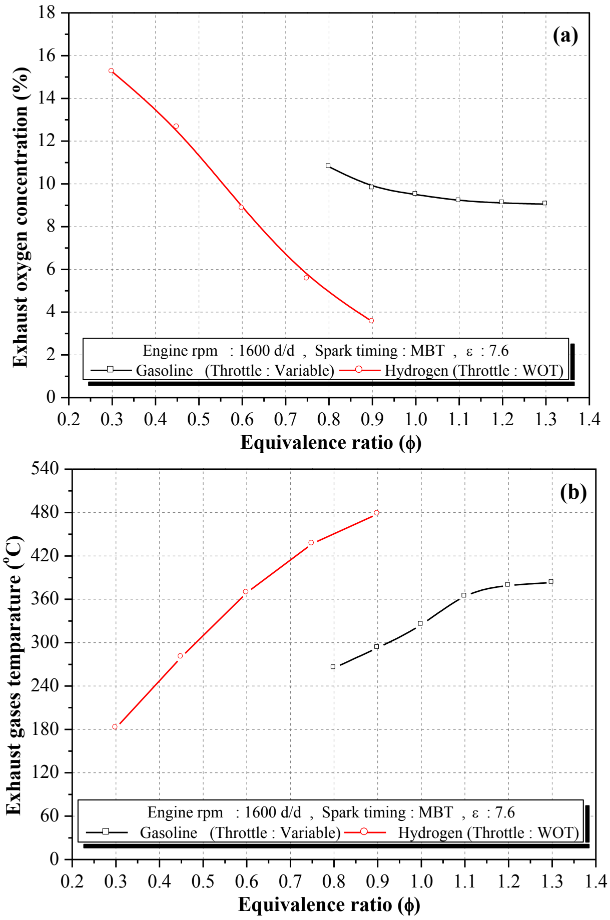

Figure 8. Variation of exhaust gas (a) oxygen concentration and (b) temperature versus equivalence ratio.

Cyclic Variation

Cyclic variation is an important factor for evaluating engine performance. One important measure of cyclic variability, derived form pressure data, is the coefficient of variation in indicated mean effective pressure (COV_{IMEP}) (Heywood, 1988).

In this study, cyclic variations for gasoline and hydrogen engines was investigated for equivalence ratios generating approximately equal engine performance (ϕ = 1.1 for gasoline, ϕ = 0.6 for hydrogen).

Figure 9 gives the variation of peak pressure (P_{max}) of consecutive 100 cycles versus position of peak pressure, φ_{Pmax}, (crank angle after TDC) at different equivalence rates for gasoline and hydrogen engine. In other words, the shape of frequency distributions in P_{max} - φ_{Pmax} data illustrates whether the combustion process is fast due to rich mixtures or slow due to lean mixtures. In general, the data extended to the upper left of the figure (fast burn line) belongs to near stoichiometric cycles and the data extended to the lower left (slow burn line) are formed by lean cycles. When combustion process is much slower, the cyclic variability becomes large and the distribution becomes skewed toward slower burning cycles which have low P_{max}. It is also noticed that the data of gasoline combustion with lean mixtures is close to slow-burn line and having wider data band, while the data of hydrogen combustion is close to fast-burn line with narrow data band.

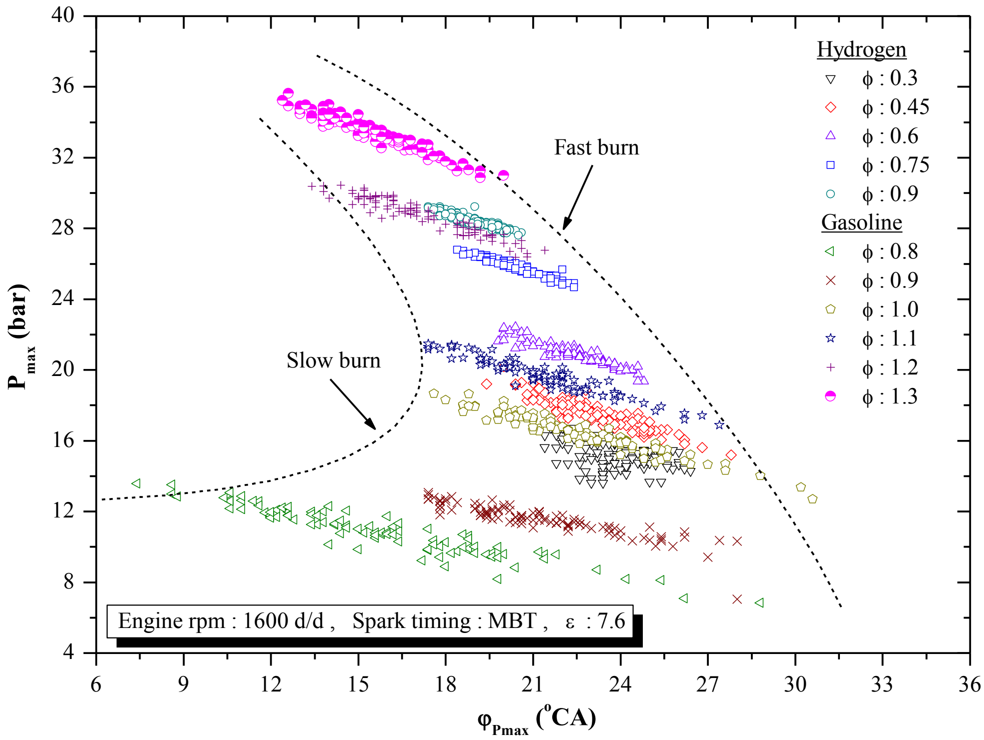

Figure 9. Individual-cycle P_{max} versus φ_{Pmax} for hydrogen and gasoline engine.

Shown in Figure 10 is the variation of COV_{IMEP} with equivalence ratio for gasoline and hydrogen engine. In this figure, the values of COV_{IMEP} are lower for hydrogen engine than that for gasoline engine at lean mixtures. This is because of the wide flammability limit and fast burning rate of hydrogen. The observed minimum value of COV_{IMEP} is around 0.85% for hydrogen engine at equivalence ratio of 0.75, while it is around % 1.3 for gasoline engine at ϕ = 1.3. From this figure it could be argued that hydrogen engine produces more stabile engine operation even with lean mixtures than gasoline engine operation with rich mixtures.

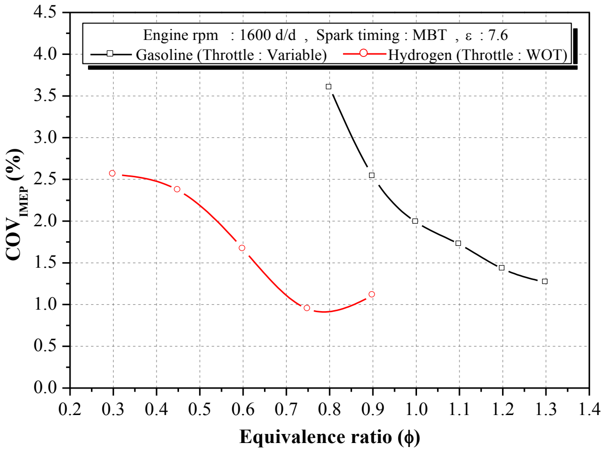

Figure 10. Variation of COV_{IMEP} with equivalence ratio for gasoline and hydrogen engine.

Figure 11 shows the variation of mass fraction burned (mfb) curves which are formed by calculating the mean of 100 consecutive cycles at selected equivalence ratios for both engines. It is clearly seen that the mixture burning rate is strongly influenced by the type of fuel. It was found that there is about 15% reduction in the combustion duration when the engine operates with hydrogen according to gasoline operation due to high combustion speed of hydrogen.

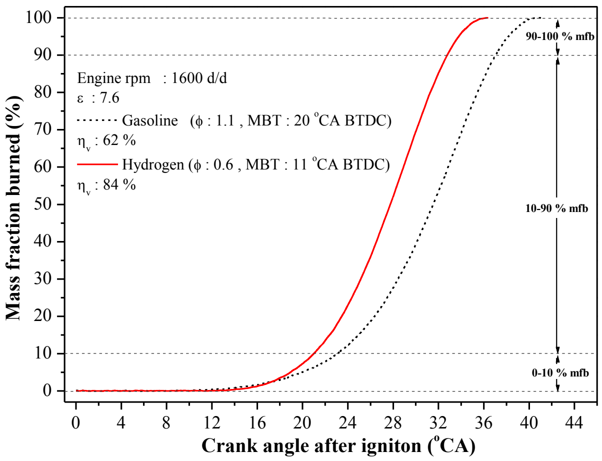

Figure 11. Variation of mass fraction burned at selected equivalence ratios for both engine.

Using the coefficient of variation in the combustion period as defined above, the coefficient of variation in the early combustion period and the coefficient of variation in the rapid combustion period were found. The early combustion period is defined as the period until the mass burning rate of 10% for "flame development angle", and the rapid burning angle as the period from 10 % mass fraction burn to the mass burning rate of 90% for "rapid burning angle" (Heywood, 1988).

"The last burning angle" is defined as the period from 90% mass fraction burn to the mass burning rate of 100%. Combustion duration for 0-10% mfb, 10-90% mfb, 90-100% mfb and total burning angle periods (sum of combustion durations for 0-10% mfb, 10-90% mfb, 90-100% mfb periods) are shown in Figure 12 for 100 consecutive cycles.

Fluctuations from the mean of combustion duration drawn for each period present alternatively the cyclic variation for burning angles of 100 consecutive cycles. In addition, these figures illustrate the effect of cyclic variation intensity in each period on the intensity of fluctuations in the total burning angle. It is argued that the cyclic variations in flame development angle which is the slowest period for both fuels were found to be higher than those in other two periods. It is also argued that the cyclic variations in all periods for hydrogen combustion are lower than that for gasoline combustion. In these figures, the magnitude of cyclic variation is defined as the standard deviation of combustion period and shown as σ_{cp}.

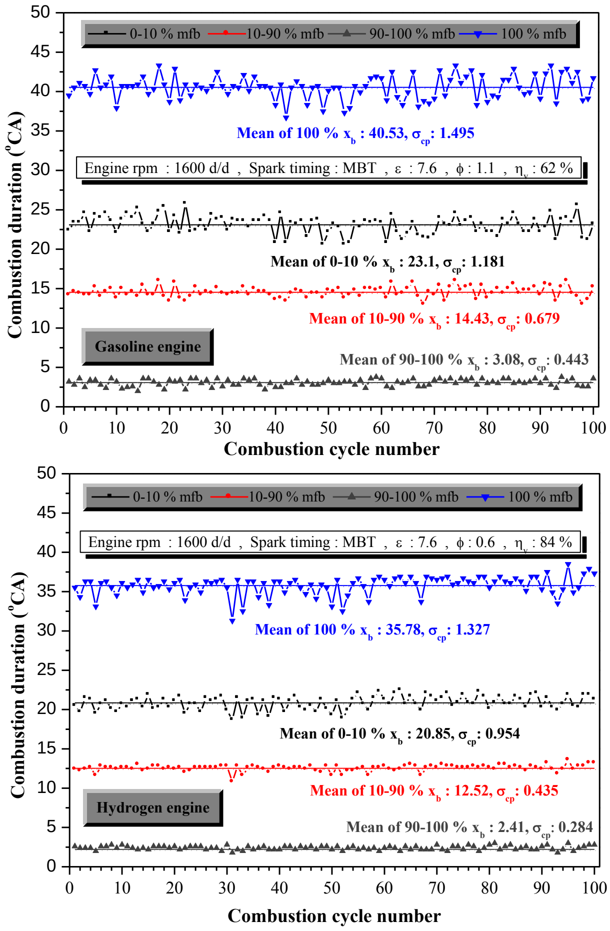

Figure 12. Variation of combustion duration in 0-10% mfb, 10-90% mfb, 90-100% mfb and total burning angle periods with engine cycle number for hydrogen and gasoline engine.

The theories on the causes of cycle variation in spark ignition engines are largely divided into two: that it is caused by variation in the early period of combustion, and that it is caused by variation in the rapid combustion period (Kim et al., 2005).

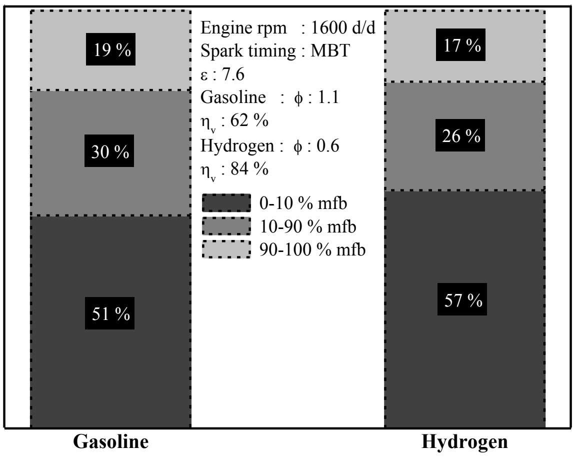

Figure 13. Effects on cyclic variations of combustion periods for hydrogen and gasoline engine.

Shown in Figure 13 is the causes of cyclic variations in each burning angle on total combustion period for both fuels. The greatest effect on total cyclic variation in the engine for gasoline and hydrogen combustion was caused by the variations in the flame development angle which is the slowest period. It should be noticed that this effect is higher for hydrogen operation than gasoline. In addition, the causes of cyclic variation in the last burning period are not as low as to be neglected.

CONCLUSIONS

A single cylinder SI gasoline engine was successfully modified to operate on hydrogen with manifold injection. This work was aimed to investigate the effect of fuel-air ratio on performance and cyclic variations of hydrogen and gasoline fueled spark ignition engine. The following conclusions are drawn on the experimental results obtained:

- A conventional spark ignition engine can be converted to hydrogen engine with minor modifications.
- The hydrogen engine was able to run smoothly with an equivalence ratio in the range of 0.3 - 0.9, whereas gasoline operation was in the range of 0.8 -1.3.
- The hydrogen engine was produced approximately 29% less maximum indicated power than that of gasoline engine due to a reduction in volumetric efficiency. There is an improvement of about 2% in indicated thermal efficiency. For the engines with similar output, the measured indicated specific fuel consumption was approximately 75% less for hydrogen engine compared to gasoline engine.
- The observed minimum value of COV_{IMEP} for hydrogen engine was around 0.85% at equivalence ratio of 0.75, whereas it was around 1.3% for gasoline engine at $\phi = 1.3$.
- It was found that the cyclic variations in all combustion periods for hydrogen engine were lower than that for gasoline engine.
- The greatest effect on total cyclic variation in the engine for gasoline and hydrogen combustion was found to be caused by the variations in the flame development period which is the slowest period. This effect was higher for hydrogen operation than gasoline. It should also be noticed that cyclic variation in the last burning angle is not to be underestimated.

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