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The Effect of Hydrogen Addition on a RCCI Engine Performance Fueled with Natural Gas\Diesel Fuel at Low Load Range

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ABSTRACT

The aim of this study is to evaluate the effect of hydrogen addition on a heavy-duty diesel engine performance and emission under RCCI combustion fueled with natural gas and diesel fuel. For this purpose, a single-cylinder heavy-duty diesel engine is used to operate at low-load range from 5.6 to 7.7 bar gross IMEP, constant engine speed, and a fixed amount of diesel fuel. Furthermore, in order to reduce hydrocarbon fuels consumption, hydrogen is gradually added to natural gas while the total fuel energy content is kept to be constant. The results show that by adding hydrogen to natural gas in the RCCI engine at 5.6, 6.3, and 7.7 bar gross IMEP, the hydrogen energy share could be enhanced up to 35.7%, 23.70%, and 9.93%, respectively. And also, the overall hydrocarbon fuels consumption per cycle can be reduced up to 45.23%, 29.26%, and 12.07%, respectively. Although, CO and UHC emissions decrease drastically, NOx increases significantly with the addition of hydrogen compared to RCCI combustion fueled with natural gas and diesel fuel.

1) Introduction

Nowadays, air pollutions in metropolis are mostly caused by internal combustion engines operation. Diesel engines are commonly used in transportation and power generation due to higher combustion efficiency and lower emissions. But today, their application has become more restricted due to more stringent regulation on emissions such as nitrogen oxide (NOx), soot, and etc. Hence, this has enforced the designers and engine manufacturers to reduce their emission levels.

Air pollution has major impact on the environment and human health. Therefore, many research has been carried out in past two decades to bring down engine emissions.

The use of after-treatment devices like diesel particulate filters (DPF), lean NOx trap (LNT), selective catalytic reduction (SCR), etc. [1], have contributed to reducing emissions. But, their operation becomes unfavorable due to high maintenance and installation costs also leading to lower fuel efficiency. Hence, researchers have turned to devise advanced techniques in combustion to bring down the engine emission.

NOx formation increases with high temperatures in the engine combustion chamber. On the other hand, any time delay in the start of combustion (SOC), causes to form a homogeneous mixture before the SOC, which reduces the soot level. Accordingly, new strategies are based on three targets: Soot reduction, high thermal efficiency, and NOx reduction.

These new strategies are classified as premixed low-temperature combustion (PLTC), such as homogeneous charge compression ignition (HCCI) and premixed charge compression ignition (PCCI), and Reactivity- controlled compression ignition (RCCI) [2-6].

In most studies related to RCCI combustion, different fuels such as methanol, natural gas, gasoline, ethanol, and mixtures of gasoline-alcohols as low-reactivity fuels and diesel fuel, biodiesel, dimethyl ether (DME) and gasoline with cetane numbers improver as high reactive fuel have been used [7-10].

Variation of the amount of two different fuels and the start of injection (SOI) timing of high reactive fuel can control the combustion phase and duration in the RCCI combustion strategy.

Recently, the use of natural gas (NG) as a clean and low reactivity fuel in RCCI combustion have been studied extensively [6, 10]. Natural gas is a mixture of methane, ethane, and etc. Since dominate percentage of NG belongs to methane (approximately 89.8%). Therefore, most researches have used methane to represent NG. Due to higher reactivity difference between methane and diesel fuel, the tendency of consuming methane as a low reactive fuel in RCCI engine is gradually increasing [6, 7, 11].

But, researches show that high unburned hydrocarbon (UHC) and carbon monoxide (CO) emission level result from the engine operation under RCCI combustion at low to medium loads and high engine speed due to low reactivity and lower burning rate of NG compared to diesel fuel. Therefore, adding a small amount of hydrogen to NG can reduce CO and UHC emissions due to the higher diffusivity, high flame propagation speed, and a small quenching effects of hydrogen compared to NG [12].

However, the use of hydrogen as an addition to hydrocarbon fuels has some difficulties. Hydrogen has higher flame speed compared to other hydrocarbon fuels such as NG and diesel fuel. Thus, it increases the combustion rate and heat release rate increases, drastically. Consequently, in- cylinder temperature is increased, therefore, NOx emission increases. EGR is one of the appropriate techniques to resolve this problem by controlling in-cylinder temperature. Although the higher hydrogen energy share (HES) is achievable by using EGR, the use of EGR causes to reduce the engine volumetric efficiency. Also, hydrogen has a higher auto-ignition temperature compared to other fossil fuels (585 ºC). Hence, hydrogen cannot be used as sole fuel in internal combustion engines (ICEs) without use of an ignition source such as spark plug (SI engine) or high reactive fuel injection (CI engine). In the RCCI engine fueled with NG/ diesel fuel, hydrogen can be used as an additive to NG due to the presence of a high reactive fuel (i.e. diesel fuel) as a combustion initiation source [13-17].

The present simulation study evaluates the effects of hydrogen addition to NG without the use of exhaust gas recirculation (EGR) in a heavy-duty diesel engine under RCCI combustion fueled with NG/ diesel fuel at low-load. The main objective of this study is the improvement of engine performance, the reduction of the engine emission, and the reduction of hydrocarbon fuel consumption without any the engine power losses. Moreover, to examine the effects of hydrogen addition on some important combustion characteristics and engine performance.

2) Computational Model

In this paper, AVL FIRE CFD tool is employed in order to simulate the computational fluid dynamics of internal combustion engine, as well as, ESE diesel tool is utilized for designing of the combustion chamber of Caterpillar engine which the relevant specifications are reported in Table 1.

Splitter et al. [18] offer an optimized bathtub piston bowl for RCCI Combustion strategy with compression ratio equal to 14.88:1. Based on Splitter et al. work [18], the same proposition to bathtub piston bowl profile is used which is shown in Fig. 1.

Table 1: Engine and fuel injector specs[5]

Caterpillar 3401E SCOTE	
Number of Cylinders	1
Compression ratio	14.88:1
Bore x Stroke	137.2mm x 165.1mm
Displacement Volume(cm3)	403
Intake valve closing (IVC)	-143 deg. ATDC
Exhaust valve opening (EVO)	130 deg. ATDC
Swirl ratio	0.7
Piston bowl type	Bathtub
Common rail diesel fuel injector	
Injector holes	6
Injector hole diameter	250μm
Included spray angle	145 deg.
Injection pressure	500 bar
Injection duration	15 crank angle degree
Natural gas and hydrogen addition	Intake port injection

ATDC: After Top Dead Center

Based on the diesel fuel injection data mentioned in Table 1, each fuel injector has six holes on its nozzle.

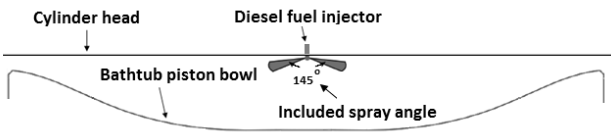

Figure 1: Cross sectional view of the engine combustion chamber geometry [6, 18]

Thus, the combustion chamber is considered to be as a 60° sector, since, it would simplify the implemented simulation model regarding to symmetrical geometry of the engine combustion chamber. According to this matter, one- sixth of entire the engine combustion chamber is meshed and is depicted in Fig. 2.

This paper considers six set of operating conditions of 2.44 liter Caterpillar heavy- duty diesel engine (three Low- load conditions and three Mid- Load conditions) in order to evaluate RCCI combustion strategy. Based on Walker et al. work [5], the relevant sets of experimental results are presented in Table 2. These experimental data have been used to validate the current simulation.

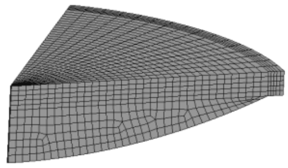

Figure 2: A 60° sector 3D computational grid [6]

Following Walker et al. work [5], as shown in Fig. 3, natural gas as low reactivity fuel is injected in the intake port while diesel fuel as high reactivity fuel through high pressure fuel injector is directly injected to the combustion chamber.

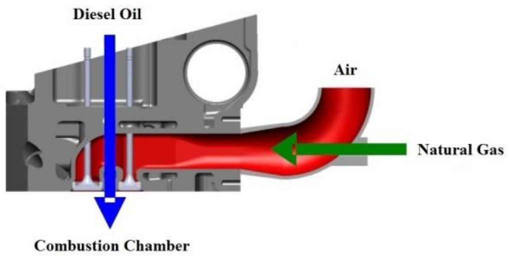

Figure 3: Schematic of the fuels injection into the engine cylinder

In this simulation study, in order to simulate RCCI combustion strategy, CHEMKIN tool has been coupled with the AVL Fire CFD tool. The reduced primary reference fuel (PRF) mechanism is used as a way to predict the chemical reaction between natural gas (i.e. methane) and diesel fuel (i.e. n- heptane). The used reduced PRF mechanism is composed of 76 species and 464 different reactions for this simulation [19].

Indeed, the modeling of diesel fuel spray in the combustion chamber is the most important part of simulation. Hence, several sub-models such as turbulence dispersion model [20], particle interaction model (Coalescence/ Collision) [21], the evaporation model [22], break- up model

[23], and spray- wall interaction model [24] are implemented.

Rigorous validation requires that the responses of the simulation model and experimental results of the real system have the same expected values. Therefore, the simulation and experimental results are comparable if they are obtained under same scenarios (set of input data as Table 2). Thus, the obtained results in the current simulation study are validated with the experimental data which obtained from the engine test under RCCI combustion fueled with NG and diesel fuel without EGR [5] and another simulation study carried out by the present authors [6].

For six mentioned gross IMEPs, based on Walker et al. experimental work [5], as duplicated in Table 2, four important characteristics, namely, the in-cylinder pressure, the HRR, the CA50, the in-cylinder peak pressure, and the PPRR are used to validate the accuracy of RCCI combustion simulation results. In Walker et al. work [5], the RCCI engine under investigation uses natural gas as low reactive fuel and 2007 CERT commercial diesel fuel (CH1.76) as high reactive fuel [5]. Since, the RCCI combustion simulation is carried out using AVL FIRE CFD tool coupled with the CHEMKIN chemistry tool, a PRF mechanism is required to predict the reactions between diesel fuel and natural gas. Also, in the present simulation study, methane (CH_4) as the representative of natural gas with the LHV of 50 MJ/kg and n-heptane (C_7H_{16}) as the representative of diesel fuel with the LHV of 44.6 MJ/kg is considered. Some relevant properties of n-heptane compared to commercial diesel fuel are listed in Table 3 [25].

Table 2. Specified experimental engine operating conditions for validation purpose [5].

	Low-Load range			Mid-Load range		
Gross IMEP (bar)	5.6	6.3	7.7	9.4	11.5	13.5
Intake P (bar)	0.97	1.08	1.32	1.6	1.9	2.2
Intake T (K)	313	313	313	313	313	313
Diesel fuel (mg)	13	13	13	13	13	13
Methane mass (mg)	42	49	62	76	96	108
Total fuel mass (mg)	55	62	75	89	109	121
Diesel fuel SOI (º ATDC)	-35	-39	-42	-45	-48	-51
Equivalence ratio	0.3	0.3	0.3	0.3	0.3	0.3

Table 3. Chemical and physical properties of n-heptane and commercial diesel fuel [25].

Fuel property	N-heptane (C_7H_{16})	Commercial Diesel fuel ($CH_{1.76}$)
LHV (MJ/kg)	44.6	42.6
Cetane Number	56	42
Initial Boiling Point (ºC)	98	174
Temperature 50% Evaporated (ºC)	98	262
Final Boiling Point (ºC)	98	350

N-heptane is a normal chain hydrocarbon fuel. Therefore, based on Ando et al. work [26], in the RCCI engine under investigation which uses a fixed amount of n-heptane in every cycle (i.e. 13mg [5]), the cool flame phenomenon or CA10 occurs due to the direct-injection of n-heptane about 10-20º prior to the start of main combustion at the in-cylinder temperature between 750 and 800 K as depicted in Figs. 4 and 5. It is important to say that the cool flame occurrence slightly advances due to the lower boiling point of n-heptane compared to the commercial diesel fuel as shown in Fig. 5.

After the cool flame occurrence, the heat release diagram is followed by two stages of the fuel energy release [9]. As depicted in Fig. 5, n-heptane as high reactive fuel displays the first stage of the heat release. As expected, at this stage, due to the higher cetane number of n-heptane compared to the commercial diesel fuel, the heat release rate of n-heptane is faster. After this stage, due to the presence of low reactive methane-air mixture in the engine combustion chamber, the second stage of the heat release occurs and the combustion will be extended with reduced heat release rate. The magnified views in Fig. 5 illustrate that when the main combustion occurs, the computational model used has potential to precisely predict the time of the SOC relevant to the experimental results [5]. Although, at the start of main combustion, the rate of the heat release is slower than that of the experimental result [5], but, after a few crank angle degrees, the heat release rate of n-heptane increases rapidly.

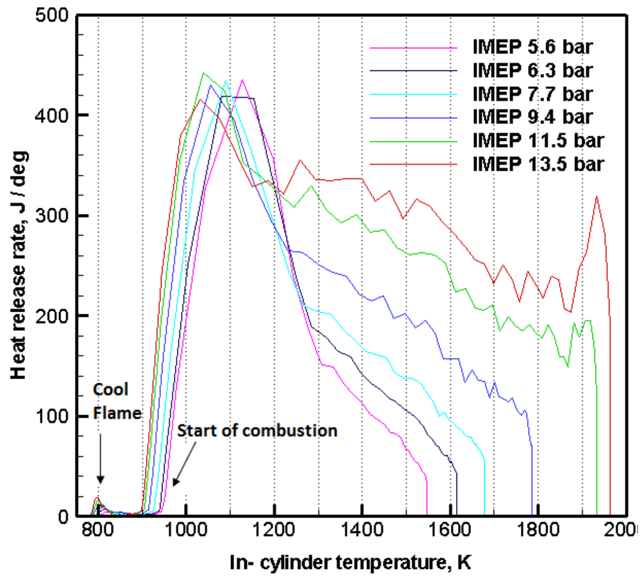

Figure 4. The cool flame phenomenon occurrence [6].

For six mentioned gross IMEPs, the CA50 as an important combustion characteristic was also assessed. Base on the experimental results [5], the CA50 takes place at 0 ºATDC or at the TDC [5]. In the current study, as presented in Table 4, 50% of the total fuel energy releases at about the TDC with a minor deviation which confirms the accurate enough prediction of the HRR by the used computational model. As mentioned in Table 2, based on the engine operating conditions, the diesel fuel mass per each cycle and the intake temperature are constant. At fixed equivalence ratio (i.e. 0.3), the engine load increases only with increasing the intake pressure and the amount of natural gas. At higher gross IMEPs, better thermodynamic characteristics in the RCCI engine combustion chamber resulted from the higher intake pressure leads to earlier start of main combustion. Thus, in the engine Mid-Load range, the CA50 takes place before the TDC.

Moreover, the simulation results for the engine in-cylinder peak pressure and the PPRR are compared with the reported experimental data [5] in Fig. 6. As depicted in this figure, the used simulation model shows an accurate prediction of the engine behavior for the in-cylinder peak pressure. It should be noted that the maximum permissible value of the PPRR in a heavy-duty diesel engine equal is 15 bar/º [9].

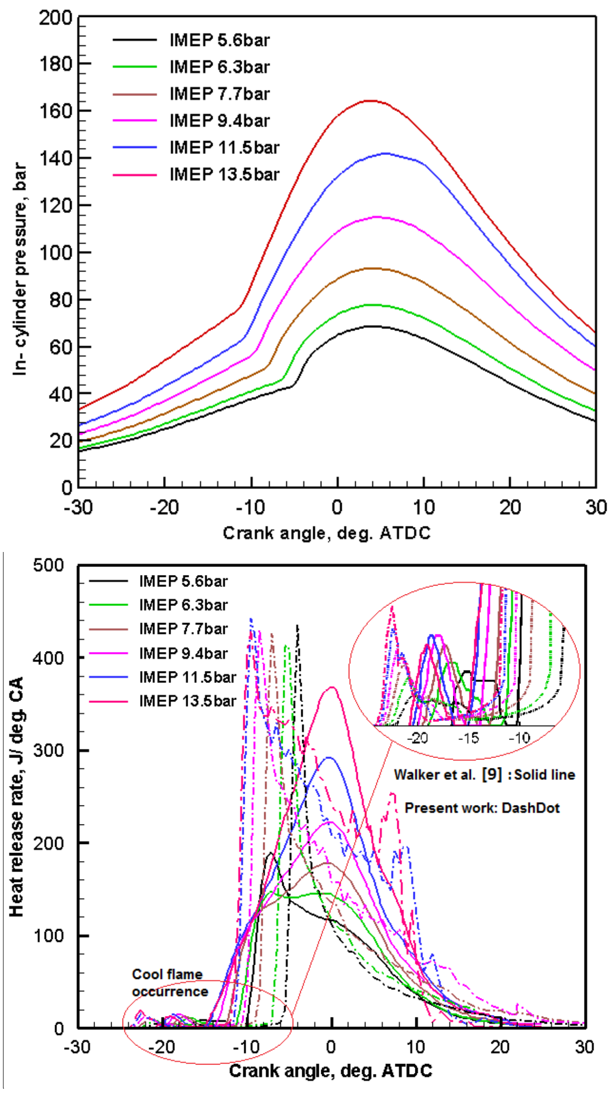

Figure 5. The in-cylinder pressure and the heat release rate for validation purpose.

Table 4. The CA50 evaluation for validation purpose.

	Low-Load range			Mid-Load range		
Gross IMEP (bar)	5.6	6.3	7.7	9.4	11.5	13.5
CA50 (º ATDC)	+2	+1.5	+1	+0.8	-0.8	-1.2

As mentioned, at the first stage of the heat release, the rate of heat release of n-heptane is very fast compared to the commercial diesel fuel which leads to a faster peak pressure rise rate. Thus, the PPRRs obtained from the RCCI combustion simulation are realistically higher than those of the experimental results [5]. However, the obtained PPRRs are below its permissible value, thus, the RCCI engine operates without any diesel knock resulted from the excessive heat release rate.

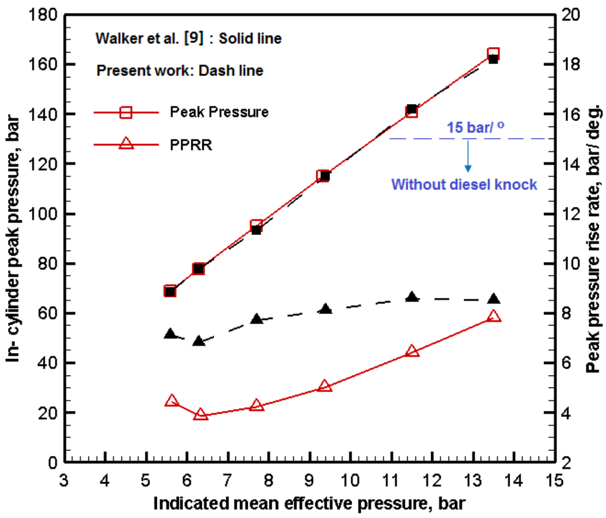

Figure 6. Validation of the engine in-cylinder peak pressure and the PPRR.

3) Engine Operating Conditions

In the current study, in RCCI engine fueled with NG enriched with hydrogen and diesel fuel, in order to provide an intake premixed charge, methane along with hydrogen addition is introduced to the combustion chamber through the intake port. Then, to provide an orderly ignition source for the available premixed charge with in combustion chamber, n-heptane is directly injected into the combustion chamber. Therefore, in order to improve the engine performance and lead to complete combustion in the RCCI engine fueled with NG/ diesel fuel at low-load, the simulation conditions are considered as in Table 5.

Based on Table 5, the total fuel energy content is kept fixed with no EGR, Hydrogen is gradually added to the NG whereas the LTC concept is satisfied (T_{peak}<1900 °K). Considering all cases of gross IMEPs, the amount of diesel fuel mass is fixed at 13 mg.

4) Results and Discussion

Based on derived simulation results in Table 5 at low- load, by fixing the total fuel energy content, it can be argued that substituting hydrogen instead of NG (i.e. methane) can reduce the amount of methane mass (as a fossil fuel) up to 45.23% for 5.6bar gross IMEP, 29.26% for 6.3bar gross IMEP, and 12.07% for 7.7bar gross IMEP Without using any exhaust gas recirculation. But, EGR should be used in order to satisfy the LTC concept (i.e. T_{peak}< 1900K) for beyond the mentioned reduction in methane mass. The goals of the present study are reducing fossil fuel consumption by substituting with hydrogen as green energy, enhancing the combustion performance, reducing the engine emissions, satisfying the LTC combustion concept, and maintaining the same engine power.

Therefore, for achieving these goals, the effects of hydrogen addition without the use of EGR which are important RCCI combustion characteristics at Low-Load are evaluated [27].

Table 5: RCCI combustion simulation conditions

Gross IMEP (bar)	Hydrogen addition (mg)	Methane mass (mg)	HES (%)	EGR (%)
5.6	0	42.38	0	0
	1.0	39.90	4.49	0
	2.0	37.56	8.89	0
	3.0	35.18	13.60	0
	4.0	32.72	17.76	0
	5.0	30.32	22.17	0
	8.0	23.21	35.70	0
	10	18.37	44.44	8.164
6.3	0	49.14	0	0
	1	46.79	3.95	0
	2	44.27	7.93	0
	3	41.92	11.84	0
	4	39.55	15.82	0
	5	37.16	19.75	0
	6	34.76	23.70	0
	8	29.95	31.58	8.18
7.7	0	61.04	0	0
	1	58.63	3.40	0
	2	56.33	6.55	0
	3	53.67	9.93	0
	4	51.39	13.21	15.44

4.1) In-cylinder Peak Pressure and Engine Load

As mentioned, the hydrogen flame propagation speed is higher than other hydrocarbon fuels. Hence, it causes to speed up the combustion and heat release rate compared with pure hydrocarbon fuel burning significantly.

Therefore, the in-cylinder peak pressure increases compared to RCCI combustion fueled with NG/ diesel fuel without hydrogen addition [28, 29]. Figure 7 shows that for all different IMEPs mentioned in Table 5, the in-cylinder pressure increases by adding hydrogen into the premixed intake charge.

Naturally, higher in-cylinder pressure occurs by enhancing the HES percentage for all mention engine loads, the EGR should be used in order to achieve a higher HES percentage (as highlighted in Table 5).

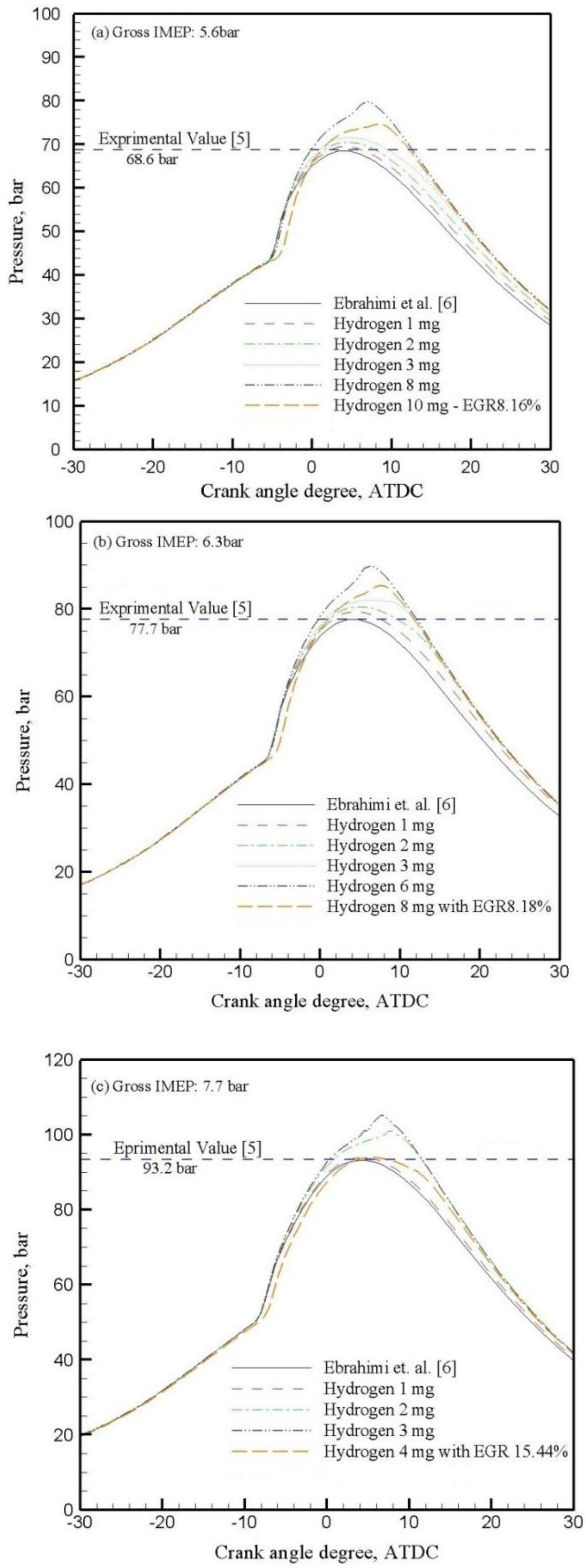

Figure 7: The effect of hydrogen addition on in-cylinder pressure at different IMEPs: (a) 5.6bar, (b) 6.3bar, (c) 7.7bar

However, the use of EGR along with hydrogen addition (i.e. the last case in each of IMEPs mentioned in Table 5) causes to reduce the combustion rate and the in-cylinder pressure. In this simulation study, gross IMEP as indicative of the engine load is calculated by dividing gross cycle work done during compression and expansion strokes at in-cylinder volume displaced for each cycle (V_d):

$$Gross\ \ IMEP = \frac{Gross\ Work}{V_d} = \frac{\int_{-180^\circ}^{+180^\circ} P\, dV}{V_d} \tag{1}$$

Based on all different cases in Table 5, gross IMEPs are assessed and presented in Fig. 8. As shown in this figure, the engine load increase by enhancing the HES percentage, sharply. But, as depicted in Fig. 8, the engine load is reduced when EGR is used to enhance the HES percentage, due to decreasing the peak pressure [30].

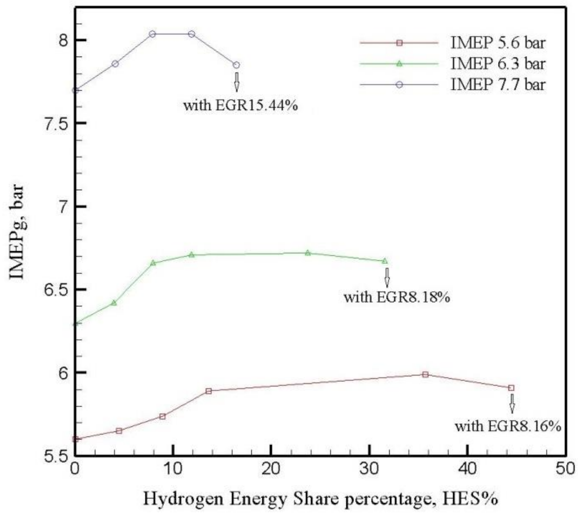

Figure 8: The effect of hydrogen addition on IMEPg

4.2) Diesel Knock Occurrence

One of the major factors which limit a heavy-duty diesel engine operation is the diesel knock phenomenon. This phenomenon is related to the excessive combustion noise resulted in the peak pressure rise rate. As mentioned earlier, hydrogen has higher flame speed compared to other hydrocarbon fuels, therefore, it causes that the fuel energy releases faster. Thus, it is expected that the in-cylinder pressure increases significantly. But, for all mentioned cases in Table 5, as presented in Table 6, despite increasing the in-cylinder pressure, the PPRRs are within acceptable limit (i.e. 15bar/º), hence, the engine operation is not exposed to diesel knock.

Another way to predict the diesel knock occurrence presented by Eng [31] is ringing intensity (RI) criterion. Based on Eng work results [24], the acceptable limit for RI is 5 MW/m2 to avoid the diesel knock occurrence in a heavy-duty diesel engine. Base on ENG work [31], RI can be calculated as

$$RI = \frac{1}{2\gamma} = \frac{\left(0.05(dP/dt)_{\max}\right)^2}{P_{\max}} \sqrt{\gamma R T_{\max}} \quad (2)$$

wherein $(dP/dt)_{\max}$, $P_{\max}$, $T_{\max}$, γ, and R are the maximum pressure rise rate, peak pressure, in-cylinder peak temperature, C_p / C_v ratio, and the gas constant, respectively. According to Eq. 2, for all cases mentioned in Table 5, RI is calculated and is presented in Table 6. As presented in this table, the calculated RIs are in its acceptable limit, thus, the engine operation at low-load is not exposed to any diesel knock.

Table 6: RI value for all different cases in Table 5

Gross IMEP 5.6bar	HSE (%)	0	4.49	8.89	13.60	35.70
	PPRR	7.9	7.82	7.81	8.07	8.33
	RI	4.01	3.95	3.94	4.24	4.25
Gross IMEP 6.3bar	HSE (%)	0	3.95	7.93	11.84	23.70
	PPRR	7.6	7.79	7.91	8.03	8.67
	RI	3.38	3.55	3.69	3.80	4.11
Gross IMEP 7.7bar	HSE (%)	0	4.06	7.85	11.85	-
	PPRR	8.06	8.13	8.21	8.36	-
	RI	3.22	3.28	3.36	3.29	-

4.3) Combustion Duration, Ignition Delay, and CA50

For all mentioned engine loads, the heat release rate versus crank angle degree is depicted in Fig. 9.

since, the flame speed of hydrogen is higher than other hydrocarbon fuels such as NG [12], the hydrogen addition causes that the combustion rate increases. Consequently, the fuel-burning rate is increased. By increasing the HES percentage, the rate of heat release is increased, drastically. In RCCI engine fueled with diesel fuel and natural gas, when natural gas is enriched with pure hydrogen, the faster heat release resulted from the faster flame velocity of hydrogen into the engine combustion chamber causes to raise the in-cylinder peak pressure rate. Thus, the probability of the excessive combustion noise occurrence will be increased.

Therefore, as shown in Fig. 10, the combustion duration reduces compared to the experimental results (i.e. 25 degree CA) [5]. One of the main challenges of using hydrogen as a sole fuel or an additive to gaseous hydrocarbon fuels in an internal combustion engine is the faster energy release due to the faster flame speed of hydrogen. This hydrogen characteristic causes to speed up the combustion process which leads to reduce the combustion duration. Generally, in a heavy-duty diesel engine, the trace of OH radical concentration leads to assess the combustion occurrence [32]. When hydrogen is added to the intake premixed charge, the OH formation as indicative of the combustion occurrence is delayed. As shown in the previous heat release curves, the start of combustion (SOC) is not affected significantly by hydrogen addition.

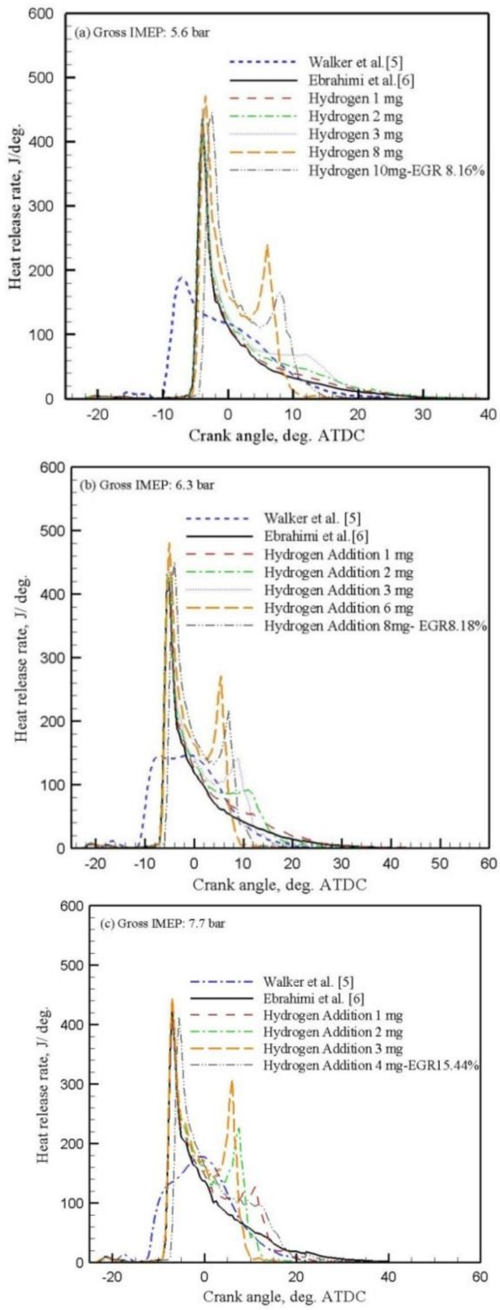

Figure 9: Heat release rate versus crank angle degree for different gross IMEPs: (a) 5.6bar, (b) 6.3bar, (c) 7.7bar

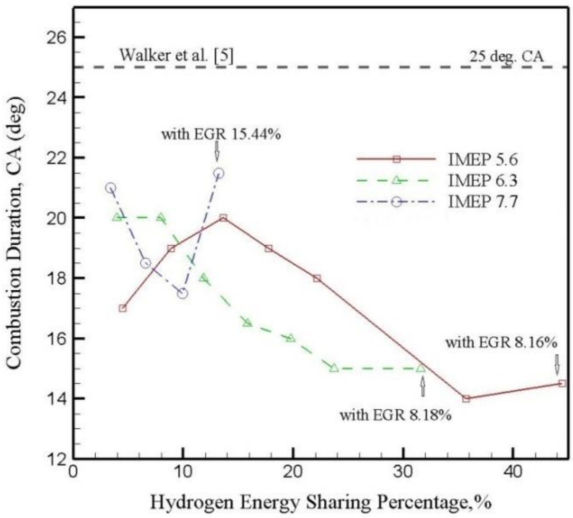

Figure 10: Combustion Duration versus the HES

This hydrogen addition effect can be seen in Fig. 11. Moreover, the delay time of the OH formation increases more by increasing the HES percentage. As for the all IMEP cases, by increasing hydrogen percentage, peak OH radical mass fraction increases. However the increase in OH radical mass fraction deceased while EGR was used. The amount of ignition delays is depicted in Fig. 12.

Due to the ignition delay time or the delay in combustion occurrence, the overall combustion duration shifts toward expansion stroke [33].

Hence, as illustrated in Fig. 13, CA50 is departed TDC when hydrogen is added to NG due to increasing the ignition delay time. But, CA50 gets close to TDC again by enhancing HES due to the reduction of the combustion duration.

Moreover, the use of EGR results in further ignition delay time and on the other hand reduces the combustion rate, thus, CA50 has pushed away from TDC compared with the cases without using EGR.

4.4) Gross Indicated Efficiency

The well- known equation which can be used to obtain the gross indicated efficiency (GIE) is presented below [34].

$$GIE=\frac{Work}{E_{in}}=\frac{\int_{-180^{\circ}}^{+180^{\circ}} P\,dV}{m_{fuel}\left[(x.LHV)_{hydrogen}+(x\cdot LHV)_{methane}+(x\cdot LHV)_{dieseloil}\right]} \qquad (3)$$

wherein E_{in}, m_{fuel}, $x_{hydrogen}$, $x_{methane}$, and $x_{diesel\ fuel}$ are the fuel energy introduced into the engine cylinder, the mass of the consumed fuel, mass fraction of hydrogen, methane, and n-heptane. Also, $LHV_{hydrogen}$, $LHV_{methane}$, and $LHV_{diesel\ fuel}$ are lower heating values of hydrogen, methane, and diesel, respectively.

The calculated GIE for all different cases mentioned in Table 5 are shown in Table 7. As shown, gross IMEPs are increased by enhancing HES, therefore, the delivered work to the engine's piston increases significantly. Since, it is assumed that the total fuel energy content to be fixed, despite the reduction in the hydrocarbon fuel consumption, the presence of hydrogen in the intake premixed charge causes to increase GIE over 50%.

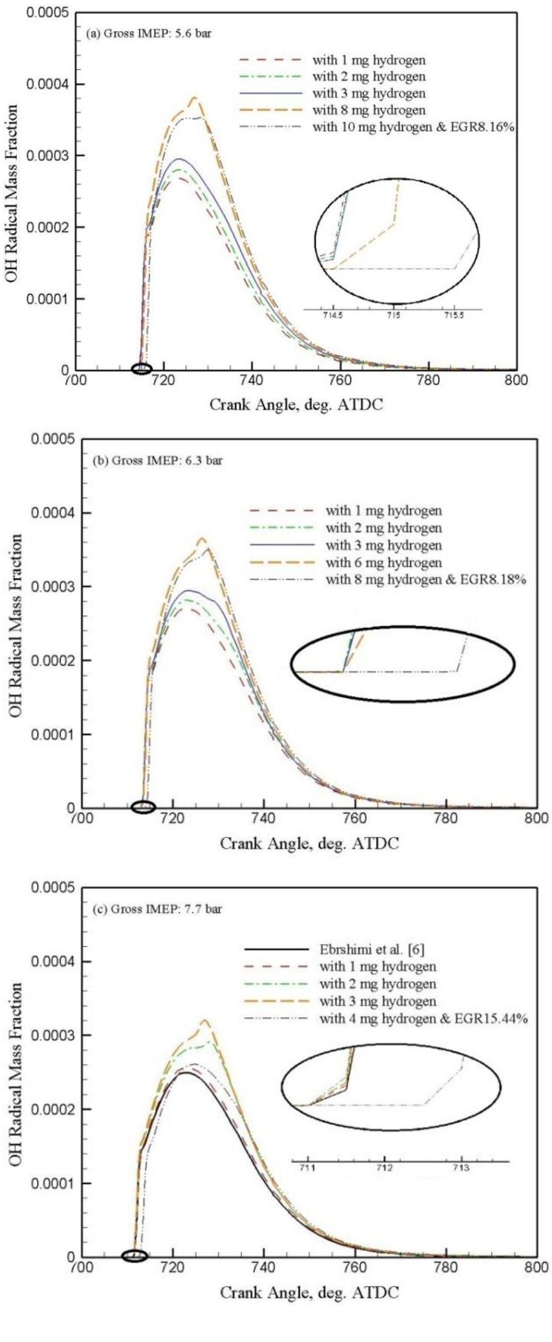

Figure 11: The delay of OH radical versus crank angle

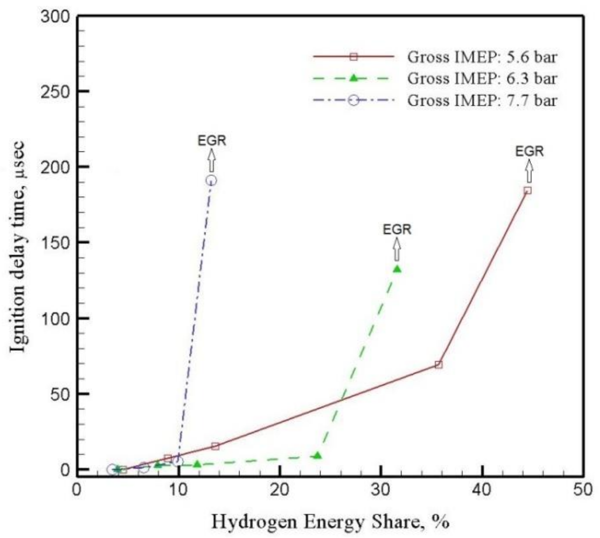

Figure 12: The ignition delay time versus HES

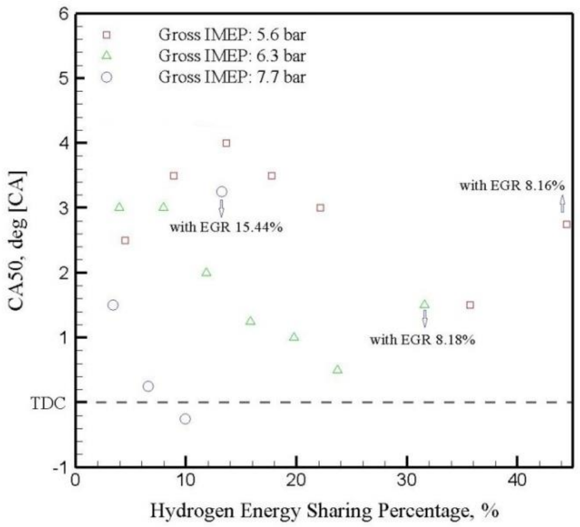

Figure 13: The CA50 versus HES

Table 7: Gross indicated efficiency for all gross IMEPs

IMEP 5.6bar	HSE (%)	0	4.49	8.89	13.60	35.70
	GIE	50.21	50.71	51.50	52.65	53.55
IMEP 6.3bar	HSE (%)	0	3.95	7.93	11.84	23.70
	GIE	50.20	51.75	53.12	53.50	53.53
IMEP 7.7bar	HSE (%)	0	4.06	7.85	11.85	
	GIE	51.30	62.51	64.34	64.09	

4.5) Engine Emissions

In the RCCI engine fueled with NG/ diesel fuel, when natural gas is partially replaced by hydrogen, due to increasing the combustion rate and faster heat release in the combustion chamber, higher in-cylinder temperature occurs in the engine cylinder and NOx emission increases. Figure 14 shows that by enhancing the HES percentage without using the appropriate EGR, NOx increases significantly compared to RCCI combustion fueled with NG/ diesel fuel without hydrogen addition. Also, when EGR is used for enhancing the HES percentage, due to decreasing the in- cylinder temperature, the NOx amount is reduced compared to the cases with hydrogen addition.
Based on Ebrahimi et al. simulation study [28] , in RCCI engine fueled with NG/ diesel fuel, substitution of hydrogen instead of NG causes that hydrogen interacts with carbon monoxide (CO) and oxygen (O2), therefore, it causes to generate two important species, namely, formaldehyde (CH2O) and hydroxyl (OH radical).

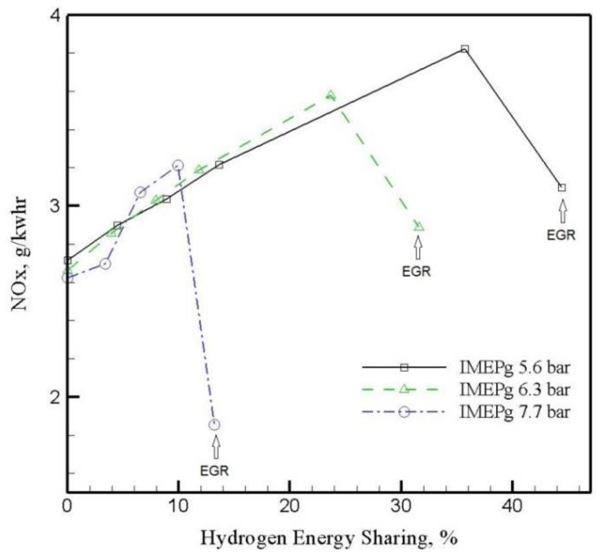

Figure 14: The effect of hydrogen addition on NOx emission

The generation of CH2O and its interaction with OH radical as an intermediate key, speed up the dissociation of methane (i.e. natural gas). Hence, when the HES percentage is increased (i.e. increasing the amount of hydrogen in the intake premixed charge), the rate of interaction between hydrogen and CO is increased and it speeds up the dissociation of methane in the combustion chamber. Therefore, as illustrated in Fig. 15, CO emission is decreased drastically. Furthermore, Hydrogen has much shorter quenching effects than other hydrocarbons fuels.
Thus, the hydrogen flame can get closer to the engine cylinder wall before it extinguishes. As a

result, in addition to the key role of hydrogen in the dissociation of methane, this hydrogen property also causes to burn much more effectively the presence hydrocarbon fuels in the RCCI engine's combustion chamber (i.e. NG and diesel fuel) especially near the cylinder walls [28, 35]. Thus, as depicted in Fig. 16, unburned hydrocarbon emission decreases significantly.

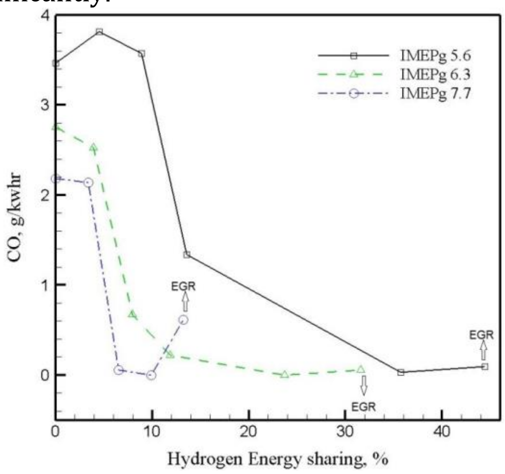

Figure 15: The effect of hydrogen addition on CO emission

Based on the United States environmental protection agency (EPA), another important species of gas engine emission is formaldehyde [36]. In heavy- duty diesel engines, near the engine cylinder wall, UHC emission would be increased due to the wall quenching effect. Based on CIMAC report [37], if methane (i.e. natural gas) is used in a diesel engine then the amount of unburned methane emission is grown drastically due to wall quenching effect. Indeed, the formation of formaldehyde emission has a direct consequence with the formation of unburned methane in the gas engines combustion chamber.

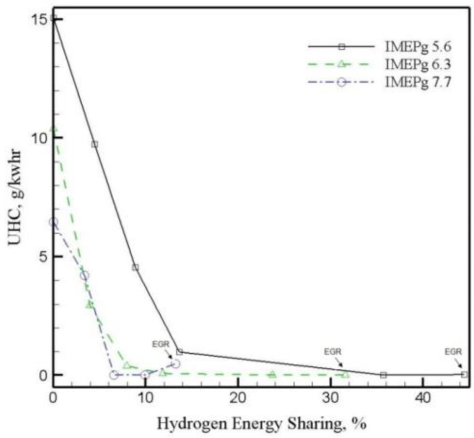

Figure 16: The effect of hydrogen addition on UHC emission

In this study, the evaluated engine operates under RCCI combustion strategy fueled with natural gas and diesel fuel. Therefore, it is expected that formaldehyde is formed in the combustion chamber. But, using hydrogen as an additive causes that the burning velocity of mixture (methane – air – hydrogen) increases as well as due to the presence of hydrogen, the wall quenching distance decreases significantly. Thus, the unburned methane decreases (Fig. 16). Moreover the formation of formaldehyde emission in the combustion chamber is decreased drastically as shown in Fig. 17. Figure 17 shows that the formaldehyde emission level of EPA (0.012 gr/kWh) is satisfied when hydrogen is added to natural gas.

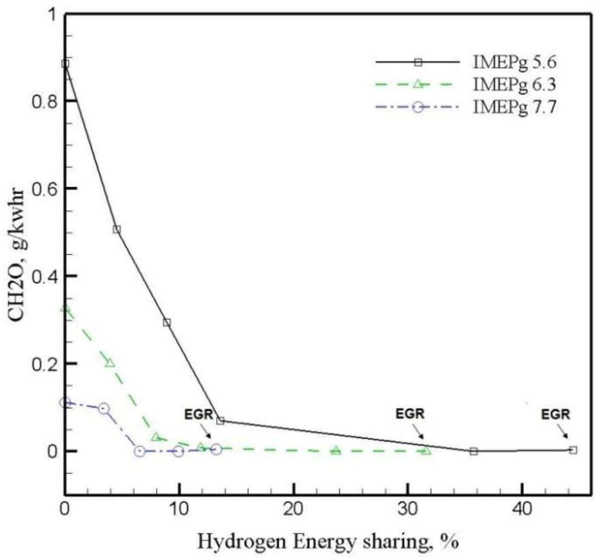

Figure 17: The effect of hydrogen addition on formaldehyde emission

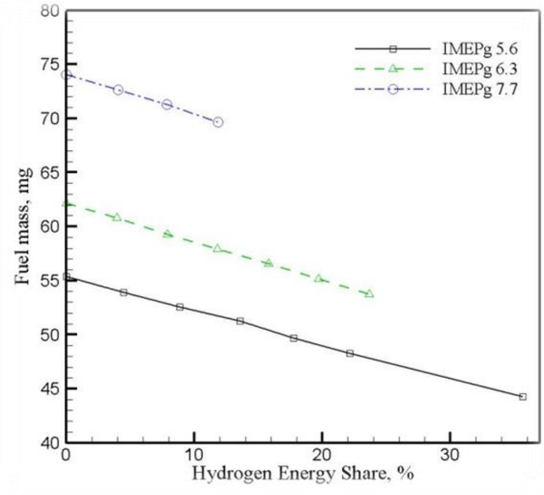

Figure 18: The hydrocarbon reduction on enhancing hydrogen energy share

4.6) Hydrocarbon fuels consumption

As mentioned earlier, in the present study, it is assumed that the total fuel content to be fixed When hydrogen is substituted instead of NG (i.e. methane). Since, the LHV of hydrogen is higher than that of methane; therefore, as shown in Fig. 18, by substituting hydrogen instead of methane, the total hydrocarbon fuels consumption is reduced significantly. It should be noted that, despite of reducing the total HC fuel consumption, the desirable goals in this study such as satisfying the low temperature combustion concept, enhancing the engine power, and decreasing the engine emissions are achievable.

5) Conclusions

In the present simulation study, for a single cylinder heavy– duty diesel engine operating under RCCI combustion fueled with diesel fuel/ natural gas enriched with hydrogen at low- load range (i.e. from 5.6bar to 7.7 bar), the following remarks can be concluded:

1) Hydrogen fuel can easily be added to natural gas without encountering any diesel knock occurrence even in the absence of EGR. This hydrogen energy share percentage could be as high as 35.7% for gross IMEP 5.6bar, 31.58% for gross IMEP 6.3bar, and 15.78% for gross IMEP 7.7bar.

2) By fixing the diesel fuel mass at 13mg, the overall hydrocarbon fuel consumption per each cycle can be reduced up to 45.23% for gross IMEP 5.6bar, 29.26% for gross IMEP 6.3bar and 12.07% for gross IMEP 7.7bar while at low-load, it is proven that there are no engine power losses.

3) In the RCCI engine operation at low- load, when the amount of hydrogen addition to natural gas increases, the ignition delay time and the combustion duration reduces. Also, CA50 gets closer to TDC.

4) When hydrogen is added to natural gas, the indicated work delivered to the piston over the compression and expansion strokes increases therefore, the GIE improves up to 53.55% for gross IMEP 5.6bar, 53.53% for gross IMEP 6.3bar, and 64.09% for gross IMEP7.7bar.

5) When hydrogen is added to natural gas that is enhancing the hydrogen energy share percentage in the absence of EGR; NOx increases significantly, but, CO and UHC emissions decrease drastically compared to RCCI combustion fueled with NG/ diesel fuel without hydrogen addition.

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تاثیر افزودن هیدروژن بر عملکرد موتور RCCI با سوخت گاز طبیعی و گازوئیل در بار پائین

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چکیده

هدف از این مطالعه بررسی اثر افزودن هیدروژن بر عملکرد و انتشار آلاینده یک موتور دیزل سنگین تحت احتراق RCCI با سوخت گاز طبیعی و گازوئیل است. بر این اساس، یک موتور دیزل تک سیلندر دیزل سنگین برای کار در محدوده بار پائین از فشار موثر متوسط اندیکاتوری ناخالص از ۵,۶ تا ۷,۷ بار، دور ثابت موتور، و مقدار ثابت گازوئیل مورد استفاده قرار می گیرد. علاوه بر این، برای کاهش مصرف سوختهای هیدروکربنی، به تدریج هیدروژن به گاز طبیعی اضافه می شود در حالیکه محتوای کل انرژی سوخت ثابت نگه داشته می شود. نتایج نشان می دهد که با افزودن هیدروژن به گاز طبیعی در موتور RCCI در فشار موثر متوسط اندیکاتوری ناخالص ۵,۶، ۶,۳ و ۷,۷ بار، سهم انرژی هیدروژن به ترتیب تا ۳۵,۷٪، ۲۳,۷۰٪ و ۹,۹۳٪ ارتقا می یابد. همچنین، مصرف کلی سوخت هیدروکربنی در هر چرخه را می توان به ترتیب تا ۴۵/۲۳ درصد، ۲۹/۲۶ درصد و ۱۲/۰۷ درصد کاهش داد. اگرچه انتشار آلاینده های CO و UHC به شدت کاهش می یابد اما آلاینده NOx به طور قابل توجهی با افزودن هیدروژن در مقایسه با احتراق RCCI با سوختهای گاز طبیعی و گازوئیل افزایش می یابد.

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