American Journal of Scientific Research
ISSN 1450-223X Issue 1(2009), pp. 23-33

http://www.eurojournals.com/ajsr.htm

Effects of Air Fuel Ratio and Engine Speed on Engine Performance of Hydrogen Fueled Port Injection Engine

M. M. Rahman
Faculty of Mechanical Engineering, Universiti Malaysia Pahang
Leburaya Tun Razak,26300 Ganbang, Kuantan, Pahang, Malaysia
E-mail: mustafizur@ump.edu.my
Tel: +6095492207; Fax: +6095492244

Mohammed K. Mohammed
Faculty of Mechanical Engineering, Universiti Malaysia Pahang
Leburaya Tun Razak,26300 Ganbang, Kuantan, Pahang, Malaysia
E-mail: muhamad@ump.edu.my

Rosli A. Bakar
Faculty of Mechanical Engineering, Universiti Malaysia Pahang
Leburaya Tun Razak,26300 Ganbang, Kuantan, Pahang, Malaysia
E-mail: rosli@ump.edu.my

Abstract

This paper focuses on the effect of air-fuel ratio (AFR) and engine rotational speed on the performance of single cylinder hydrogen fueled port injection internal combustion (IC) engine. GT-Power was utilized to develop the model for port injection engine. One dimensional gas dynamics was represented the flow and heat transfer in the components of the engine model. Air-fuel ratio was varied from stoichiometric limit to a lean limit and the rotational speed varied from 2500 to 4500 rpm while the injector location was considered fixed in the midway of the intake port. The effects of air fuel ratio and engine speed are presented in this paper. From the acquired results show that the air-fuel ratio and engine speed were greatly influence on the performance of hydrogen fueled engine. It was shown that decreases the brake mean effective pressure (BMEP) and brake thermal efficiency with increases of the engine speed and air-fuel ratio however the increases the brake specific fuel consumption (BSFC) with increases the speed and air-fuel ratio. The cylinder temperature increases with increases of engine speed however temperature decreases with increases of air-fuel ratio. The volumetric efficiency increases with increases of engine speed and equivalent ratio. The volumetric efficiency of the hydrogen engines with port injection is a serious problem and reduces the overall performance of the engine. This emphasized the ability of retrofitting the traditional engines with hydrogen fuel with minor modifications.

Keywords: Hydrogen Fueled Engine, Port Injection, Air Fuel Ratio, Engine Speed, Performance Characteristics.

1. Introduction

Energy is currently uppermost in everyone's minds. Recent price hikes for domestic gas and electricity, coupled with fluctuating oil prices, concerns about where our future energy will come from and the impact of energy use on climate change have all brought energy policy to the top of the political agenda. The alarming rise in pollution levels in the atmosphere and increased concern for energy independency are the two major driving factors for seeking alternative fuels. Among the viable options, hydrogen is the only non-carbonaceous fuel available on the earth. Therefore, one can expect smoke, hydrocarbon and carbon monoxide free engine operation with hydrogen. Hydrogen has been regarded as a future secondary fuel for power systems due to CO_2 and hydrocarbon free operation. Recent drastic increase in the price of petroleum, rapid increase in emission of green house gases and very strict environmental legislations are major motivating factors for usage of hydrogen in fuel cells and internal combustion engines. Hydrogen as a fuel for internal combustion engine is best suited for spark ignition due to its high self-ignition temperature (Ganesh et al., 2008). A lean mixture results in an improvement of fuel economy. More complete combustion, lower combustion temperature and reduced pollutants (COD, 2001). Normally, cycle by 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 et al., 2005). Hydrogen needs very low ignition energy, which ensures prompt ignition. Hydrogen induction techniques play a very dominant and sensitive role in determining the performance characteristics of the hydrogen fueled internal combustion engine (H_2ICE) (Suwanchotchoung, 2003). Hydrogen fuel delivery system can be broken down into three main types including the carbureted injection, port injection and direct injection (COD, 2001).

The port injection fuel delivery system (PFI) injects hydrogen directly into the intake manifold at each intake port rather than drawing fuel in at a central point. Typically, hydrogen is injected into the manifold after the beginning of the intake stroke (COD, 2001). Hydrogen can be introduced in the intake manifold either by continuous or timed injection. The former method produces undesirable combustion problems, less flexible and controllable (Das, 1990). But the latter method, timed port fuel injection (PFI) is a strong candidate and extensive studies indicated the ability of its adoption (Das et al., 2000; Das, 2002). The calling sounds for adopting this technique are supported by a considerable set of advantages. It can be easily installed only with simple modification (Lee et al.,1995) and its cost is low (Li and Karim, 2006). The flow rate of hydrogen supplied can also be controlled conveniently (Sierens and Verhelst, 2001). External mixture formation by means of port fuel injection also has been demonstrated to result in higher engine efficiencies, extended lean operation, lower cyclic variation and lower NO_x production (Yi et al. 2000; Rottengruber et al., 2004; Kim et al. 2006). This is the consequence of the higher mixture homogeneity due to longer mixing times for PFI. Furthermore, external mixture formation provides a greater degree of freedom concerning storage methods (Verhelst et al., 2006). The most serious problem with PFI is the high possibility of pre-ignition and backfire, especially with rich mixtures (Kabat and Heffel, 2002; Ganesh et al., 2008). However, conditions with PFI are much less severe and the probability for abnormal combustion is reduced because it imparts a better resistance to backfire (COD, 2001). Combustion anomalies can be suppressed by accurate control of injection timing and elimination of hot spots on the surface of the combustion as suggested by Lee et al. (1995). The present study introduces a model for a single cylinder, port injection hydrogen fueled internal combustion engine. The objective if this study is to investigate the effect of air fuel ratio and engine speed on the performance characteristics of this engine.

2. Materials and Methods

2.1. Engine Performance Parameters

The brake mean effective pressure (*BMEP*) can be defined as the ratio of the brake work per cycle W_b to the cylinder volume displaced per cycle V_d, and expressed as:

$$BMEP = \frac{W_b}{V_d} \tag{1}$$

This equation can be extended for the present four stroke engine to:

$$BMEP = \frac{2P_b}{NV_d} \tag{2}$$

where P_b is the brake power, and N is the rotational speed.

Brake efficiency (η_b) can be defined as the ratio of the brake power P_b to the engine fuel energy as:

$$\eta_b = \frac{P_b}{\dot{m}_f(LHV)} \tag{3}$$

where $\dot{m}_f$ is the fuel mass flow rate; and *LHV* is the lower heating value of hydrogen.

The brake specific fuel consumption (*BSFC*) represents the fuel flow rate $\dot{m}_f$ per unit brake power output P_b and can be expressed as:

$$BSFC = \frac{\dot{m}_f}{P_b} \tag{4}$$

The volumetric efficiency (η_v) of the engine defines the mass of air supplied through the intake valve during the intake period, $\dot{m}_a$, by comparison with a reference mass, which is that mass required to perfectly fill the swept volume under the prevailing atmospheric conditions, and can be expressed as Eq. (5):

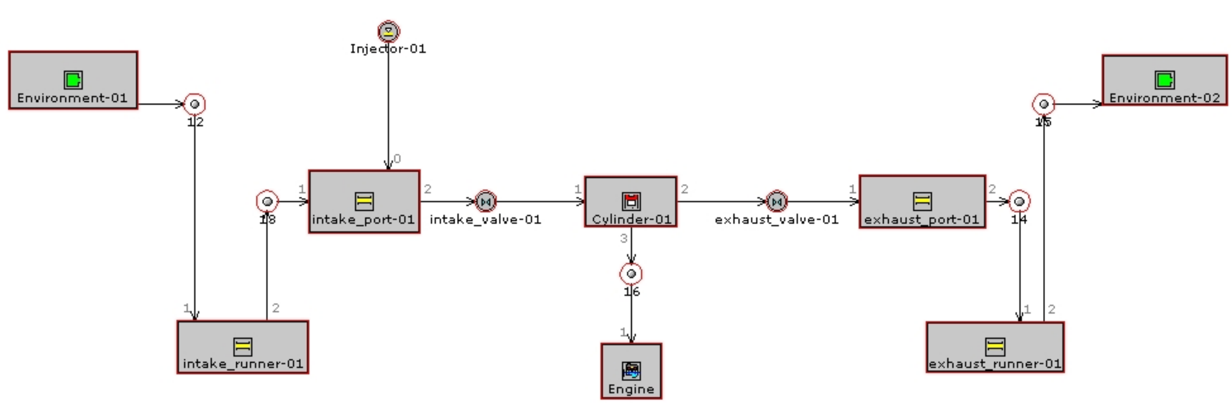
$$\eta_v = \frac{\dot{m}_a}{\rho_{ai} V_d} \tag{5}$$

where ρ_{ai} is the inlet air density.

2.2. Engine Model

A single cylinder, four stroke, port injection hydrogen fueled engine was modeled utilizing the GT-Power software. The injection of hydrogen was located in the midway of the intake port. The model of the hydrogen fueled single cylinder four stroke port inject engine is shown in Fig. 1. The specific engine parameters are used to make the model which is listed in Table 1. It is important to indicate that the intake and exhaust ports of the engine cylinder are modeled geometrically with pipes. Several considerations were made the model more realistic. Firstly, an attribute “heat transfer multiplier” is used to account for bends, roughness, and additional surface area and turbulence caused by the valve and stem. Also, the pressure losses in these ports are included in the discharge coefficients calculated for the valves.

Figure 1: Model of Single Cylinder, Four Stroke, Port Injection Hydrogen Fueled Engine



The in-cylinder heat transfer is calculated by a formula which closely emulates the classical Woschni correlation. Based on this correlation, the heat transfer coefficient (h_c) can be expresses as Eq. (6):

$$h_c = 3.26B^{-0.2}p^{0.8}T^{-0.55}w^{0.8} \tag{6}$$

where B is the bore in meters, p is the pressure in kPa, T is temperature in K and w is the average cylinder gas velocity in m/s.

The combustion burn rate (X_b) using Wiebe function, can be expressed as Eq. (7):

$$X_b = 1 - \exp\left[-a\left(\frac{\theta-\theta_i}{\Delta\theta}\right)^{n+1}\right] \tag{7}$$

where θ is the crank angle, θ_i is the start of combustion, $\Delta\theta$ the combustion period; and a and n are adjustable parameters.

Table 1: Hydrogen Fueled Engine Parameters

Engine Parameter (Unit)	Measure
Bore (mm)	100
Stroke (mm)	100
Connecting rod length (mm)	220
Piston pin offset (mm)	1.00
Displacement (liter)	0.785
Compression ratio	9.5
Inlet valve close IVC (^{0}CA)	-96
Exhaust valve open EVO (^{0}CA)	125
Inlet valve open IVO (^{0}CA)	351
Exhaust valve close EVC (0CA)	398

3. Results and Discussion

It is worthy to mention that one of the most attractive combustive features for hydrogen fuel is its wide range of flammability. A lean mixture is one in which the amount of fuel is less than stoichiometric mixture. This leads to fairly easy to get an engine start. Furthermore, the combustion reaction will be more complete. Additionally, the final combustion temperature is lower reducing the amount of pollutants.

Figure 2 shows the effect of air-fuel ration on the brake mean effective pressure. The air-fuel ratio AFR was varied from stoichiometric limit (AFR = 34.33:1 based on mass where the equivalence ratio ϕ = 1) to a very lean limit (AFR =171.65 based on ϕ = 0.2) and engine speed varied from 2500 rpm to 4500 rpm. BMEP is a good parameter for comparing engines with regard to design due to its independent on the engine size and speed. If torque used for engine comparison, a large engine was always seem to be better when considering the torque, however, speeds become very important when considered the power (Pulkrabek, 2003). It can be seen that the decreases of the *BMEP* with increases of *AFR* and speed. It is obvious that the *BMEP* falls with a non-linear behavior from the richest condition where *AFR* is 34.33 to the leanest condition where the *AFR* is 171.65. The differences of *BMEP* are increases with the increases of speed and *AFR*. The differences of the BMEP are decreases 6.682 bar at speed of 4500 rpm while 6.12 bar at speed 2500 rpm for the same range of *AFR*. This implied that at lean operating conditions, the engine gives the maximum power (*BMEP* = 1.275 bar) at lower speed 2500 rpm) compared with the power (*BMEP* = 0.18 bar) at speed 4500 rpm. Due to dissociation at high temperatures following combustion, molecular oxygen is present in the burned gases under stoichiometric conditions. Thus some additional fuel can be added and partially burned. This increases the temperature and the number of moles of the burned gases in the cylinder. These effects increases the pressure were given increase power and mean effective pressure (Heywood, 1988).

Figure 2: Variation of Brake Mean Effective Pressure with Air Fuel Ratio for Various Engine Speeds

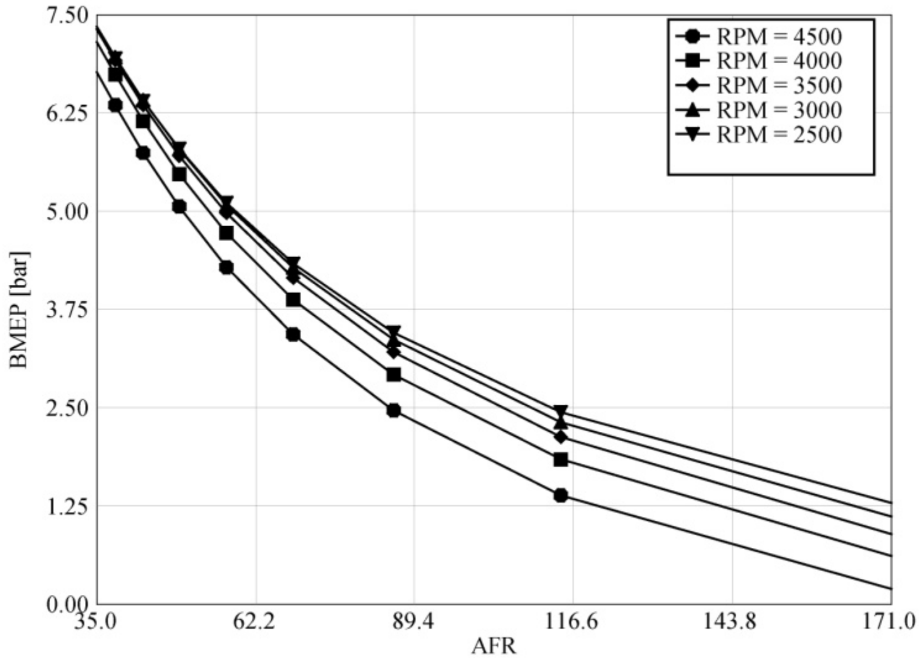


Figure 3 shows the variation of the brake thermal efficiency with the air fuel ratio for the selected speeds. It is seen that the brake power (useful part) as a percentage from the intake fuel energy. The fuel energy are also covered the friction losses and heat losses (heat loss to surroundings, exhaust enthalpy and coolant load). Therefore lower values of η_b can be seen in the Fig.3. It can be observed that the brake thermal efficiency is increases nearby the richest condition (AFR ≅ 35) and then decreases with increases of AFR and speed. The operation within a range of AFR from 49.0428 to 42.91250 (ϕ = 0.7 to 0.8) give the maximum values for η_b for all speeds. Maximum η_b of 31.8% at speed 2500 rpm can be seen compared with 26.8% at speed 4500 rpm. Unaccepted efficiency η_b of 2.88% can be seen at very lean conditions with AFR of 171.65 ($\phi = 0.2$) for speed of 4500 rpm while the efficiency was observed 20.7% at the same conditions with speed of 2500 rpm. Clearly, rotational speed has a major effect in the behavior of η_b with AFR. Higher speeds lead to higher friction losses.

Figure 3: Variation of Brake Thermal Efficiency with Air Fuel Ratio

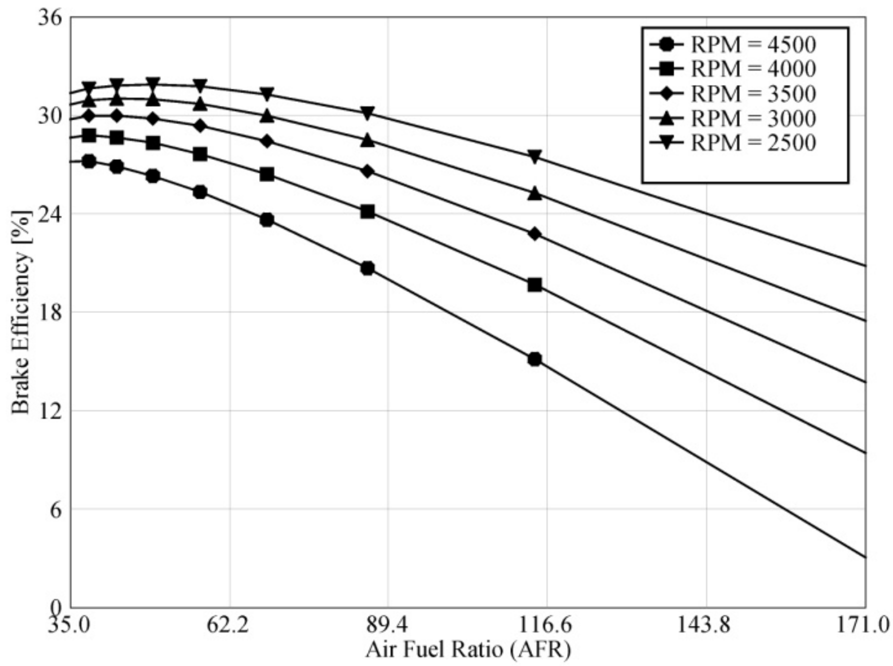


Figure 4 depicts the behavior of the brake specific fuel consumption *BSFC* with AFR. The AFR for optimum fuel consumption at a given load depends on the details of chamber design (including compression ratio) and mixture preparation quality. It varies for a given chamber with the part of throttle load and speed range (Heywood, 1988). It is clearly seen from Fig. 4 that the higher fuel is consumed at higher speeds and AFR due to the greater friction losses that can occur at high speeds. It is easy to perceive from the figure that the increases of BSFC with decreases in the rotational speed and increases the value of AFR. However, the required minimum *BSFC* were occurred within a range of AFR from 38.144 ($\phi = 0.9$) to 49.0428 ($\phi = 0.7$) for the selected range of speed. At very lean conditions, higher fuel consumption can be noticed. After AFR of 114.433 ($\phi = 0.3$) the *BSFC* goes up rapidly, especially for high speed. At very lean conditions with AFR of 171.65 ($\phi = 0.2$), a *BSFC* of 144.563 *g/kW-h* was observed for the speed of 2500 rpm while 1038.85*g/kW-h* for speed of 4500 rpm. The value *BSFC* at speed 2500 rpm was observed around 2 times at speed 4000 rpm however around 7 times at speed 4500 rpm. This is because of very lean operation conditions can lead to unstable combustion and more lost power due to a reduction in the volumetric heating value of the air/hydrogen mixture. This behavior can be more clarified by referring to Fig. 3, where the brake efficiency reduced considerably at very lean operation conditions.

Figure 4: Variation of Brake Specific Fuel Consumption with Air Fuel Ratio for Different Engine Speed

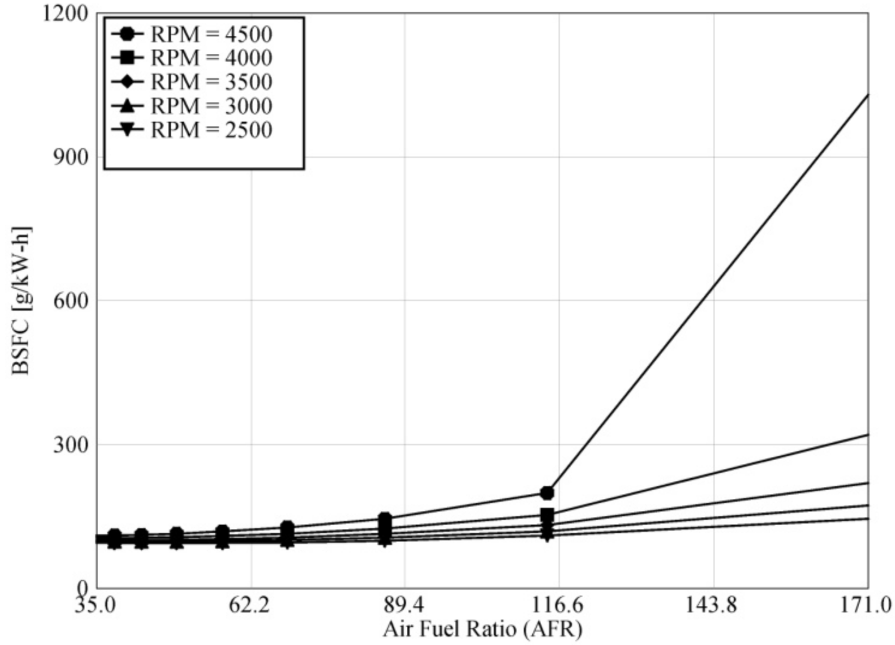


Figure 5 shows how the AFR can affect the maximum temperature inside the cylinder. In general, lower temperatures are required due to the reduction of pollutants. It is clearly demonstrated how the increase in the AFR can decrease the maximum cylinder temperature with a severe steeped curve. The effect of the engine speed on the relationship between maximum cylinder temperatures with AFR seems to be minor. At stoichiometric operating conditions (AFR= 34.33), a maximum cylinder temperature of 2752.83 *K* was recorded. This temperature dropped down to 1350 *K* at AFR of 171.65 ($\phi = 0.2$). This lower temperature inhibits the formation of NO_x pollutants. In fact this feature is one of the major motivations toward hydrogen fuel.

Figure 5: Variation of Maximum Cylinder Temperature with Air Fuel Ratio

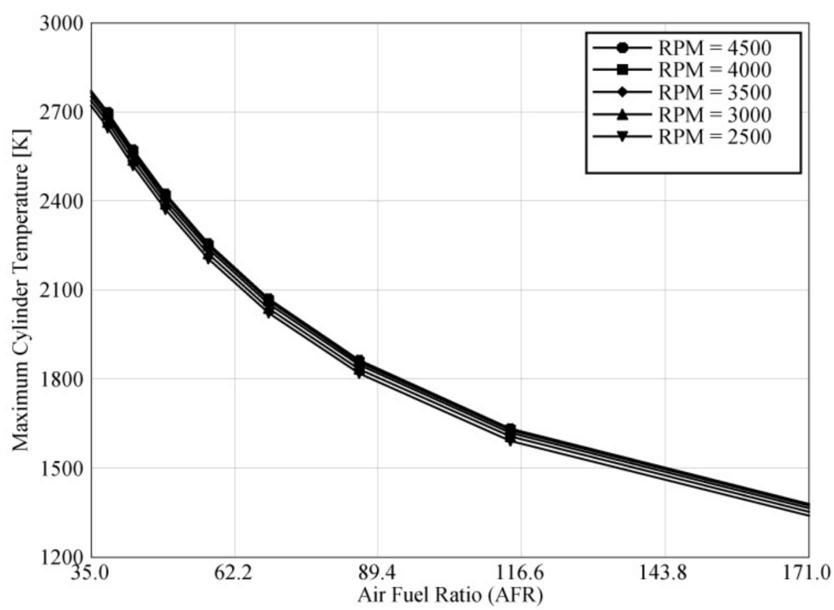
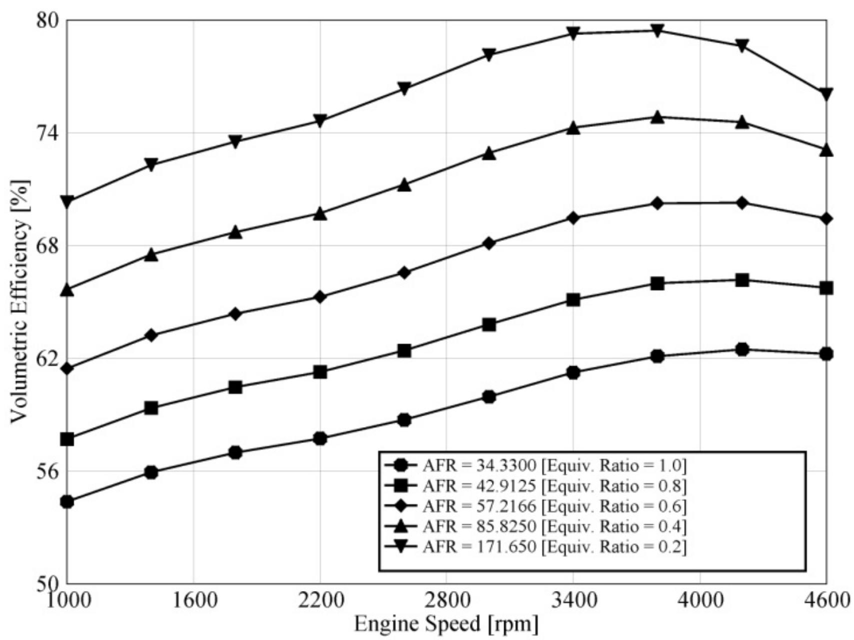


Figure 6 shows the variation of the volumetric efficiency with the engine speed. In general, it is desirable to have maximum volumetric efficiency for engine. The importance of volumetric efficiency is more critical for hydrogen engines because of the hydrogen fuel displaces large amount of the incoming air due to its low density (0.0824 kg/m^3 at 25^{0}C and 1 atm.). This reduces the volumetric efficiency to high extent. For example, a stoichiometric mixture of hydrogen and air consists of approximately 30% hydrogen by volume, whereas a stoichiometric mixture of fully vaporized gasoline and air consists of approximately 2% gasoline by volume (White, 2006). Therefore, the low volumetric efficiency for hydrogen engine is expected compared with gasoline engine works with the same operating conditions and physical dimension. This lower volumetric efficiency is apparent in Fig. 6. Leaner mixture gives the higher volumetric efficiency. The maximum volumetric efficiency was observed 79.4% at lean conditions with AFR = 171.65 ($\phi = 0.2$) while 62.4% at stoichiometric conditions.

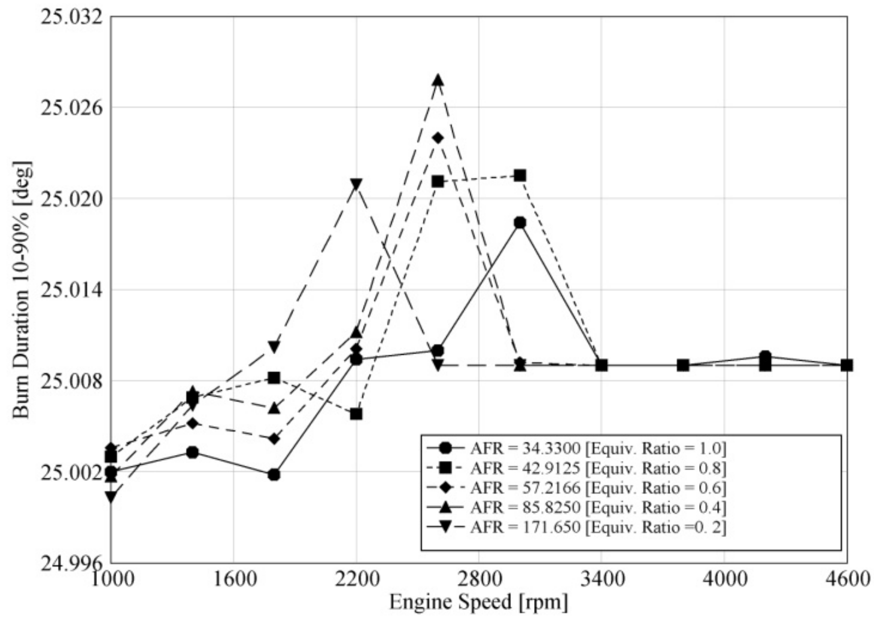
Figure 6: Effect of Volumetric Efficiency with the Rotational Speed for Different Equivalence Ratio



Higher speeds lead to higher volumetric efficiency because of the higher speeds give higher vacuum at the intake port and consequent larger air flow rate that goes inside the cylinder. Further increase in engine speed leads toward the maximum value of η_v. For the considered speeds, and with equivalence ratios of 1, 0.8 and 0.6, the maximum η_v was recorded at 4200 rpm. For equivalence ratio of 0.4 and 0.2, the maximum η_v was recorded at 3800 rpm. At further higher engine speeds beyond these values, the flow into the engine during at least part of the intake process becomes chocked. Once this occurs, further increase in speed do not increase the flow rate significantly so volumetric efficiency decreases sharply. This sharp decrease happens because of higher speed is accompanied by some phenomenon that have negative influence on η_v. These phenomenons include the charge heating in the manifold and higher friction flow losses which increase as the square of engine speed. In fact a lot of solutions were suggested to solve this problem. Nagalingam et al. (1983), Furuhama and Fukuma (1986) and Lynch (1983) suggested and carried out tests with pressure boosting systems for hydrogen engines.

Figure 7 shows the combustion duration as a function of the engine speed for different equivalence ratio. As stated earlier, hydrogen combustion velocity (1.85 m/s) is rapid compared with that of gasoline (0.37-0.43 m/s). Therefore short combustion duration is expected. It is well established that the duration of combustion in crank angle degrees only increases slowly with increasing speed for gasoline and diesel engines (Heywood, 1988). Figure 7 shows that this fact is also true for hydrogen engines. The fluctuation shown is very small, however, it was enlarged in the figure with a very high scale. All the changes take place within a range of 0.0248 degree. This is too small value, especially if one knows that at 4500 rpm, the crank shaft rotates 27000 degree within 1 second.

Figure 7: Variation of Combustion Duration with Engine Speed for Different Equivalence Ratio



5. Conclusions

The present study considered the performance characteristics of single cylinder hydrogen fueled internal combustion engine with hydrogen being injected in the intake port. The emphasis was paid to the effects of engine speed, AFR. The instantaneous behavior was also studied. The following conclusions are drawn:

(1) At very lean conditions with low engine speeds, acceptable *BMEP* can be reached, while it is unacceptable for higher speeds. Lean operation leads to small values of *BMEP* compared with rich conditions.

(2) Maximum brake thermal efficiency can be reached at mixture composition in the range of (ϕ= 0.7 to 0.8) and it decreases dramatically at leaner conditions.

(3) The desired minimum *BSFC* occurs within a mixture composition range of (ϕ = 0.7 to 0.9). The operation with very lean condition ($\phi < 0.2$) and high engine speeds (> 4500) consumes unacceptable amounts of fuel.

(4) Lean operation conditions results in lower maximum cylinder temperature. A reduction of around 1400 *K* can be gained if the engine works properly at ($\phi = 0.2$) instead of stoichiometric operation.

(5) The low values of volumetric efficiency seem a serious challenge for the hydrogen engine and further studied are required.

Acknowledgement

The authors would like to express their deep gratitude to Universiti Malaysia Pahang (UMP) for provided the laboratory facilities and financial support.

References

[1] Blair, G.P. 1999. Design and simulation of four stroke engines. SAE International. Society of Automotive Engineers Inc. Warrandale, Pa., USA. ISBN: 978-0-7680-0440-3, pp. 1-840. http://www.sae.org/technical/books/R-186

[2] COD (College of the Desert). 2001. Hydrogen fuel cell engines and related technologies, module 3: Hydrogen use in internal combustion engines. Rev. 0, pp. 1-29. http://www1.eere.energy.gov/hydrogenandfuelcells/tech_validation/h2_manual.html

[3] Das, L. 2002. Hydrogen engine: research and development programmes in Indian Institute of Technology (IIT), Delhi. Int. J. Hydrogen Energy, 27(9): 953-965. doi:10.1016/S0360-3199(01)00178-1

[4] Das, L., R. Gulati and P. Gupta. 2000. A comparative evaluation of the performance characteristics of a spark ignition engine using hydrogen and compressed natural gas as alternative fuels. Int. J. Hydrogen Energy, 25(8): 783-793. doi:10.1016/S0360-3199(99)00103-2

[5] Das, L.M. 1990. Fuel induction techniques for a hydrogen operated engine. Int. J. Hydrogen Energy,15(11):833-842. doi:10.1016/0360-3199(90)90020-Y

[6] Furuhama, S. and T. Fukuma. 1986. High output power hydrogen engine with high pressure fuel injection, hot surface ignition and turbocharging. Int. J. Hydrogen Energy, 11(6): 399–407. doi:10.1016/0360-3199(86)90029-7

[7] Ganesh, R.H., V. Subramanian, V. Balasubramanian, J.M. Mallikarjuna, A. Ramesh and R.P. Sharma. 2008. Hydrogen fueled spark ignition engine with electronically controlled manifold injection: An experimental study. Renv. Energy,33(8): 1324-1333. doi:10.1016/j.renene.2007.07.003

[8] Heywood, J.B. 1988. Internal combustion engine fundamentals. London: McGraw-Hill, 1st Ed., pp. 1-930, ISBN 007028637X

[9] Kabat, D.M. and J.W. Heffel. 2002. Durability implications of neat hydrogen under sonic flow conditions on pulse-width modulated injectors. Int. J. Hydrogen Energy, 27(10):1093-1102. doi:10.1016/S0360-3199(02)00007-1

[10] Kim Y.Y., T.L. Jong and H.C. Gyeung. 2005. An investigation of the cycle to cycle variation in direct injection hydrogen fueled engine. Int. J. Hydrogen Energy, 30: 69-76.

[11] Kim, Y.Y., J.T. Lee and J.A. Caton. 2006. The development of a dual-Injection hydrogen-fueled engine with high power and high efficiency. J. Eng. Gas Turbines Power, ASME, 128(1): 203-212. doi: 10.1115/1.1805551.

[12] Knorr, H., W. Held, W. Prümm and H. Rüdiger. 1997. The man hydrogen propulsion system for city buses. Int. J. Hydrogen Energy, 23(3): 201–208. doi:10.1016/S0360-3199(97)00045-1

[13] Lee, S.J., H.S. Yi and E.S. Kim. 1995. Combustion characteristics of intake port injection type hydrogen fueled engine. Int. J. Hydrogen Energy, 20(4): 317-322. doi:10.1016/0360-3199(94)00052-2

[14] Li, H. and G.A. Karim. 2006. Hydrogen fueled spark-ignition engines predictive and experimental performance. J. Eng. Gas Turbines Power, ASME, 128(1): 230-236. doi: 10.1115/1.2055987

[15] Lynch, F.E. 1983. Parallel induction: a simple fuel control method for hydrogen engines. Int. J. Hydrogen Energy,8(9):721–730. doi:10.1016/0360-3199(83)90182-9.

[16] Meier, F., J. Karhler, W. Stolz, W.H. Bloss and M. Al-Garni. 1994. Cycle-resolved hydrogen flame speed measurement with high-speed Schlieren technique in a hydrogen direct injection SI engine. SAE Paper No. 942036. http://www.sae.org/technical/papers/942036.

[17] Nagalingam, B., M. Dübel and K. Schmillen. 1983. Performance of the supercharged spark ignition hydrogen engine. SAE Paper No. 831688. http://www.sae.org/technical/papers/831688

[18] Pulkrabek, W.W. 2003. Engineering fundamentals of the internal combustion engines. Second edition. New York: Prentic Hall. ISBN-10: 0131405705

[19] Rottengruber, H., M. Berckmüller, G. Elsässer, N. Brehm and C. Schwarz. 2004. Direct-injection hydrogen SI-engine operation strategy and power density potentials. SAE Paper No. 2004-01-2927. http://www.sae.org/technical/papers/2004-01-2927

[20] Sierens, R. and S. Verhelst. 2001. Experimental study of a hydrogen-fueled engine. J. Eng. Gas Turbines Power, ASME, 123(1): 211-216. doi: 10.1115/1.1339989.

[21] Sierens, R. and S.Verhelst. 2003. Influence of the injection parameters on the efficiency and power output of a hydrogen fueled engine. J. Eng. Gas Turbines Power, ASME, 125(2): 444–449. doi: 10.1115/1.1496777.

[22] Sierens, R., S. Verhelst and S. Verstraeten. 2005. An overview of hydrogen fueled internal combustion engines. Proc. Int. Hydrogen Energy Cong. and Exhib., IHEC 2005, Istanbul, Turkey, 13-15 July, 2005. http://www.unido-ichet.org/ihec2005/files/manuscripts/Roger%20Sierens-Belgium.pdf

[23] Suwanchotchoung, N. 2003. Performance of a spark ignition dual-fueled engine using split-injection timing. Ph.D. thesis, Vanderbilt University, Mechanical Engineering. ISBN 049633606X

[24] Verhelst, S., R. Sierens and S. Verstraeten S. 2006. A critical review of experimental research on hydrogen fueled SI engines. SAE Paper No. 2006-01-0430. http://www.sae.org/technical/papers/2006-01-0430.

[25] White, C.M., R.R. Steeper and A.E. Lutz. 2006. The hydrogen-fueled internal combustion engine: a technical review. Int. J. Hydrogen Energy, 31(10):1292–1305. doi:10.1016/j.ijhydene.2005.12.001

[26] Yi, H.S., K. Min and E.S. Kim. 2000. The optimised mixture formation for hydrogen fuelled. Int. J. Hydrogen Energy, 25(7):685-690. doi:10.1016/S0360-3199(99)00082-8