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The Combustion Characteristics of a Diesel Engine with Hydrogen Addition

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Abstract

Among the alternative fuels, hydrogen (H_2) shows great potential in the near future. In this work, an experiment was conducted to investigate the effect of H_2 addition on the combustion characteristics of a heavy-duty diesel engine with. The results showed that at 25% engine load, the peak cylinder pressure (PCP) and peak heat release rate (PHRR) observed slightly decreased with increasing the H_2 addition compared to neat diesel mode while they were ignorable at 50% of load. Under high loads (75% and 100% loads), the addition of H_2 to conventional diesel fuel enhanced the PCP and PHRR due to the increase of injection spray plume in which more H_2-air mixture entrained into diesel spray plume, mixed with diesel vapor and simultaneously burned. On the other hand, the indicated mean effective pressure (IMEP) gradually increased at 25% load and slightly declined at 75% and 100% loads as H_2 was premixed with intake air. It was also found that the ignition delay slightly increased except 100% load due to the reduction of local oxygen concentration and lower cylinder temperature at the end of compression stroke while there were inconsiderable changes in coefficient of variation (COV_{imep}) with H_2 enrichment.

***Keywords**:* diesel engine, hydrogen combustion, dual fuel, combustion characteristics.

1. Introduction

The dependency on IC engines will not decrease in the next few decades. They are widely applied to a power source for land vehicles, commercial marine vessels, and stationary power plants. However, the crises of energy and environment are caused by consuming fossil fuels long and using the engines widely. Therefore, researchers around the world have been studying on technology for decreasing exhaust emissions from engines and on alternative fuels for solving these crises.

As a long time recognized carbon-free fuel, hydrogen (H_2) can be used in traditional IC engines, gas turbines and also the fuel cells. Compared with conventional fuels, H_2 has the following characteristics [1-3], such as a long term renewable, recyclable and non-polluting fuel because it is without carbon. And H_2 gains much higher flame speed and larger diffusion speed so it can benefit the energy efficiency and emissions. The limits of flammability of H_2 vary from an equivalence ratio of 0.1 to 7.1, and the engine is hence operated with a wide range of air/ fuel ratio.

In spark ignition (SI) engines, adding H_2 to intake air (known as H_2-enrichment) have had some benefits such as improving the brake thermal efficiency, accelerating the flame propagation, extending the lean operational region and enhancing the combustion stability at lean operation. In addition, by replacing a part of gasoline fuel by H_2 it results in reduction of the exhaust emissions of carbon monoxide (CO), unburned hydrocarbons (HC) and possibly nitrogen oxide (NO_x) if operated at an extremely lean mixture [4-8].

Alternative hydrocarbon fuels for compression ignition (CI) engines have been considered because diesel engines have higher thermal efficiency and lower CO_2 emission than those of gasoline engines [9]. However H_2 cannot be used as a sole fuel in compression ignition (CI) engines, since the compression temperature is not enough to initiate the combustion due to its higher self-ignition temperature [10]. Hence an ignition source is required while using it in a CI engine. Hydrogen has an auto-ignition temperature of 858 K requiring an ignition source in an IC engine. Diesel fuel, which has an auto-ignition temperature of 525 K, can be used as a pilot fuel to ignite hydrogen. The literature on diesel pilot-ignited hydrogen combustion hints that hydrogen substitution is a promising method of reducing undesired exhaust emissions, especially at high rates of hydrogen substitution.

Miyamoto et al. [11] conducted an experiment to investigate the effect of H_2 addition on combustion characteristics and emissions of a direct injection, single cylinder diesel engine. On this work, sent hydrogen amount has been increased till 16% by volume, and injection of diesel has been made after top dead point. With this method, NO_x emissions and in-cylinder pressure increase rate kept under control. Also, HC emissions did not increase and NO_x emissions are reduced with hydrogen addition.

Adding H_2 is also expected to improve combustion of diesel or biodiesel. Nagalingam et al. [12] conducted an experiment on a diesel engine primarily fueled with a vegetable oil, namely Jatropha oil and small quantities of H_2 to evaluate the effects of H_2 on combustion process. Results indicated that there were benefits for performance and emission. Particularly, the brake thermal efficiency increased from 27.3% to a maximum of 29.3% at 7% of H_2 mass share at the maximum engine load. For emissions, smoke significantly reduced (by 20%) and there was also a decrease in HC and CO emissions from 130 to 100 ppm and 0.26%– 0.17% (by volume), respectively, at full load. However, the NO level was increased from 735 to 875 ppm due to higher in-cylinder temperature with H_2 burning at maximum output. The influence of H_2 addition on combustion process was observed in their work. There were increases in ignition delay, peak pressure and the maximum rate of pressure rise in the dual-fuel mode of operation. However, combustion duration was reduced due to higher flame speed of H_2.

In order to know the effects of the ambient gas on auto-ignition delays of H_2 jets, Tsujimura et al. [13] conducted an experiment by using a constant volume vessel. The thermodynamic states of the ambient gas inside the vessel, which influenced on auto-ignition delays of H_2 jets, were observed in his work. His results indicated that the ambient gas temperature has a significant effect on the auto-ignition delay of the H_2 jet. There was a linear relationship between the auto-ignition delay and temperature in the Arrhenius coordinates when the ambient gas temperatures were below about 1100 K. In addition, above this temperature, the temperature dependence of the auto-ignition delay is weak, and the auto-ignition delay reaches a limited value.

Heat release rates from H_2-diesel fuel co-combustion tend to be higher than those for diesel fuel combustion, resulting in a shorter duration of combustion with less heat transfer to the surroundings, and can improve thermal efficiencies [14-16]. However, Masood et al. [15] and Christodoulou et al. [16] reported a small deterioration in thermal efficiency when operating the engine with a H2-diesel fuel mixture, at low load and low speed conditions, which they attributed to the incomplete burning of all the hydrogen aspirated into the engine.

Although lots of studies in the literature show that CO, CO_2 and smoke emissions decrease with H_2 addition in diesel fueled engines, its effects on thermal efficiency, brake specific fuel consumption, THC and NO_x emissions are likely to depend on specific diesel engines and operating conditions [17-21]. Furthermore, the number of published papers in the field of H_2–diesel dual fuel operation is not as rich as for the H_2 used in spark ignited engines. For better understanding of the potential of H_2 in affecting the brake thermal efficiency and exhaust emissions of heavy duty diesel engines, there is a need to explore the detailed effect of H_2 addition and engine load on the combustion process including peak cylinder pressure, rate of pressure rise, heat release rate, the ignition delay, and combustion phasing. An experiment designed by ourselves is expected to help to clearly explain the effects of the addition of H_2 on H_2–diesel co-combustion at different engine loads. In this work the objective is to investigate the effect of hydrogen aspiration on the diesel combustion process as H_2 was used as a supplemental fuel in a commercial diesel engine to replace a portion of the diesel fuel burned in the engine.

2. Experimental apparatus

Fig. 1 shows a block diagram illustrating most of the experimental apparatus used in this study. It consists of a four-stroke, in-line six cylinder, naturally aspirated diesel engine coupled with an eddy current dynamometer (Schenck W230) for engine loading, an intake air system for measurement of volumetric air and H_2 flow rate, an electronic weighing balance for measurement of liquid fuel consumption. The main specifications of the engine are listed in Table 1. The engine was modified for dual-fuel operation. In particular, the intake and exhaust manifolds of 6th cylinder are separated from others in order to avoid the effect of difference of volumetric efficiency among cylinders. The average engine torque and speed for each test mode are given in Table 2. A piezoelectric pressure transducer with nominal sensitivity 15.78 pico-coulomb/bar was mounted on the 6th cylinder head of the engine for in-cylinder pressure measurement. An optical encoder was mounted on the engine's crankshaft for crank angle measurement with an accuracy of 0.1CA. The in-cylinder pressure analog signals were amplified by the charge amplifier and then, the analog signals were converted into digital signals by a data acquisition for further processing.

Table 1. Specifications of test engine

	Hino W06E
Year	1992
Configuration	In-line 6-cylinder
Air intake	Naturally aspirated
Fuel injection	Direct injection
Displacement	6.014L
Max torque	412 Nm @ 1800 rpm
Max power	121 kW @ 3000 rpm
EGR	No

Table 2. Engine test modes

Load (%)	European Transient Cycle mode	Torque (N. m)	Speed (rpm)
25	7	92	1650
50	5	184	1650
75	6	276	1650
100	2	360	1650

The H_2 quantities added was defined as the volumetric fraction of H_2 in the intake mixture [H_2/(Air + H_2), vol. %]. When the engine run with H_2 addition, the flow rate of diesel fuel was gradually reduced to keep the output remain the same. The engine load was varied from 25 to 100% at 1650 RPM by using a dynamometer. In this research, the H_2 fuel was brought from a local company (Taichung, Taiwan) having a purity of over 99.5% and the commercial diesel was also used.

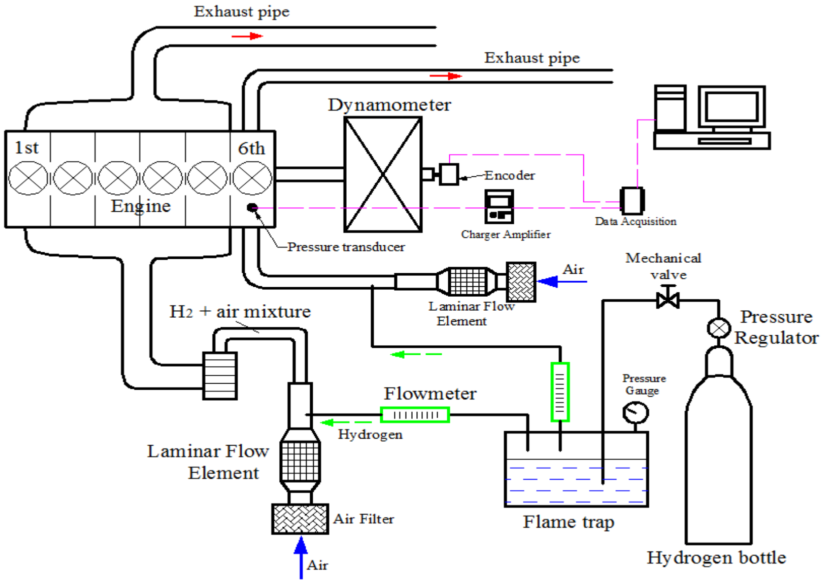

Fig. 1 Experimental apparatus

From the First law of thermodynamics for the engine work cycle and neglecting both heat transfer to the walls and fuel flow into crevices, the heat release rate Q_{hr} released by combustion can be determined by Eq. (1).

$$\frac{Q_{hr}}{d\theta} = \frac{\gamma}{\gamma - 1} p \frac{dV}{d\theta} + \frac{1}{\gamma - 1} V \frac{dp}{d\theta} \tag{1}$$

where:

γ – ratio of specific heats,
p – in-cylinder pressure,
V – cylinder volume.
θ - crank angle

Many researchers recommended COV_{imep} as the main parameter for evaluating the combustion stability of an engine because the comprehensive characteristics of combustion pressure can be described by COV_{imep}, especially in consideration of engine output. COV_{imep} is defined as Equation (2) in which the indicated mean effective pressure of the standard deviation σ_{imep} and the average indicated mean effective pressure $imep_{avg}$ are determined by equation s (3-4).

$$COV_{imep}\% = \frac{\sigma_{imep}}{imep_{avg}} \times 100\% \tag{2}$$

$$\sigma_{imep} = \sqrt{\frac{1}{N_{cycle}} \sum_{i=1}^{N_{cycle}} \left(imep_i - imep_{avg}\right)^2} \tag{3}$$

$$imep_{avg} = \frac{1}{N_{cycle}} \sum_{i=1}^{N_{cycle}} imep_i \tag{4}$$

A sample printout of net heat release rate vs. crank angle for the engine run on diesel only has been shown in Fig. 2. The net heat release rate becomes negative due to heat transfer from gas to the cylinder chamber and also by evaporation of fuel droplets. In this study, the start of combustion (SOC) is defined as the crank angle at which the heat release rate becomes zero and then tends to be positive.

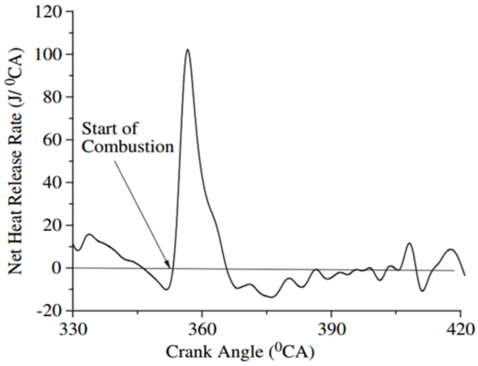

Fig. 2 Net heat release rate (J/°CA) vs. crank angle (°CA) [22]

3. Results and discussion

Fig. 3 shows the effect of H_2 addition on in-cylinder pressure at 25% load. As a result of H_2 supplement, it is observed that there were a modest retardant in the initiation of the premixed combustion and slight reduction of the PCP obtained in the premixed combustion due to the lengthened mixing-controlled combustion and H_2-air mixture was probably ignited during mixing-controlled combustion phase. The PCP is found to be 5791 kPa without H_2 and it reduced to 5768 kPa with 3% H_2 addition. Further increase in H_2 fraction of 4%, 5%, 6% and 7% reduced the PCP to 5743, 5720, 5684 and 5657 kPa, respectively. Moreover, the ignition delay that slightly increased with H_2 supplement was also considered as the reason making PCP lower. Although the PCP slightly decreased with H_2 addition, there was not the difference in the PHRR in the cases with and without H_2 addition (as shown in Fig. 4). In addition, at 50% load, the results show that the H_2 addition has very small effect on combustion process and so it is not presented in this paper.

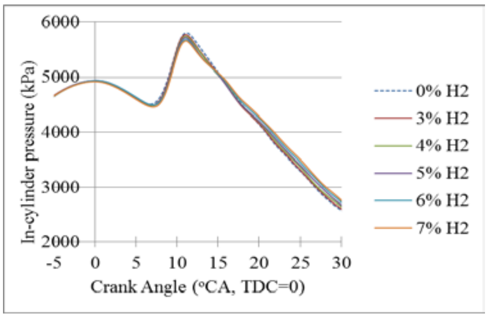

Fig. 3 Effect of H_2 addition on in-cylinder pressure at 25% engine load

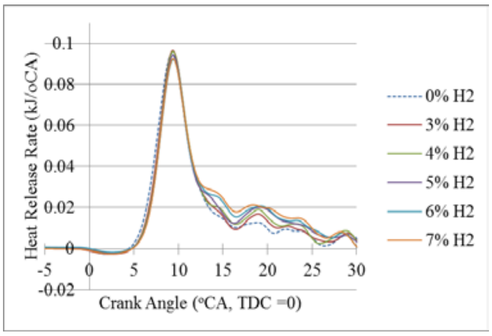

Fig. 4 Effect of H_2 addition on heat release rate at 25% engine load

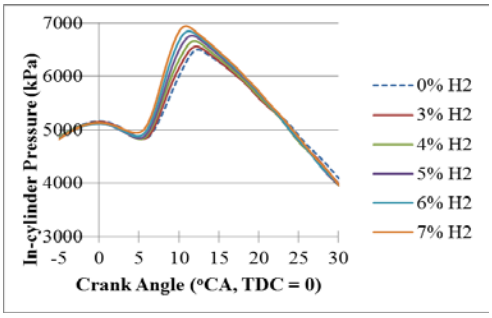

Fig. 5 Effect of H_2 addition on in-cylinder pressure at 100% engine load

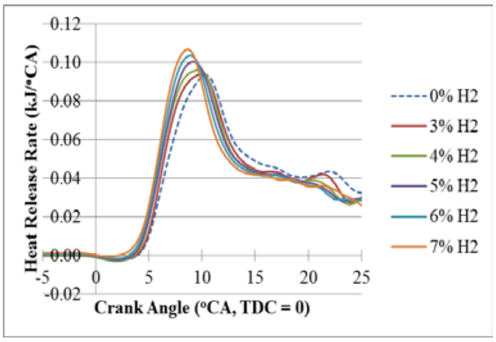

Fig. 6 Effect of H_2 addition on heat release rate at 100% engine load

The effect of H_2 addition on in-cylinder pressure and heat release rate at 75% and 100% loads are similar and thus the latter are plotted in Figs. 5 and 6. When operated at higher loads, a portion of H_2-air mixture being outside the injection spray plume is burned due to the increase of in-cylinder gas temperature and the high flame velocity of H_2 resulting in development of turbulent flame. In other part of my experiment, the unburned H_2 in exhaust gas was measured by using GC (Gas Chromatography) system in order to know the H_2 combustion efficiency. However, the result is not presented in this paper. There is a relationship between the H_2 combustion efficiency and the size of diesel spray. The higher engine load is the better H_2 combustion efficiency is. It is due to the increase of diesel spray plume and then apparent increase of heat release rate during diffusion combustion phase.

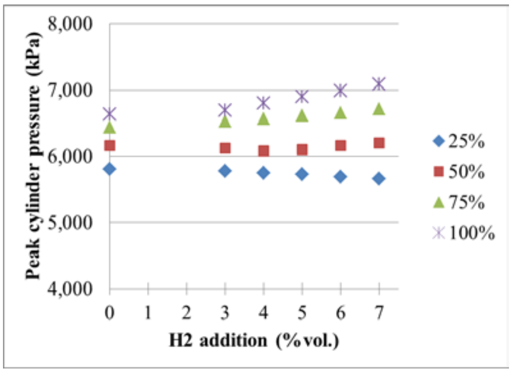

Fig. 7 Effect of H_2 addition and engine load on peak cylinder pressure

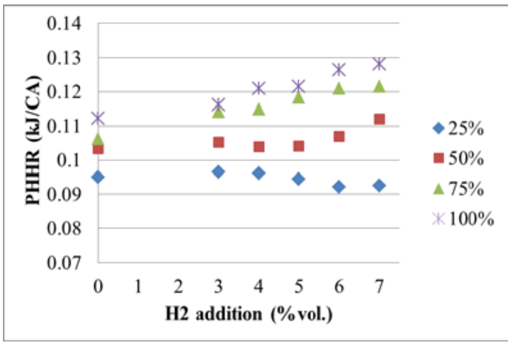

Fig. 8 Effect of H_2 addition and engine load on peak release rate

The effect of H_2 addition and engine loads on PCP and PHRR are shown in Figs. 7 and 8, respectively, in which the data have been averaged over 100 consecutive cycles. From these Figs, at low load, with the increase of H_2 addition, there is a gradual decrease of PCP and PHRR along with retardation of SOC due to the lengthened diffusion combustion phase and H_2 is probably ignited during this phase. The combustion efficiency of H_2 could also be used to explain to this phenomenon. When the engine was operated at 25% load and that unburned H_2 in the engine exhaust increased almost linearly with the increase in H_2 addition. As suggested in Ref. [23], there is an only limited amount of H_2–air mixture being inside the diesel spray ignited and burned with diesel-air mixture while the rest of the H_2 directly escapes into the exhaust. The higher reduction of size of diesel plume as a result of the more H_2 substitution is main reason for lower H_2 combustion efficiency.

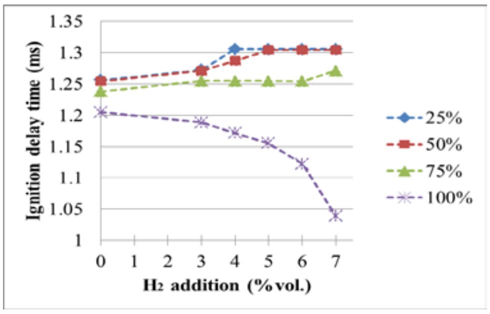

Fig. 9 Effect of H_2 addition on ignition delay at engine loads

Fig. 9 shows the influence of H_2 addition on ignition delay at various loads. As shown in this Fig, the ignition delay slightly increased with the increase of H_2 addition at tested loads apart from 100% load. In particular, at 25% load, the ignition delay for diesel is 1.256 ms and it is 1.306 ms and at 50% load, the ignition delay is 1.255 ms for diesel and it is 1.304 ms for 7% added hydrogen. The delay period slightly decreased at 75% load compared to lower loads (around 0.03ms). However, under maximum load, addition of H_2 reduced ignition delay. It was found that the ignition delays were 1.205 ms for neat diesel operation and 1.040 ms 7% of H_2 addition.

These variations of ignition delay may be explained on the basis of existing literature. There are two main factors affecting on ignition delay as H_2 is supplied to intake manifold. They are the reduction of oxygen concentration and the residual time of high temperature gas. The former increases ignition delay and the later decrease it. At lower engine loads, with H_2 enrichment, the reduction of the air intake and then, the local oxygen pressure and also the temperature of the charge (H_2-air mixture) in the cylinder at top dead center position [24] have dominated. Therefore, ignition delay increased. Conversely, when operated at maximum load, the smaller ratio of air-fuel and also higher combustion efficiency of H_2 made the higher in-cylinder temperature become the main factor impacting on ignition delay.

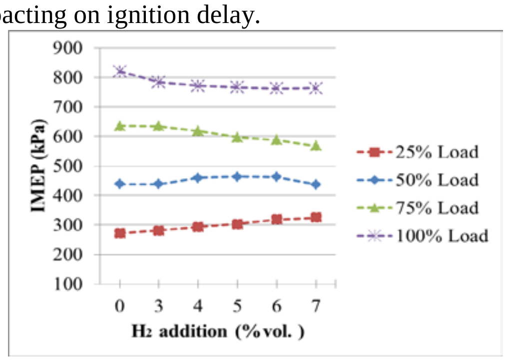

Fig. 10 Effect of H_2 addition on IMEP at engine loads

Fig. 10 shows the dependence of indicated mean effective pressure (IMEP) on the H_2 enrichment. At 25% load, IMEP gradually increased with H_2 addition. As already discussed, at low loads, the ignition sources decrease due to smaller diesel plume. However, the high flame velocity of H_2 property enhances the mixing controlled combustion process. Therefore, the heat release rate considerably reduced observed during the premixed combustion but it increased and elongated in diffusion combustion phase and correspondingly deteriorated the engine performance. In contrast, increasing H_2 concentration made IMEP slightly decrease at 75% and 100% of engine load. This is due to the presence of H_2 – air mixture in combustion chamber, at higher loads, the burning rate was enhanced and also increasing addition of H_2 made diesel fuel being reduced which lead to the end of injection earlier. Accordingly, the combustion duration was shorter.

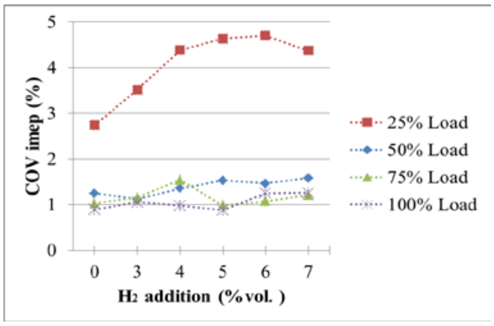

Fig. 11 The effect of H_2 addition on COV of IMEP at engine loads

The coefficient of variation (COV) of indicated mean effective pressure (IMEP) with engine loads at different H_2 addition is shown in Fig. 11. Problems often occur when COV of IMEP exceeds about 10% [25]. Fortunately, the COV of IMEP values for all modes with and without H_2 addition are lower than 10% (from 0.9% to 4.8%). As shown in this Fig., with the presence of H_2 in cylinder, the combustion stability tends to reduce compared to without H_2 at low load. However, at higher loads there is not much different in cyclic variability for both cases, with and without H_2 addition. In dual mode, the combustion process was initiated by diesel fuel due to its lower auto-ignition temperature. At lower engine load, the late start of combustion observed with H_2 addition result in the combustion instability. At higher loads and the same H_2 flow rate, diesel percentage on energy basis increased. The amount of diesel fuel to ignite the premixing of H_2 with air was increased and resulted in better combustion stability. Although H_2 addition reduced the percentage of diesel fuel it is sufficient to produce the efficient combustion. Therefore, generally speaking, adding H_2 has small effect on the cyclic variation of combustion of diesel/H_2 mixtures in the diesel engine.

4. Conclusions

In a heavy-duty diesel engine, the engine load and amount of H_2 added play important role in the combustion process with addition of H_2. At 25% load, the attendance of H_2 and decrease of diesel fuel deteriorated the first phase of combustion process but slightly enhanced and elongated the diffusion combustion. Consequently, increasing the addition of H_2 resulted in slightly decrease PCP, PHRR and slightly increase in ignition delay. When operated at 50% load, the addition of H_2 only had mild or negligible effect on in-cylinder pressure and heat release rate.

The addition of H_2 at 75% and 100% loads gradually increased the PCP and PHRR with slightly advance in phasing. The H_2 combustion efficiency is improved due to the larger diesel plume and fast turbulent combustion of H_2.

Cyclic variability tends to increase with the increase of H_2 enrichment at low loads. The cyclic variability reduces at higher load operations. The amount of injected diesel fuel has influences on the combustion stability from one cycle to another cycle. However, the impact of H_2 addition on COV of IMEP is not considerable at tested conditions in this work.

5. References

[1] F.H. Ma, Y. Wang, H.Q. Liu, Y. Li, J.J. Wang, S.F. Ding, Effects of hydrogen addition on cycle-by-cycle variation in a learn burn natural gas spark-ignition engine, Int. J. Hydrogen Energy 33 (2008) 391-402.
[2] J. Demuynck, M. De Paepe, H. Huisseune, R. Sierens, J. Vancoillie, S. Verhelst, Investigation of the influence of engine settings on the heat flux in a hydrogen- and methane-fueled spark ignition engine, Appl. Thermal Energy 31 (2011) 1220-1228.
[3] E.T.N. Kawahara, Z. Piao, S. Fujita, Hydrogen Combustion and Exhaust Emissions Ignited with Diesel Oil in a Dual Fuel Engine (2001) SAE TECH Paper 2001-01-3503.
[4] Das LM. Hydrogen engines: a review of the past and a look into the future. Int J Hydrogen Energy 1990;15:425-43.
[5] Bauer CG, Forest TW. Effect of hydrogen addition on the performance of methane-fueled vehicles, part I: effect on S.I.engine performance. Int J Hydrogen Energy 2001;26:55-70.
[6] Li H, Karim GA. "An experimental investigation of S.I. engine operations on lean gaseous fuels mixtures". SAE Paper 2005- 01-3765, 2005.
[7] Topinka J, Gerty JB. Heywood, Knock behavior of a lean-burn, H2 and CO-enhanced, SI gasoline engine concept. SAE Paper 2004-01-0975, 2004.
[8] Ma F, Wang Y, Liu H, Li Y, Wang J, Zhao S. Experimental study on thermal efficiency and emission characteristics of a lean burn hydrogen enriched natural gas engine. Int J Hydrogen Energy 2007;32:5067-75.
[9] Heywood JB. Internal Combustion Engine Fundamentals. New York: McGraw Hill International Editions; 1988.
[10] Das LM. Near-term introduction of hydrogen engines for automotive and agricultural application. Int. J. Hydrogen energy 2002;27:479–87.
[11] Miyamoto T, Mikami M, Kojima N, Kabashima H, Urata Y. Effect of hydrogen fraction in intake mixture on combustion and exhaust emission characteristics of a diesel engine. 2009. SAE Paper No. 2009-24-0086.
[12] Senthil Kumar M, Ramesh A, Nagalingam B. Use of hydrogen to enhance the performance of a vegetable oil fueled compression ignition engine. Int J Hydrogen Energy 2003;28: 1143–54.
[13] Tsujimura T, Mikami S, Achiha N, Takunaga Y, Senda J, Fujimoto H. A study of direct injection diesel engine fueled with hydrogen. SAE 2003-01-0761; 2003.
[14] Saravanan N, Nagarajan G. Combustion analysis on a DI diesel engine with hydrogen in dual fuel mode. Fuel 2008;87:3591-9.
[15] Masood M, Ishrat M, Reddy A. Computational combustion and emission analysis of hydrogen-diesel blends with experimental verification. Int J Hydrogen Energy 2007;32:2539-47.
[16] Christodoulou F, Megaritis A. Experimental investigation of the effects of separate hydrogen and nitrogen addition on the emissions and combustion of a diesel engine. Int. J. Hydrogen Energy 2013;38:10126-40.
[17] Saravanan N, Nagarajan G. Experimental investigation on performance and emission characteristics of dual fuel DI diesel engine with hydrogen fuel. 2009. SAE Paper No. 2009-26-032.
[18] Bika AS, Franklin LM, Kittelson DB. Emissions effects of hydrogen as a Supplemental fuel with diesel and biodiesel. 2008. SAE Paper No. 2008-01-0648.
[19] Miyamoto T, Mikami M, Kojima N, Kabashima H, Urata Y. Effect of hydrogen fraction in intake mixture on combustion and exhaust emission characteristics of a diesel engine. 2009. SAE Paper No. 2009-24-0086.
[20] Bose PK, Maji D. An experimental investigation on engine performance and emissions of a single cylinder diesel engine using hydrogen as inducted fuel and diesel as injected fuel with exhaust gas recirculation. Int. J. Hydrogen Energy 2009;34:4847-54.
[21] Saravanan N, Nagarajan G. An experimental investigation of hydrogen-enriched air induction in a diesel engine system. Int. J. Hydrogen Energy 2008;33:1769-75.
[22] Kumar MS, Ramesh A, Nagalingam B., "Use of hydrogen to enhance the performance of a vegetable oil fueled compression ignition engine", Int. J. Hydrogen Energy 2003; 10:1143-54.
[23] Gatts T, Liu SY, Liew C, Ralston B, Bell C, Li HL. An experimental investigation of incomplete combustion of gaseous fuels of a heavy-duty diesel engine supplemented with hydrogen and natural gas. Int J Hydrogen Energy 2012;37:7848–59.
[24] Karim GA, "The dual fuel engine of the compression ignition type-prospects, problems and solutions- a review; 1983, SAE Paper 831073, 3. pp. 569-77.
[25] Verhelst S, Woolley R, Lawes M, Sierens R., "Laminar and unstable burning velocities and

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Markstein lengths of hydrogen-air mixtures at engine-like conditions", Int. J. Hydrogen Energy 2005; 30:209-16.