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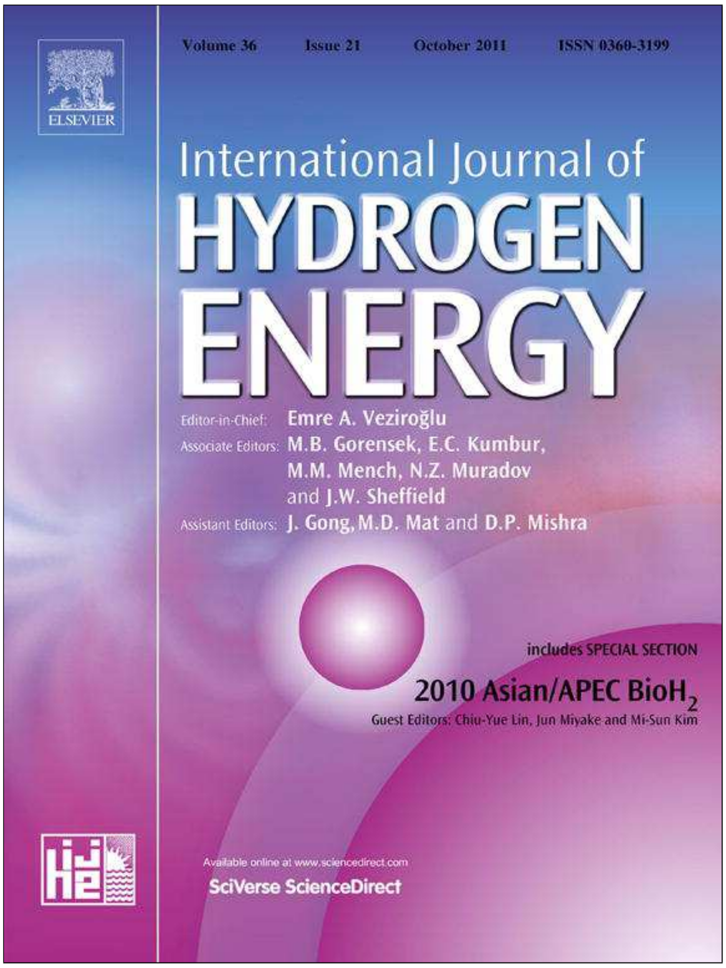
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An experimental study on the effect of hydrogen enrichment on diesel fueled HCCI combustion

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ABSTRACT

This paper experimentally investigates the influence of hydrogen enrichment on the combustion and emission characteristics of a diesel HCCI engine using a modified Cooperative Fuel Research (CFR) engine. Three fuels, n-heptane and two middle distillates with cetane numbers of 46.6 and 36.6, are studied.

The results show that hydrogen enrichment retards the combustion phasing and reduces the combustion duration of a diesel HCCI engine. Besides, hydrogen enrichment increases the power output and fuel conversion efficiency, and improves the combustion stability. However, hydrogen enrichment may narrow the operational compression ratio range and increase the knocking tendency. Both the overall indicated specific CO emissions (isCO) and CO emissions per unit burned diesel fuel mass are reduced by hydrogen enrichment. Although hydrogen enrichment decreases the overall indicated specific unburned hydrocarbon emissions (isHC), it does not significantly affect the HC emissions per unit burned diesel fuel mass.

1. Introduction

Homogeneous charge compression ignition (HCCI) combustion was initially introduced about 30 years ago [1–3]. It is attracting more and more attention from scientists and engineers recently due to its potential to virtually eliminate NO_x and particulate matter (PM) emissions from engine combustion. The disadvantages of HCCI combustion are high carbon monoxide (CO) and unburned hydrocarbon (HC) emissions, along with high peak pressures and high heat release rates [4]. It is of great importance to control HCCI combustion so that the advantages can be maintained while the drawbacks are overcome as much as possible. Since it has been well known that HCCI combustion is mainly controlled by chemical kinetics [3], appropriate fuels that allow an HCCI engine to operate at optimal conditions are key to the development of the technology. Numerous fuels have been investigated for HCCI combustion [5]. For diesel HCCI combustion, it has been shown that fuels with a relatively low cetane number and short combustion duration are desirable for higher fuel efficiency and lower pollutant emissions [6–8].

Hydrogen is a clean fuel because its combustion does not produce carbon dioxide (CO_2), CO, HC and PM. Therefore, the potential for hydrogen as a fuel for internal combustion engines has been studied by many researchers. The use of hydrogen in spark ignition (SI) engines has been reviewed by

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Karim [9], White et al. [10] and Verhelst et al. [11]. It has been shown that hydrogen is an excellent fuel for satisfactory performance in SI engine applications, although there are various challenging issues [12]. Many studies have also contributed to hydrogen fueled HCCI combustion and showed that fuel efficiency obtained in a hydrogen fueled HCCI engine was significantly higher than that obtained when in a conventional diesel-fueled engine [13–18].

Unfortunately hydrogen is an energy carrier and has to be obtained from other hydrocarbon fuels or water. The use of pure hydrogen as a primary fuel for internal combustion engines is too expensive. Alternatively, hydrogen enrichment, i.e. use of a small amount of hydrogen as additive to conventional fossil fuels, is a more practical way to use hydrogen. Therefore, many fundamental and application studies have been conducted for hydrogen enrichment combustion. It has been shown that hydrogen enrichment helps to extend the lean flammability limits and increase the flame speeds of hydrocarbon fuels [19–25]. The addition of hydrogen to an SI engine has been shown to extend the lean operational limits and thus allow an SI engine to operate at leaner conditions with reduced NO_x emissions [12,26,27].

The effect of hydrogen enrichment on HCCI combustion has also been investigated by many researchers in the past decade. Shudo and collaborators [28–30] investigated the effect of hydrogen or reformed gas enrichment on HCCI combustion of dimethyl ether and found that the addition of hydrogen retarded the combustion phasing. Hydrogen enrichment of natural gas fueled HCCI combustion was also investigated by several groups [31–34]. The studies showed that hydrogen enrichment advanced combustion phasing and extended the lean operational limits of a natural gas fueled HCCI engine. Recently, Checkel and collaborators [33–37] investigated the effect of reformer gas (hydrogen rich gas) enrichment on iso-octane and n-heptane fueled HCCI combustion. For each fuel, the experiment was conducted at a fixed compression ratio. They observed that the effect of reformer gas enrichment on combustion phasing depended on the octane number of the base fuel. For lower octane number fuels, the addition of reformer gas retarded combustion phasing, while the effect on higher octane number fuels varied with the change in intake temperature.

Diesel engines are widely used in transportation and off-road vehicles due to their high power density and fuel conversion efficiency. The advantages of HCCI combustion make it a potential alternative combustion mode to conventional diesel combustion [38]. Diesel fuels usually have two-stage combustion, a low temperature heat release (LTHR) stage that initiates the ignition process and a primary high temperature heat release (HTHR) stage. Appropriate combustion phasing and reasonable combustion duration are crucial for a diesel HCCI engine to obtain higher fuel conversion efficiency and lower pollutant emissions [6,7]. Hydrogen is a fuel with higher autoignition temperature [10] which may help to control the combustion phasing of a diesel engine. Meanwhile, hydrogen has higher flame speed and lower extinction limits [18–25], suggesting that the addition of hydrogen may enhance the primary high temperature combustion process and thus reduce the combustion duration of a diesel HCCI engine. Therefore, it is of great interest to investigate the effect of hydrogen enrichment on the combustion and emission characteristics of a diesel HCCI engine. Although the studies of Refs. [33–37] investigated the effect of reformer gas enrichment on an HCCI engine fueled by primary reference fuels, the experiments and simulations were conducted at very limited conditions. The information on the effect of hydrogen enriched diesel HCCI combustion is very limited.

In this paper, the effect of hydrogen enrichment on the combustion and emission performances of a diesel HCCI engine was experimentally investigated. One primary diesel reference fuel and two real diesel fuels were tested. The experiments were conducted over a wide range of operational conditions. The paper starts with the introduction of experimental setup and procedure, followed by the presentation of results and discussion. Finally, concluding remarks are provided.

2. Engine setup and experimental procedure

2.1. *Engine setup*

A Cooperative Fuel Research (CFR) engine was used for the experiments. It is a single-cylinder, four-stroke, variable compression ratio engine. The basic engine specifications are listed in Table 1.

The engine was coupled to an eddy current dynamometer for measuring engine load and controlling engine speed. A variable speed AC motor, coupled to an over-drive clutch, was used to motor the engine before initiating HCCI combustion and to maintain engine speed when combustion was unstable or the power produced was not sufficient to maintain the desired engine speed. The engine setup was modified from the standard CFR configuration by the addition of an air-assisted port fuel injection system and the hardware to control important engine parameters, such as intake temperature, air/fuel ratio, intake and exhaust pressures, and exhaust gas recirculation (EGR).

A port fuel injector was used to atomize the fuels just upstream of the intake port. Compressed air, taken from the intake air after the mass flow meter, was used as blast air to improve the atomization process. For this study, the fuel and air-blast pressures were maintained at 500 kPa and 200 kPa, respectively. The timing and duration of the air-blast relative to the fuel injection was optimized to minimize emissions. This fuel injector produces droplets with approximately 15 μm

Table 1 – Engine specifications.

Cylinder bore	8.255 cm
Stroke	11.43 cm
Displacement volume	611.7 cm^3
Connection road length	25.4 cm
Compression ratio	6–16
Combustion chamber	Pancake shape, flat top piston
Intake valve open	10 °CA ATDC*
Intake valve close	34 °CA ABDC
Exhaust valve open	40 °CA BBDC
Exhaust valve close	5 °CA ATDC
Fuel system	Air-assisted port fuel injection

* ATDC: after top dead center.

Sauter mean diameter under the conditions used in this study. A heated section was added directly downstream of the fuel injector to increase the fuel spray temperature and partially vaporize the diesel fuels before they entered the cylinder. This helped to improve the fuel–air mixing process during the compression stroke. The exit temperature of the fuel vaporizer (T_{vap}) was maintained at a constant value of 220 °C in the experiments. Further details about the engine setup and the fuel injector can be found elsewhere [7,8].

The intake air was provided from a pressurized dry air source. The intake air flow was measured by a mass flow meter before entering the intake surge tank, where the air was mixed with recycled exhaust gases. The mixture of EGR and air then passed through a lengthy intake system (about 2 m), which provided additional time to achieve a homogeneous mixture of air and EGR. The mixture was cooled by a heat exchanger whose coolant was heated to just below the intake air temperature to avoid water vapor condensation. An intake heater was also used to maintain the intake mixture at the desired temperature.

Hydrogen was introduced to the intake port after the diesel fuel injector. The mass flow rate of hydrogen was measured by a flow meter prior to its introduction to the intake port. The intake mixture temperature (T_{mix}) was measured in the intake port downstream of the fuel vaporizer and the introduction point of hydrogen and was maintained at a constant value of 75 °C in the experiments.

Air/fuel mass ratio (AFR) was calculated based on the air mass flow rate measured upstream of the intake surge tank and the measured flow rates of diesel fuel and hydrogen. The stoichiometric air/fuel ratio was calculated based on the fraction of hydrogen in the fuel mixture and the measured H/C ratio of the diesel fuel under investigation.

The engine exhaust was routed to an exhaust surge tank equipped with a safety rupture disk. A back pressure valve was used to maintain the exhaust pressure slightly above the intake pressure to provide the pressure difference needed to recycle exhaust gases to the intake stream. The EGR rate was defined as the ratio of the CO_2 volume fraction in the intake mixture to that in the exhaust.

An engine data acquisition and control system (Sakor Technologies Inc., DynoLab™) was used to acquire temperatures, pressures, and flow rates of intake air, diesel fuel and hydrogen. Exhaust gas composition (NO_x, HC, CO, CO_2, O_2) was measured using an emission analyzer (California Analytical Instruments, 600 series). Indicated specific emissions were calculated by using the direct exhaust emission measurement and assuming equal mass flow rates for intake and exhaust but considering the water removal from the exhaust samples.

Table 3 – Engine parameters.

Engine speed	900 rpm
Intake pressure	150 kPa
Exhaust pressure	170 kPa
Intake mixture temperature (T_{mix})	75 °C
Vaporizer exit temperature (T_{vap})	220 °C
(EGR, λ)	(60%, 1.2), (0%, 3.5)
Hydrogen fraction (α_{H_2})	0–15%
Compression ratio	Varied

A water-cooled pressure transducer (Kistler Corp., model 6041A) flush-mounted in the cylinder head was used to measure cylinder pressure. A real-time combustion analysis system (AVL LIST GmbH, IndiModule) was used to collect cylinder pressure data with 0.2 °CA resolution at each test condition. The pressure data for three hundred consecutive engine cycles were collected for each operational case. AVL's library function for fast heat release calculation (Thermodynamics1) was used to calculate heat release rate. The model is based on a single zone model with air as working fluid and heat transfer to cylinder wall being neglected. All combustion-related parameters were first calculated for individual cycles and then averaged for 300 cycles, including indicated mean effective pressure (IMEP), coefficient of variation of IMEP (COV_{IMEP}), and the maximum pressure rise rate ($dP/d\theta_{max}$), etc.

2.2. *Experiment conditions and procedure*

n-Heptane is a primary diesel reference fuel. Therefore, it was selected as the first fuel in this paper. In addition, two Canadian Oilsands derived fuels with cetane numbers of 36.6 (OS-CN36) and 46.6 (OS-ULSD), respectively, were also investigated. Giving the cetane number of 54 for n-heptane, the fuels investigated cover a wide range in terms of cetane number. The properties of the three fuels are listed in Table 2.

The mass fraction of hydrogen added is defined as

$$\alpha_{H_2} = \frac{m_{H_2}}{m_{diesel} + m_{H_2}} \quad (1)$$

where m_{H_2} and m_{diesel} are the mass flow rates of hydrogen and the diesel fuel (n-heptane, OS-ULSD or OS-CN36), respectively.

For each investigated fuel, two combinations of EGR rate and relative air/fuel ratio (λ), defined as the ratio of real air/fuel ratio to the stoichiometric air/fuel ratio, were studied, with one having EGR = 60% and λ = 1.2 and the other having EGR = 0% and λ= 3.5. For each case, the experiment started with the base fuel and then the hydrogen fraction was

Table 2 – Fuel properties.

Fuel	Liquid density, g/cm^3	Cetane number	C/H ratio (mass)	Boiling point, °C	10% Distillation temperature (T10), °C	90% Distillation temperature (T90), °C	Aromatics (%m)
n-Heptane	0.684	54	5.25	98.42	–	–	–
OS-ULSD	0.844	46.6	6.42	–	177	338	18.4
OS-CN36	0.835	36.6	6.48	–	158	291	27.6

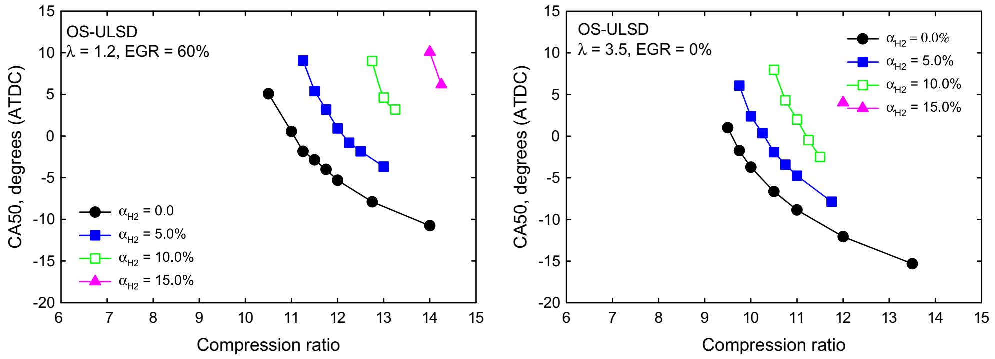

Fig. 1 – Effect of hydrogen enrichment on combustion phasing (CA50).

gradually increased. At each hydrogen fraction, a compression ratio sweep was conducted, while all other engine parameters were held constant. The low and high limits for compression ratio sweep were determined by a COV_{IMEP} of 5% and a maximum pressure rise rate of 10 bar/°CA, respectively. At each compression ratio, the data from 300 cycles were taken and averaged.

The maximum investigated hydrogen fraction was 15.0% for n-heptane and OS-ULSD, while it was 10.0% and 2.5% for OS-CN36 at the two λ–EGR combinations of 1.2–60% and 3.5–0%, respectively, because stable combustion condition could not be obtained at higher hydrogen fractions for OS-CN36.

The detailed engine parameters used are listed in Table 3.

3. Results and discussion

Due to the space limit, only the data for fuel OS-ULSD are presented below for most phenomena. Unless explicitly indicated, other two fuels have qualitatively similar results. Complete set of data for all three investigated fuels can be found from Supplementary material.

3.1. *Combustion performance*

Combustion phasing is an important parameter that directly affects fuel conversion efficiency of an HCCI engine. In this paper, combustion phasing is represented by CA50, the crank angle at which 50% of total heat release is reached. The highest cycle efficiency of a diesel engine is usually obtained when heat release happens near top dead center (TDC) [39]. If combustion phasing is overly advanced or retarded relative to TDC, cycle efficiency will be reduced. Therefore, we first examine how hydrogen enrichment affects combustion phasing for the three investigated fuels.

Fig. 1 shows the variations of CA50 as a function of compression ratio for fuel OS-ULSD. It is observed that CA50 is retarded with increasing hydrogen fraction at a constant compression ratio, suggesting that hydrogen enrichment retards the combustion phasing of a diesel HCCI engine. This is qualitatively consistent with the results obtained by Checkel and collaborators [33–37] for the effect of reformer gas (hydrogen rich gas) enrichment on HCCI combustion of low octane number fuels.

A fuel additive may vary the combustion phasing of a diesel HCCI engine by three possible effects, thermal effect, dilution effect and chemical effect. Thermal effect is due to the variation in the thermal properties by the fuel additive. Since the specific heat of hydrogen is smaller than that of a diesel fuel, the thermal effect of hydrogen addition should not have retarded the combustion phasing of a diesel HCCI engine. Therefore, the retardation of combustion phasing observed in Fig. 1 can only be caused by dilution and chemical effects. Dilution effect is caused by the variation in the concentrations of reactants and chemical effect is due to the participation of hydrogen in reactions. It has been shown that in an HCCI engine, ignition is controlled by H_2O_2 decomposition [4]. Any variation that gets the reactive fuel/air mixture to reach the H_2O_2 decomposition temperature (about 1000 K) later will retard the combustion phasing. For diesel fuel

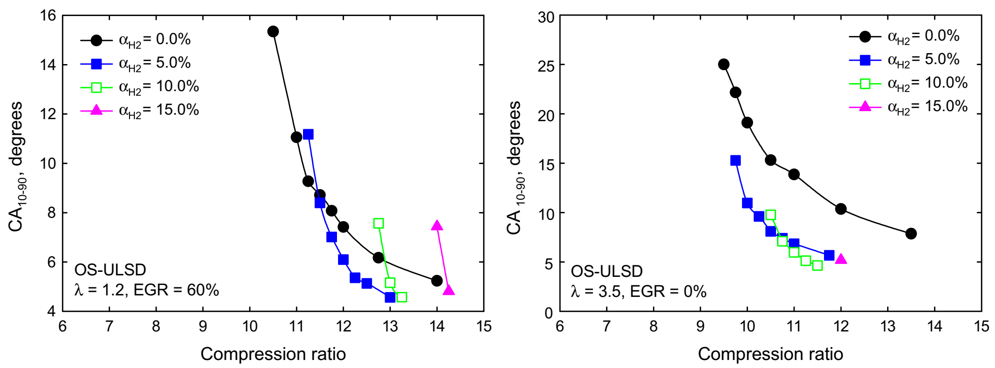

Fig. 2 – Variation of combustion duration as a function of compression ratio.

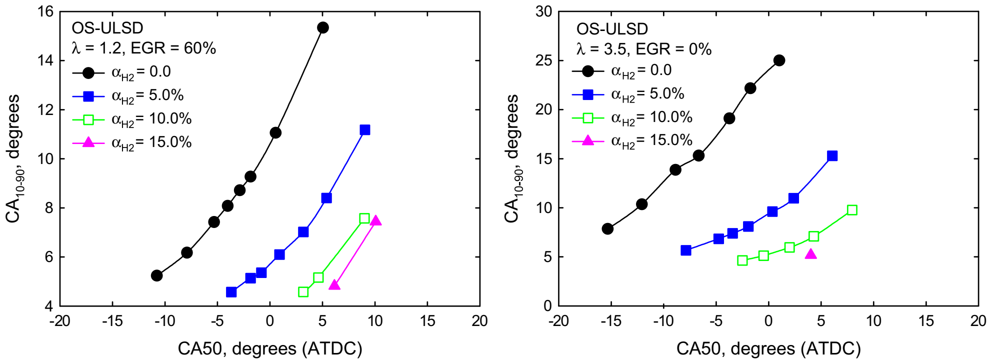

Fig. 3 – Variation of combustion duration as a function of CA50.

combustion, the energy needed for the mixture to reach the H_2O_2 decomposition temperature mainly comes from the heat release due to some low temperature hydrocarbon oxidation reactions that are usually initiated by hydrogen abstraction reactions. Since the combustion of hydrogen requires less oxygen than most diesel fuels, the concentrations of both diesel fuel and oxygen are decreased when hydrogen is added. This slows down the reaction process of diesel fuel during the low temperature stage and therefore retards the combustion phasing. This is how hydrogen addition retards the combustion phasing by dilution effect. Meanwhile, hydrogen addition also retards the combustion phasing due to chemical effect, because hydrogen participates in some reactions during the low temperature stage, especially the reaction $H_2 + OH = H_2O + H$. Although the heat release from this reaction is negligible and only a small amount of OH is consumed by this reaction during the low temperature stage, it clearly slows down the oxidation reactions of diesel fuel and reduces the overall heat release rate. This is because OH is involved in many diesel fuel oxidation reactions during the low temperature stage. More or less OH radicals in the mixture will enhance or slow down the low temperature chemical kinetics process. Therefore, both dilution and chemical effects of hydrogen addition reduce the heat release rate during the low temperature stage. Although not shown, the heat release profiles do show that the heat release rate during the low temperature stage decreases with an increase in hydrogen fraction when other conditions are constant. The reduction in heat release rate due to dilution and chemical effects during the low temperature stage retards the combustion phasing of a diesel HCCI engine when hydrogen is added.

It is also noted from Fig. 1 that the collected compression ratio data range narrows with an increase in hydrogen fraction. As mentioned in the experimental procedure section, the range of compression ratios investigated was limited by the maximum allowable coefficient of variation of indicated mean effective pressure and the maximum allowable pressure rise rate for low and high compression ratios, respectively. It was found in the experiment that the low compression ratio limit increased and the high compression ratio limit decreased, when the hydrogen fraction was increased. Therefore, hydrogen enrichment narrows the acceptable compression ratio window. This may be a disadvantage of hydrogen enrichment.

Combustion duration is another parameter that significantly affects the fuel conversion efficiency of an HCCI engine. Fig. 2 shows the variation of combustion duration as a function of compression ratio for OS-ULSD. Other fuels have qualitatively similar results. In this paper, combustion duration is defined as CA_{10-90}, the difference between two crank angles at which 10% and 90% of total energy release are reached, respectively. Combustion duration increases with a decrease in compression ratio at a constant hydrogen fraction for all the investigated fuels. The increase rate in combustion duration with a decrease in compression ratio becomes higher when hydrogen fraction is increased. This implies that hydrogen enrichment increases the sensitivity of combustion duration to compression ratio, which causes hydrogen enrichment to increase combustion duration at lower compression ratios, while it reduces combustion duration at higher compression ratios.

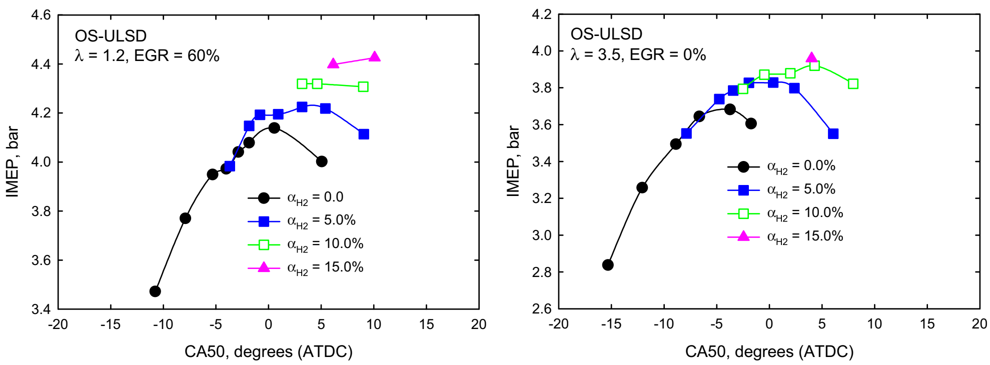

Fig. 4 – Variation of IMEP as a function of CA50.

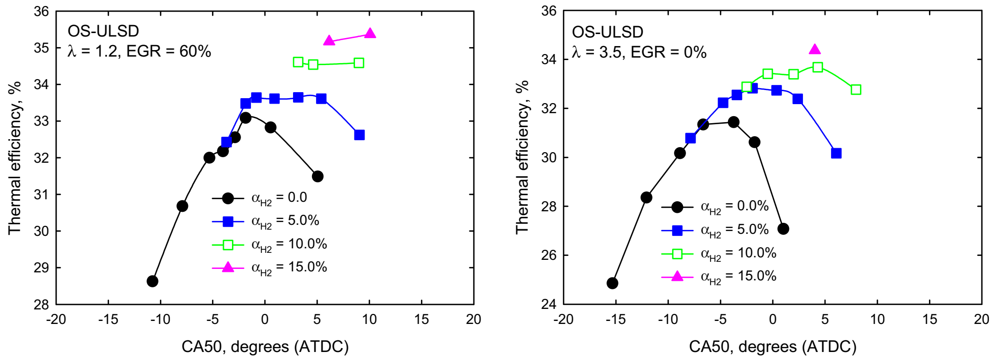

Fig. 5 – Variation of thermal efficiency as a function of CA50.

The combustion duration data are re-plotted as a function of CA50 in Fig. 3, since combustion phasing is a more meaningful parameter for HCCI combustion. It is revealed that hydrogen enrichment reduces combustion duration at a constant CA50. This is caused by three possible reasons. Firstly, since hydrogen enrichment retards combustion phasing, as shown in Fig. 1, higher compression ratio is required to keep CA50 constant when hydrogen is added. Higher compression ratio results in lower combustion duration, as shown in Fig. 2. Secondly, hydrogen enrichment intensifies the reactivity of the high temperature combustion stage, which is usually controlled by the chain branching reaction $H + O_2 = O + OH$ [4]. Hydrogen enrichment enhances the rate of this reaction and thus intensifies the heat release rate during the high temperature heat release stage and helps reduce the combustion duration. The heat release profiles (although not shown) do show that the maximum heat release rate during the high temperature stage increases with an increase in hydrogen fraction. Thirdly, hydrogen enrichment may help reduce combustion duration due to the decrease in the total mass of diesel fuel not well vaporized. Although a port fuel injector was used to atomize the fuels upstream of the intake port, part of the liquid fuel might not be fully vaporized for OS-ULSD and OS-CN36 before entering the cylinder due to their low volatility, as shown by the higher 10% and 90% distillation temperatures in Table 2. Therefore, part of liquid droplets might enter cylinder and be evaporated during compression and even combustion processes, which might result in longer combustion duration. When hydrogen was added, the fraction of liquid fuel decreased and therefore the combustion duration was reduced. This effect should be negligible for n-heptane, since most liquid fuel has been vaporized before entering cylinder due to the lower boiling point.

The variations in combustion phasing and combustion duration lead to changes in engine power output. Figs. 4 and 5 plot the variations of indicated mean effective pressure (IMEP) and thermal efficiency, respectively, as a function of CA50 for OS-ULSD. Thermal efficiency is defined as the ratio of indicated work to total energy input to cylinder per engine cycle. Generally IMEP and thermal efficiency increase with an increase in hydrogen fraction at a constant CA50, except for cases when the combustion phasing is overly advanced. This suggests that diesel HCCI combustion does benefit from hydrogen enrichment in terms of power output and fuel conversion efficiency. The increase in IMEP and thermal efficiency with hydrogen enrichment is caused primarily by the retarded combustion phasing and reduced combustion duration. Hydrogen enrichment retards combustion phasing and thus allows a diesel HCCI engine to operate at a higher compression ratio which increases the power output of an engine. The reduced combustion duration also helps improve power output and thermal efficiency since it allows more energy to be released near TDC and results in a more complete combustion (lower CO and HC emissions), as will be shown later.

When CA50 changes from advanced to retarded due to reducing compression ratio at a constant hydrogen fraction, IMEP and thermal efficiency first increase and then decrease. For a given hydrogen fraction, IMEP and thermal efficiency reach their maximum at a certain CA50. When hydrogen fraction is increased, the CA50 at which IMEP and thermal efficiency reach their maximum is more retarded and the maximum IMEP and thermal efficiency increase. Fig. 6 illustrates the variations of normalized maximum thermal efficiency, which is defined as the ratio of the maximum thermal efficiency of a hydrogen enriched case to that of the corresponding case without hydrogen enrichment, for all three investigated fuels as a function of hydrogen fraction. It is noted that for all three fuels, the maximum thermal efficiency at $\lambda = 3.5$ and EGR = 0% always increases faster with an

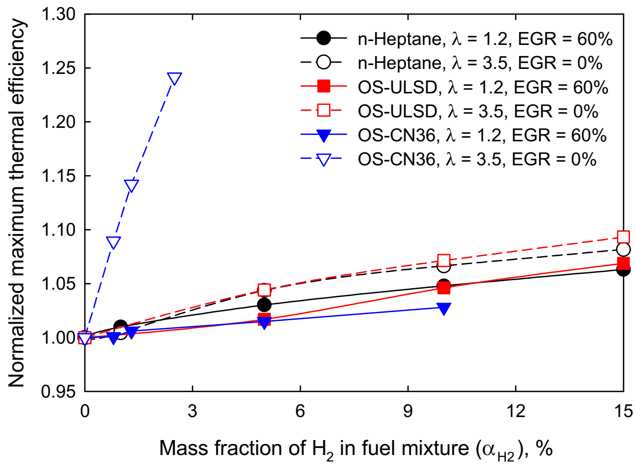

Fig. 6 – Variation of normalized maximum thermal efficiency as a function of hydrogen fraction.

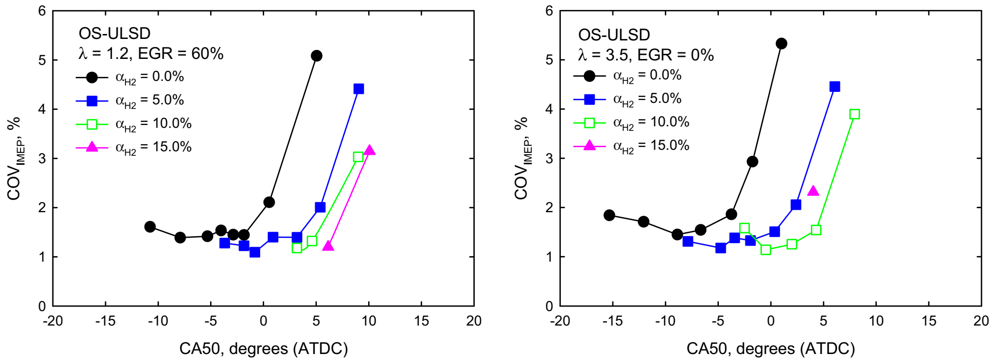

Fig. 7 – Variation of COV_{IMEP} as a function of CA50.

increase in hydrogen fraction than that at $\lambda = 1.2$ and EGR = 60%, suggesting that hydrogen enrichment is more effective for diesel HCCI combustion at an extra lean condition without EGR than at a near-stoichiometric condition with higher EGR rate. This might be due to the fact that an extra lean diesel/air mixture without EGR has a shorter ignition delay than that of a near-stoichiometric mixture with high EGR rate. We can indirectly confirm this by Figs. 4 and 5 which show that the maximum IMEP and thermal efficiency usually happen at a more advanced CA50 for the extra lean mixture without EGR ($\lambda = 3.5$, EGR = 0%) than for the near-stoichiometric mixture with high EGR rate ($\lambda = 1.2$, EGR = 60%). For an extra lean mixture without EGR, hydrogen enrichment significantly retards combustion phasing, which results in a significant improvement in thermal efficiency. On the other hand, ignition happens at a crank angle that is closer to TDC for a near-stoichiometric diesel/air mixture with high EGR rate even when hydrogen is not added. Therefore, hydrogen enrichment is less effective in improving thermal efficiency for near-stoichiometric mixture with high EGR rate in a diesel HCCI engine.

Although not shown, the results also show that n-heptane requires more hydrogen while OS-CN36 only needs a small amount of hydrogen so that the maximum thermal efficiency occurs near TDC. This suggests that higher cetane number fuels require more hydrogen to retard the combustion phasing to the optimal values.

The effect of hydrogen enrichment on coefficient of variation of IMEP (COV_{IMEP}) is shown in Fig. 7 for OS-ULSD. It is found that when CA50 is advanced, the effect of hydrogen enrichment on COV_{IMEP} is negligible. However, COV_{IMEP} usually decreases with an increase in hydrogen fraction when CA50 is retarded beyond TDC. This suggests that hydrogen enrichment improves combustion stability, since it is desirable to operate HCCI engines with a combustion phasing retarded slightly after TDC.

The effect of hydrogen enrichment on the maximum pressure rise rate is displayed in Fig. 8 for OS-ULSD. At a constant CA50, the maximum pressure rise rate increases with an increase in hydrogen fraction. It might be caused by the fact that hydrogen enrichment allows higher compression ratios and enhances the high temperature stage combustion of a diesel fuel, as mentioned before. This suggests that hydrogen enrichment may increase the knocking tendency of a diesel HCCI engine, which is a disadvantage. However, the maximum pressure rise rate decreases when CA50 is retarded at a constant hydrogen fraction. Since hydrogen enrichment allows a diesel engine to operate at a more retarded combustion phasing, the disadvantage of increasing knocking tendency may be moderated by retarding the combustion phasing.

3.2. *Emission characteristics*

One of the disadvantages of HCCI combustion is higher CO and HC emissions. Therefore, it is of great interest to examine the effect of hydrogen enrichment on CO and HC emissions from a diesel HCCI engine.

Figs. 9 and 10 show the indicated specific CO and HC emissions (isCO and isHC), respectively, for OS-ULSD. It is

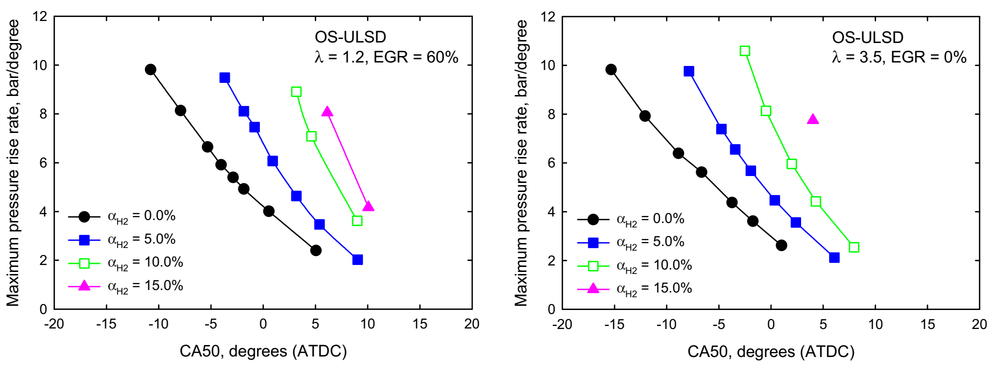

Fig. 8 – Effect of hydrogen enrichment on the maximum pressure rise rate.

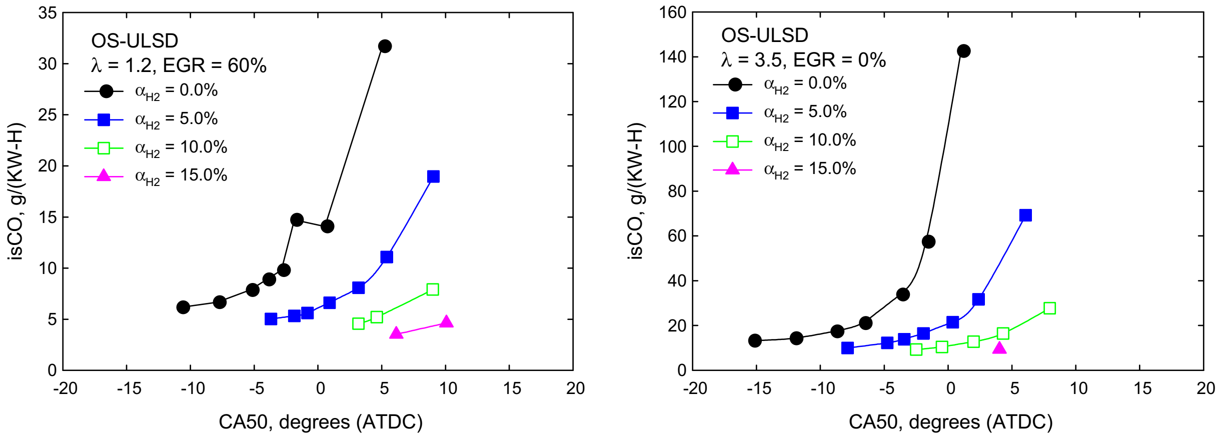

Fig. 9 – Variation of indicated specific CO emissions (isCO) as a function of CA50.

revealed that hydrogen enrichment reduces the indicated specific CO and HC emissions at a constant CA50, especially when the CA50 occurs after TDC. This is a desired result, but it is not surprising, since the percentage of hydrocarbon fuel input to cylinder decreases with an increase in hydrogen fraction. However, we note that this result is not consistent with the observation by Hosseini and Checkel [36] for the effect of reformer gas enrichment on n-heptane HCCI combustion. This difference may be due to the fact that reformer gas used in Ref. [36] contained CO.

It is more interesting to examine if hydrogen enrichment affects the normalized CO and HC emissions based on per unit diesel fuel mass consumed, i.e. the ratio of emitted CO/HC mass to the input diesel mass. Fig. 11 shows that the normalized CO emissions decrease at a constant CA50 with an increase in hydrogen fraction, suggesting that hydrogen enrichment reduces not only the overall CO emissions but also the CO emissions per unit burned diesel fuel mass. This might be owing to the fact that hydrogen enrichment enhances the high temperature kinetics of diesel fuel combustion and thus intensifies the primary CO combustion reaction $CO + OH = CO_2 + H$.

However, Fig. 12 indicates that the normalized HC emissions do not significantly change with a variation in hydrogen fraction for most cases, indicating that the reduction in specific indicated HC emissions in Fig. 10 is primarily due to the decrease in the fraction of diesel input to the cylinder when hydrogen is added. This might be because although hydrogen enrichment enhances the high temperature kinetics, it slows down the low temperature kinetics of diesel fuel combustion, as discussed previously. Although not shown due to space limit, the results actually reveal that the normalized HC emissions slightly increase with an increase in hydrogen fraction for fuel OS-CN36 at $\lambda = 1.2$ and EGR = 60%. It is due to the fact that OS-CN36 is a low cetane number diesel fuel and the combustion phasing for the maximum thermal efficiency is close to TDC without hydrogen enrichment at $\lambda = 1.2$ and EGR = 60%. When hydrogen is added for OS-CN36 at $\lambda = 1.2$ and EGR = 60%, the combustion phasing is further retarded relative to TDC and the time available for completing the combustion process is reduced.

Although hydrogen enrichment does not significantly reduce or even slightly increase the normalized HC emissions, it may still help to improve combustion efficiency, since the absolute value of CO emissions is much higher than that of the corresponding HC emissions at a given operational condition, as shown in Figs. 9–12.

Although due to the space limit, the results for the effect of hydrogen enrichment on NO_x emissions are not shown, it deserves to mention that NO_x emissions do not significantly change or slightly increase at an advanced constant CA50 with an increase in hydrogen fraction. However, NO_x emissions decrease at a retarded CA50 with an increase in hydrogen fraction. This variation in the effect of hydrogen enrichment on NO_x emissions at different CA50s might be caused by the difference in NO_x formation mechanisms. At a retarded CA50

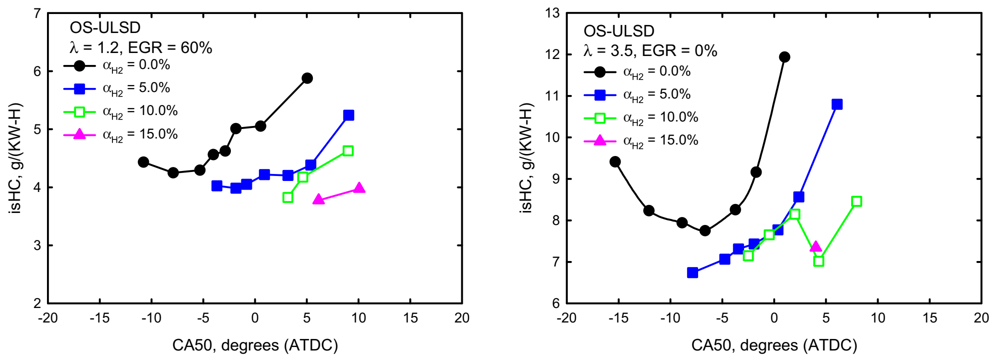

Fig. 10 – Variation of indicated specific HC emissions (isHC) as a function of CA50.

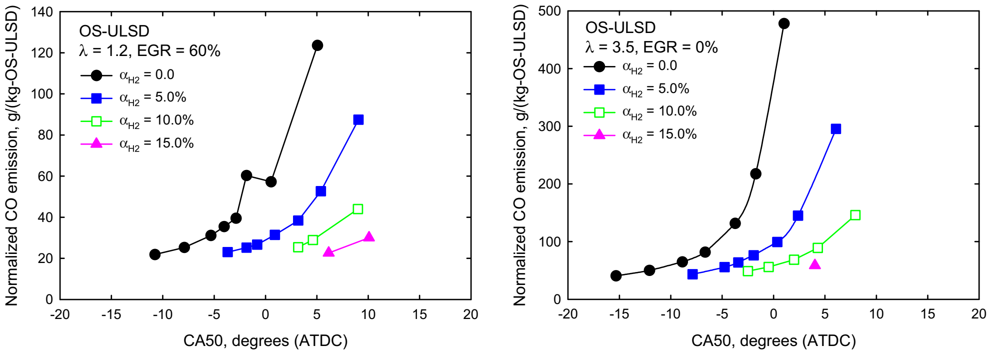

Fig. 11 – Variation of normalized CO emissions as a function of CA50.

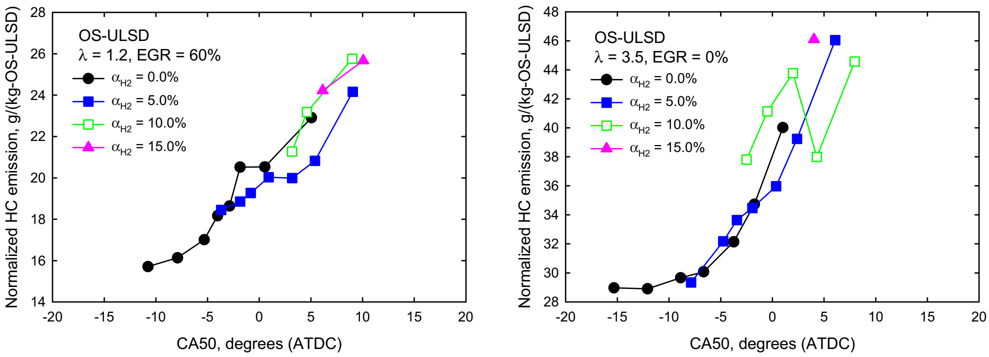

Fig. 12 – Variation of normalized HC emissions as a function of CA50.

(lower compression ratio), NO_x formation in an HCCI engine is primarily due to the N_2O intermediate mechanism and the contribution of the thermal mechanism is negligible [40,41]. For the N_2O intermediate mechanism of NO_x formation, the effect of temperature variation is relatively weak. When hydrogen fraction is increased, the mole fraction of nitrogen is decreased in the intake fuel/air mixture due to the decrease in volume stoichiometric air/fuel ratio, which results in a decreased rate of NO_x formation. On the other hand, at an advanced CA50 (higher compression ratio), both the thermal and N_2O intermediate mechanisms contribute to the NO_x formation [41]. The increase in hydrogen fraction increases cylinder temperature and thus the rate of NO_x formation due to the thermal route, but it might decrease the rate of NO_x formation due to the N_2O intermediate route because of the decrease in nitrogen mole fraction. The result is that NO_x emissions do not significantly change or slightly increase at an advanced CA50 with an increase in hydrogen fraction. Therefore, we can conclude that hydrogen enrichment may decrease or at least does not significantly increase NO_x emissions of a diesel HCCI engine.

4. Conclusions

The effect of hydrogen enrichment on the combustion and emission characteristics of a diesel HCCI engine has been experimentally investigated by using a CFR engine. The base fuels were a primary reference fuel (n-heptane) and two middle distillates with cetane numbers of 36.6 and 46.6. For each fuel, a compression ratio sweep was conducted at each given hydrogen enrichment level and constant engine operating conditions. Following conclusions can be drawn:

(1) Hydrogen enrichment retards combustion phasing and reduces combustion duration of a diesel HCCI engine. As a result, hydrogen enrichment allows a diesel HCCI engine to operate at a higher compression ratio and thus leads to higher power output and fuel conversion efficiency. Higher cetane number fuels require more hydrogen to retard the combustion phasing to the optimal values;
(2) Hydrogen enrichment is more effective for a diesel engine operating at a higher air/fuel ratio without EGR than for that operating at a near-stoichiometric air/fuel ratio with higher EGR rate, in terms of the improvement in thermal efficiency;
(3) Hydrogen enrichment improves the combustion stability of a diesel HCCI engine under retarded combustion phasing;
(4) The operational compression ratio range is narrowed by hydrogen enrichment. Besides, knocking tendency of a diesel HCCI engine may be increased by hydrogen enrichment;
(5) Hydrogen enrichment not only reduces the overall indicated specific CO and HC emissions, but also decreases CO emissions per unit mass of diesel fuel consumption. However, hydrogen enrichment does not significantly affect HC emissions per unit burned diesel fuel mass at a constant CA50;
(6) Indicated specific NO_x emissions decrease or do not significantly change with hydrogen enrichment at a constant CA50.

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Appendix. Supplementary data

Supplementary data related to this article can be found online at doi:10.1016/j.ijhydene.2011.07.143.

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