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Influence of hydrogen co-combustion with diesel fuel on performance, smoke and combustion phases in the compression ignition engine

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ABSTRACT

The main objective of this study was to examine impact of hydrogen addition to the compression ignition engine fueled with either rapeseed methyl ester (RME) or 7% RME blended diesel fuel (RME7) on combustion phases and ignition delay as well as smoke and exhaust toxic emissions. Literature review shows in general, hydrogen in those cases is used in small amounts below lower flammability limits. Novelty of this work is in applying hydrogen at amounts up to 44% by energy as secondary fuel to the compression ignition engine. Results from experiments show that increase of hydrogen into the engine makes ignition delay shortened that also affects main combustion phase. In all tests the trends of exhaust HC and CO toxic emissions vs. hydrogen addition were negative. The trend of smokiness decreased steadily with increase of hydrogen. Amounts of hydrogen addition by energy share were limited to nearly 35% due to combustion knock occurring at nominal load.

Introduction

The fuel consumption of whole EU transport fleet was 195,315 million tons of diesel fuel (DF) and 88,325 million tons of petrol in 2010. The forecast for 2018 were 214,344.5 million tons of DF and 72,896 million tons of petrol [1]. New diesel cars registrations increased from 23% to 51% in EU during the period of 1995–2010, while the diesel cars in EU shared 35.5% in 2010 [2]. However, diesel engines contribute the environmental pollution in significantly higher level then gasoline engines. Therefore, considerable efforts have been focused toward reducing the diesel exhaust toxic emission as it has negative effect on both the natural environment and human health. In 2015, global CO_2 emissions reached 32.3 $GtCO_2$, while all kinds of the transport accounted for 7.75 $GtCO_2$. With increasing emissions from road transport by 68% since 1990, it accounts almost for the quarter of the global CO_2 emissions – 5.8 $GtCO_2$ [3]. The United Nations and the EU adopted a number of legal regulations, which set the legal basis to take care of the sharpening trend in climate change. The EC took the commitment up to year 2050 to reduce the emissions by

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80–95% in comparison to 1990. Whereas the transport has to reduce its emissions by 60% by 2050 in regards with 1990 [4]. To achieve and satisfy these emission obligations, progress research and development in engine technologies is required. The use of renewable fuels has the potential to reduce the emissions and, thus mitigate the effects of the environmental crisis and climate change. Among the currently available renewable fuels there are alternative biomass based biofuels. The physical properties and availability makes hydrogen quite attractive alternative fuel for road transport. Verhelst et al. [5] described the long-term scenario of hydrogen usage as an energy source including it qualitative and quantitative descriptions in order to implement the transition towards clean and sustainable energy. The authors demonstrated the importance of variety of hydrogen-based energy technologies, which enable the efficient and economical way to ensure energy needs. Although the use of sole hydrogen for combustion engines are hardly possible, the co-combustion with various fuels including renewable fuels makes objectives of research interesting. The current alternative, first generation, biomass based biofuels are produced from commonly available, edible feedstock's using well-established conversion technologies [6–9]. Biofuels produced with aid of second-generation biomass conversion technologies do not compete with food production. High raw material costs is the issue, which is decisive in making biofuel processes economically attractive. The other issue associated with production of biofuels is the energy return to energy invested. This ratio should be at least 3:1 to cover expenses for infrastructure and transportation, while now for biofuels it is approximately 1.3:1. The main sources of biofuels are fatty acids of vegetable oils and animal fats. Vegetable oils consist of a mixture of triglycerides, i.e. esters of glycerol and unsaturated fatty acids. Transesterification of triglycerides with methanol gives a mixture of fatty acid methyl ester (FAME) and glycerol, which can be also considered as engine fuel [10]. The final products of this reaction are glycerol ($C_3H_5(OH)_3$) and RME ($C_{17}H_{31}COOCH_3$) [11]. The standard EN 590:2009 in accordance to the Directive 2009/30/EC defines properties of B7 diesel fuel sold at retail and limits the content of the FAME (RME) to max. 7 vol% in the fossil diesel fuel. This fuel was denoted here as RME7. These two fuels (RME7 and pure RME) with hydrogen were used in engine combustion tests presented in this paper.

As known from literature survey, there are several works presenting impact of hydrogen on diesel fuel combustion in the compression ignition (CI) engine. Szwaja et al. [14] carried out tests with amounts of 5% hydrogen by energy share to fossil diesel fuel and revealed the shorten ignition lag and decrease in the rate of combustion pressure rise. With a hydrogen energy share (HES) of 5–15%, the entire combustion duration did not change significantly, but with hydrogen share of 15%, peak combustion pressure p_{max} increased. With HES of 17% the combustion knock occurred and with HES of 25% the fast combustion was accompanied by severe combustion knock. Labeckas et al. [12] conducted experimental study on a CI engine fueled with 5–15 vol% ethanol blended DF and additionally blend (E15B) consisted of ethanol (15 vol %), DF (80 vol%) and RME (5 vol%). They observed that oxygen content in the mixture reflects the auto-ignition delay caused by use of E15B blend more predictably than the Cetane Number (CN) does. The auto-ignition delay for oxygenated blend E15B was 15.4% longer than for DF and the indicated specific fuel consumption (ISFC) was increased. Yang et al. [15] studied influence of hydrogen addition on the performance of the CI engine and determined the best ratio for H_2 addition. They found hydrogen enrichment decreases particulate matter (PM) emissions and provides optimal results for maximum heat release rate (HRR) at 17% H_2 by volume. In the investigation by Rocha et al. [16] performed on a diesel generator, hydrogen was supplied at HES of 5–24% to the diesel – biodiesel (7%) blend (B7). With increase of HES, both CO_2, CO and HC emissions decreased. However, the NO_x increased due to increase of in-cylinder temperature. There was also both increase in the peak pressure and heat release rate noticed, since ignition delay was reduced due to increase of HES. Experiments carried out on the CI engine [17] with fossil DF blended with 20 vol% RME and HES of 0–5% revealed the lower engine performance, efficiency and toxic emissions (CO, HC) except NO_x, which slightly increased. As they observed, addition of hydrogen did not affect auto-ignition delay. No significant increase in NO_x was also observed during testing the CI engine with EGR [18] running with DF blended 7% FAME. However, HES of 25% caused the reduction of p_{max} and reduction of CO_2 emission by 22%. Chelladorai et al. [19] investigated the grapeseed oil as a fuel substitute obtained from biomass waste from winery industry. He studied effect of hydrogen addition to a CI engine fueled the grapeseed oil. At full load, the maximum indicated thermal efficiency (ITE) with diesel, grapeseed biodiesel (GSBD) and neat grapeseed oil (NGSO) increased from 32.34%, 30.28% and 25.94% to 36.04%, 33.97% and 30.95% for HES of 14.46%, 14.1% and 12.8%, respectively. Ignition delay increased with hydrogen induction as a result of reduced oxygen concentration in the in-cylinder charge. Based on studies by Zhou [23] on the CI engine with various amounts of hydrogen added, it shows that emissions and performance parameters are dependent on the parameters as follows: injection timing of DF, its combustion duration, HES, BMEP and engine speed.

As known, the lower flammability limit (LFL) of the hydrogen-air mixture changes with change of temperature and pressure. The experiments performed by Schroeder and Holtappels [24] show that LFL decreases with increase in temperature. The temperature in region of the actual start of combustion (SOC) was 370–412 °C with both fuels during the experiment performed by authors. At this temperature, the LFL decreased to 1.5% hydrogen by volume. However, this decrease of LFL does not cause hydrogen ignition due to temperature is still too low (370–412 °C) and not sufficient for hydrogen auto-ignition. Moreover, increase of pressure has opposite effect to temperature. The LFL increased from 4% to 5.6% with increase of pressure up to 50 bar and further increase of pressure has no influence on LFL as reported from Refs. [24,25]. At ignition point the pressure in the cylinder was 3.5–3.9 MPa, while the LFL was 3.0–3.1% hydrogen by volume. Therefore, one concludes that hydrogen can be effectively co-combusted with injected biofuels if only LFL was achieved in the engine cylinder. Before that, the lean mixture of air – hydrogen with biofuel burns incompletely and hydrogen does not make positive effect on the combustion duration and engine performance [5,26].

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Table 1 – Fuel properties [5,7,12,13].

Properties	RME7	RME	Hydrogen
Chemical formula	$C_{14}H_{24.2}O_{0.8}$	$C_{17}H_{31}COOCH_3$	H_2
Composition, %wt	81.9 C, 11.9 H, 6.2 O	77.5–77.9 C, 11.3–11.7 H, 10.8 O	100
RME addition, %vol	7	100	–
Density, kg/m^3 at 15 °C and 1.01 bar	838.7	883.7	0.08985
Molar mass, g/mol	205.3	294.5–318.5	2.016
Lower heating value, MJ/kg	42.14	37.4	120
Lower heating value, MJ/Nm^3	36,095	32,700	10.7
Stoichiometric air-fuel ratio, kg/kg	14.35	12.4	34.2
Stoichiometric air-fuel ratio, Nm^3/Nm^3	not applicable		2.6
Heating value of stoichiometric mixture, MJ/kg	2.74	2.79	3.40
Heating value of stoichiometric mixture, MJ/Nm^3	3.60	3.58	3.00
Auto ignition temperature at STP,°C	~260	342	585
Flammability limits at NTP, %vol	0.6–7.5	0.8–10	4–75
Cetane number	51.7	51.7	5–10
Carbon to hydrogen ratio (C/H)	6.9	6.5	0

As reviewed, there are some gaps in knowledge dealing with hydrogen assisted combustion in the CI engine fueled various fuels. This paper presents results of investigation on hydrogen co-combusted with RME7 and pure RME on both performance, combustion phases and toxic exhaust emissions from the CI engine. It was observed, that emissions and engine performance are dependent on the following: HES, injection parameters as timing and its duration, and equivalence ratio. The novelty of this work deals with relatively high hydrogen content in the entire amounts of fuels combusted at in the CI engine at the hydrogen knock onset. Contribution to the state-of-art is realized by knowledge extension in the field of thermodynamic analysis of the diesel engine working in the dual-fuel mode and use hydrogen as the secondary fuel at amounts up to 44% by energy.

Experimental set-up and procedure

For tests in this study two fuels were applied as follows:

- RME7 – mixture of diesel fuel and 7% RME in accordance to the standard EN 590:2013,
- pure RME.

The properties of the pure RME along with RME7 and hydrogen are presented at Table 1.

Tests were performed on an engine modified to work in dual fuel mode: gaseous fuel and liquid fuel (Fig. 1). The single cylinder stationary compression ignition engine Andoria S320 was used for this purpose. It was equipped with the high pressure

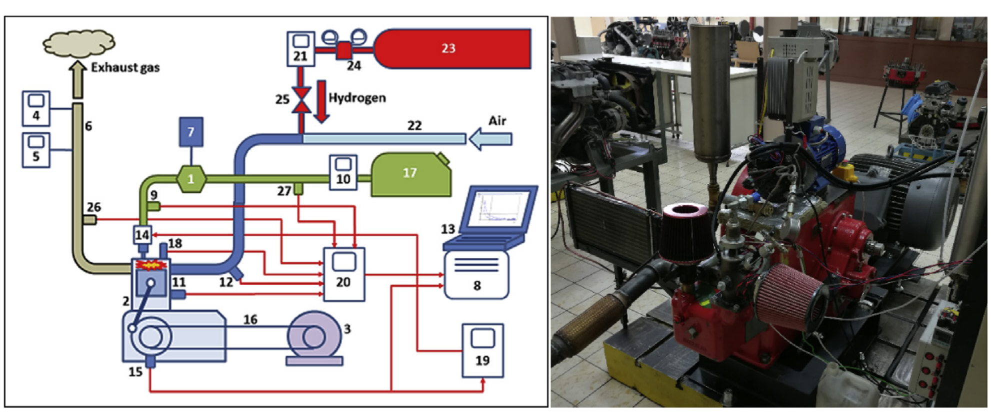

Fig. 1 – Test bed. 1 – DF pump, 2 – CI engine, 3 – Generator, 4 – Smoke analyzer, 5 – Exhaust gas emission analyzer, 6 – Exhaust pipe, 7 – electric motor, 8 – Data acquisition system, 9 – DF pressure sensor, 10 – DF flow meter, 11 – Engine temp. sensor, 12 – Inlet air temp. sensor, 13 – PC with SAWIR software, 14 – DF common rail injector, 15 – Crank angle encoder, 16 – Drive belt, 17 – DF tank, 18 – In-cylinder pressure sensor, 19 – DF injection controller, 20 – Amplifiers & A/D converters, 21 – Hydrogen flow meter, 22 – Air intake pipe, 23 – Hydrogen high-pressure tank, 24 – Hydrogen one-stage pressure controller, 25 – Hydrogen firebreak arrestor, 26 – Exhaust gas temperature sensor, 27 - DF temperature sensor.

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common rail fuel pump Bosch CR/CP1S3 driven by the 2.2 kW electric motor GL-90L2-4. Another electric motor was used as the starter for this engine. After starting up the engine, it transmits energy to a power generator. The engine was set to operate at the constant speed of 965 rpm ±0.83%. The generated electric power was supplied to the power grid and was measured. The technical specifications of the engine are given at Table 2.

Each experiment was conducted at various loads expressed by indicated mean effective pressure (IMEP). The IMEP was controlled by changing both the liquid and gaseous fuel (H_2) doses to the engine. The hydrogen was supplied together with air into the engine intake manifold. Inside the engine cylinder, air–hydrogen mixture was mixed with vapors and droplets of the injected liquid fuel. Next, this entire mixture was self-ignited at elevated temperature and pressure due to piston action at the compression stroke. Hydrogen was supplied into the engine intake manifold out of a tank with a single-stage pressure regulator to reduce its pressure to 1 bar. A firebreak valve was installed upstream to the air intake manifold to prevent flashback phenomenon. Toxic emissions in the exhaust gases were analyzed using Bosch and Maha (smoke) analyzers. In-cylinder pressure was measured by the piezo-ceramic sensor Kistler 6061B installed at location of the pre-heating plug. The crank angle (CA) was measured by the encoder Kistler type 2612C. The data acquisition A/D converter PCI-DAS 6036 was used in line with PC software SAWIR for online real time combustion pressure analysis.

Two liquid fuels were tested for this experimental study: RME7 and RME. Tests of the RME7 were performed as follows:

- Low Load (LL) with IMEP = 259.4–288.3 kPa (lean mixtures with $\lambda = 3.74$–4.05),
- Medium Load (ML) with IMEP = 422.3–468 kPa ($\lambda = 2.28$–2.48),
- Nominal Load (NL) with IMEP = 576.4–622.5 kPa ($\lambda = 1.57$–1.72).

Whereas tests of the RME were performed at following conditions with relative equivalence ratio lambda similar to previous tests as depicted in Fig. 2:

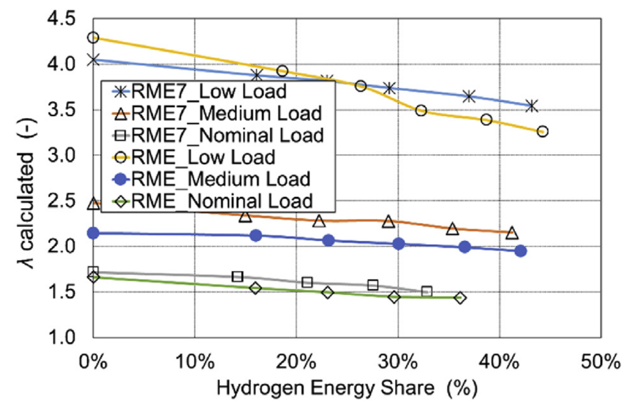

Fig. 2 – Relative equivalence ratio λ vs. *HES* at various engine loads.

- LL with IMEP = 262.5–295.6 kPa,
- ML with IMEP = 379.7–508.5 kPa,
- NL with IMEP = 519.2–625.3 kPa.

As the presence of hydrogen affects the combustion duration, start of diesel injection timing φ had to be fixed that enabled comparison and analysis of engine parameters at various HES.

Optimal injection timings were found at preliminary tests. The optimal injection timing was determined as the timing which causes position of 50% MFB (mass fuel fraction burnt) located in between 8 and 12 CA deg aTDC. The optimal injection timings φ for the fuels applied in tests are showed in Table 3.

As seen from the test matrix (Table 3) hydrogen amounts at nominal loads were reduced to 33 and 36% by energy for RME7 and RME, respectively, due to combustion knock occurrence while hydrogen exceeded these limits.

Analysis and discussion

The main objective of the presented research was to examine impact of HES as follows:

Table 2 – Technical specifications of the test engine Andoria S320.

Parameter	Value
Number of cylinders	1
Bore diameter, mm	120
Piston stroke, mm	160
Displacement, cm^3	1810
Compression ratio	17
Rated power, kW/HP	13.2/18
Rated speed, rpm	1500
Peak torque, Nm	84.4
Peak torque speed, rpm	1200
Length of connecting road, mm	275
Intake valve opening	23° bTDC
Intake valve closing	40° aBDC
Exhaust valve opening	46° bBDC
Exhaust valve closing	17° aTDC

Table 3 – Test matrix - injection timings, loads and hydrogen additions.

Test No.	Composition of combustible mixture	Injection timing φ, deg before the top dead center (bTDC)	Loads
1.	RME7 + H_2 0%	$\varphi_1 = 18°$	Low Load (LL)
2.	RME7 + H_2 (16–43%)	$\varphi_2 = 16°$	
3.	RME7 + H_2 0%	24°	Medium
4.	RME7 + H_2 (15–41%)	20°	Load (ML)
5.	RME7 + H_2 0%	30°	Nominal
6.	RME7 + H_2 (14–33%)	26°	Load (NL)
7.	RME + H_2 0%	16°	Low Load (LL)
8.	RME + H_2 (19–44%)	14°	
9.	RME + H_2 0%	22°	Medium
10.	RME + H_2 (16–42%)	20°	Load (ML)
11.	RME + H_2 0%	28°	Nominal
12.	RME + H_2 (16–36%)	26°	Load (NL)

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- engine performance including combustion properties and combustion duration,
- exhaust toxic emissions

In the CI engine working at the constant speed of 965 rpm and under various load ranges as follows: low (260–298 kPa), medium (380–508 kPa) and nominal (520–661 kPa) loads.

Combustion properties

The positive trends in maximum combustion pressure were observed with increase of HES within all ranges of loads and both tested fuels (Fig. 3). In fact, at the LL the maximum pressure fluctuates within the ranges of 4.88–5.08 MPa and 4.79–5.04 MPa with RME7 and RME, respectively. The negligible influence of hydrogen fraction at the LL and partially at the ML–NL with low HES was probably caused by low hydrogen fraction in the engine combustion chamber, which was below LFL for hydrogen (Table 1). As hydrogen affects the combustion duration, hence, start of diesel injection timing φ was set at fixed position during tests of hydrogen–diesel mixture, that makes it possible to compare and analyze combustion phases with various HES. Experiments revealed that the hydrogen–diesel mixture combustion leads to higher in-cylinder peak pressure with HES over 20% as depicted in Fig. 3b.

ISFC decreases with increase of HES. RME at medium and nominal loads has the highest decrease of ISFC by 23.3% in comparison to 19.8% for RME7 (Fig. 4). Hydrogen due to high flame speed and short quenching distance extends the flammability limits of RME–hydrogen mixture, provides RME completely combusted especially under higher loads, what leads to reduce ISFC as it was stated by Baltacioglu [20]. Main reason that ISFC is remarkably reduced, comes from relatively high calorific value of hydrogen. Hence, higher hydrogen addition, lower ISFC. Additionally, as observed, the engine load is limited by abnormal combustion (knocking), which might appear at nominal loads and HES higher 35%. At those conditions knock can be easily transformed to heavy knock and form extremely high in-cylinder pressure pulsations over 1 MPa [27] leading to increase heat transfer rate to the piston crown and can quickly damage the piston.

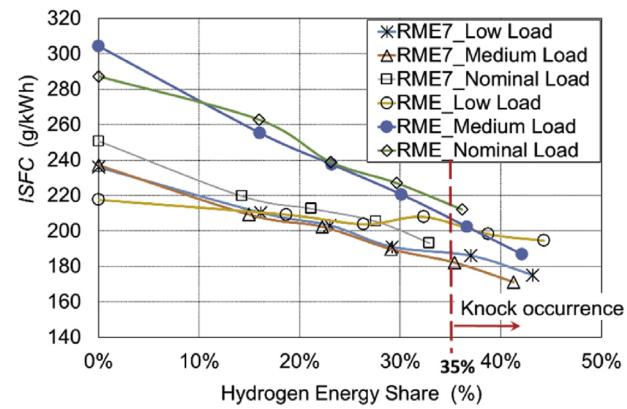

Fig. 4 – *ISFC at various engine loads against HES.*

Indicated thermal efficiency (ITE) is inversely proportional to ISFC taking into account lower heating value (LHV) for entire combustible mixture consisted of RME, DF and hydrogen. Although, ISFC decreases as presented in Fig. 4, but ITE is approximately at the same level in between 0.33 and 0.36 except test with RME at low load as shown in Fig. 5.

The ignition lag can be expressed by the initial combustion phase CA0-10 starting from the ignition point until 10% fuel burnt. Hence, with increase in HES the ignition delay (lag) gets shorten as depicted in Fig. 6. The CA0–10 shortens with increase in HES due to high premixed combustion rate and impact of higher laminar speed of hydrogen flame at all engine loads and for both RME and RME7. Increase of hydrogen also reduces the main combustion duration CA10–90 (Fig. 7) which is accelerated by the first combustion phase CA0–10.

Exhaust toxic emissions

Investigation on RME and RME7 with hydrogen addition concerns also measurements of exhaust toxic emissions focusing on NOx, HC and CO. Additionally, CO_2 was also measured with respect to confirm hydrogen impact on CO_2 reduction.

As seen in Fig. 8a, HC emission strictly goes up with higher engine load. Hydrogen addition only slightly reduces HC emission, particularly in low loads. At engine higher loads, HC increase is observed with lower HES as result of accelerating

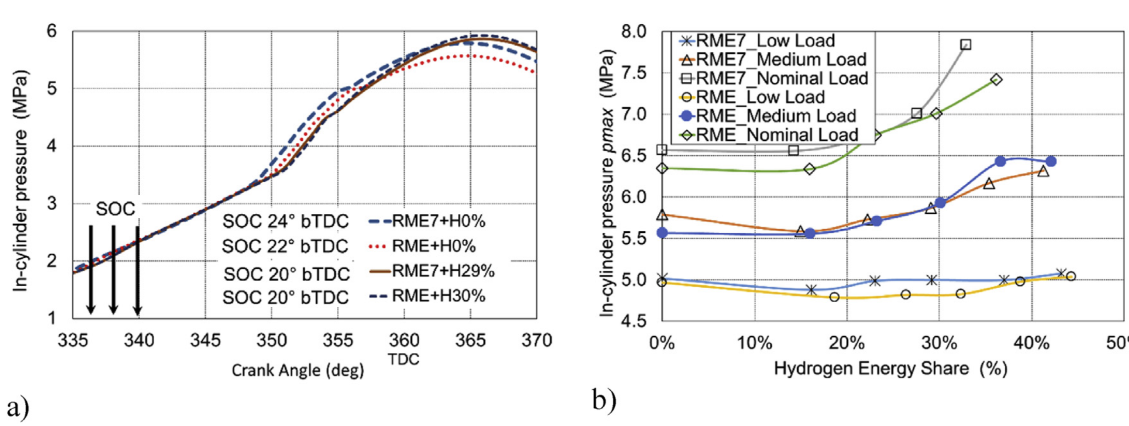

Fig. 3 – a) In-cylinder pressure at Medium Load b) In-cylinder maximum pressure p_{max} vs. *HES*.

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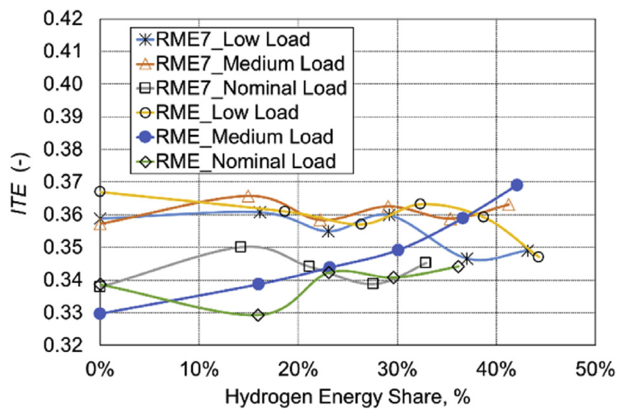

Fig. 5 – *ITE* vs. *HES* at various engine loads.

the combustion process. Next, HC drops with further HES increase due to higher overall combustion temperature. The hydrogen induction up to 15% decreases the NO. However, NO increases with HES higher 15% (Fig. 8b). This trend can be explained with the same phenomenon as it was discussed for HC trend line. Additionally, these trends in both HC and NO are confirmed by well-known NO-HC trade-off, which presents these both emissions inversely proportional to each other. Reduction at the low HES up to 5% was observed and confirmed by Senthil Kumar [21] and Bika et al. [22]. They also explained NO reduction in this HES range as result of slower combustion that forced ignition timing to be more advanced what decreased the combustion rate just after start of combustion [14]. At higher hydrogen rates NO emission increased. The highest increase rate of NO was at the nominal load at max. HES.

Similar trend to HC is observed for CO emission (Fig. 9a). CO presence in the exhaust gases resulted from incomplete combustion due to both short residence time for fuel molecules in the engine combustion chamber as well as relatively low combustion temperature, which decreases overall combustion reaction rate.

Unlike CO, CO_2 presence in exhaust gases results from complete combustion. Its emission is associated with carbon balance in the combustion reaction, therefore, higher carbon content in fuel implies higher CO_2 emission as far as the CO_2 is the product of complete combustion. The carbon content is usually expressed by the C to H ratio of the specific fuel. Hence, higher hydrogen content in the fuel (denoted with HES) makes the C/H ratio lower, what contributes to lower CO_2 emission as shown in Fig. 9b.

Smokiness is a parameter, which characterizes exhaust gases from the CI engine. Smokiness depicts transparency of exhaust gases contaminated with condensed unburnt fuel and soot, which are considered as major substances causing

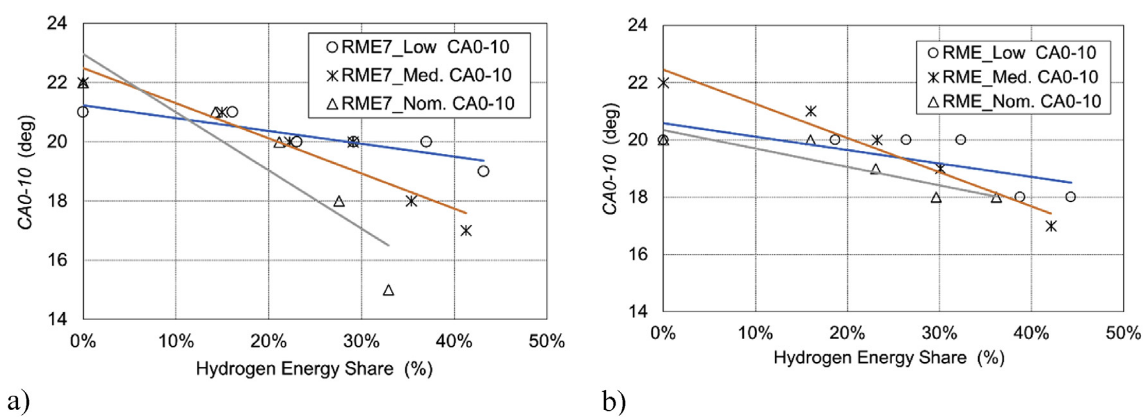

Fig. 6 – The combustion phase CA0-10 vs. HES for a) RME7 and b) RME.

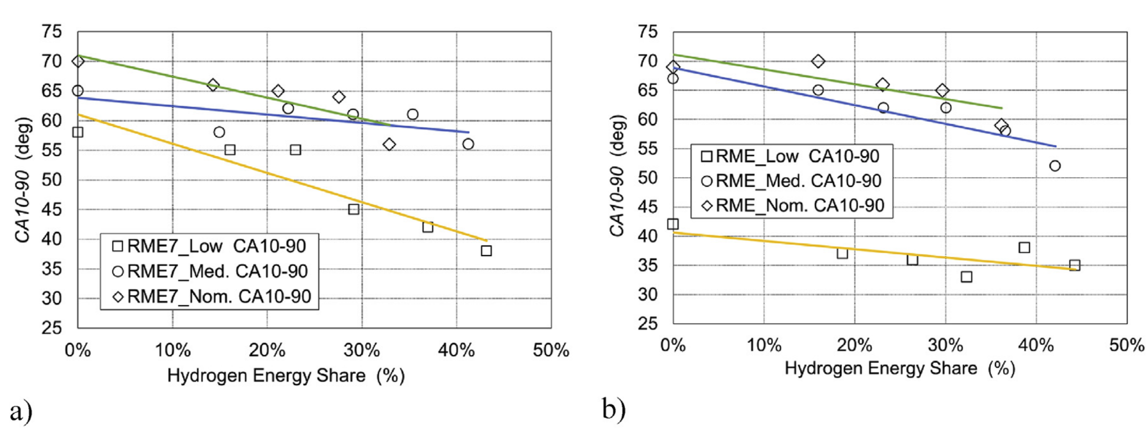

Fig. 7 – The main combustion phase CA10-90 vs. HES for a) RME7 and b) RME.

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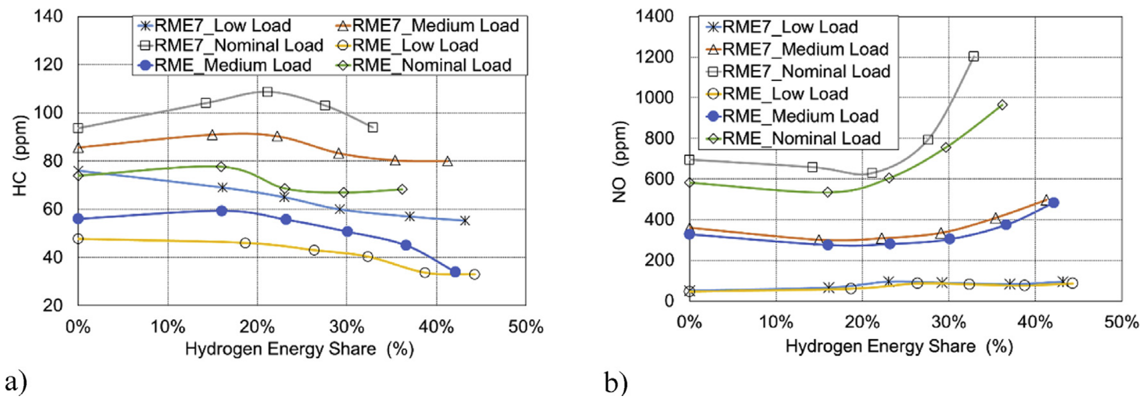

Fig. 8 – a) HC and b) NO against *HES* at various engine loads.

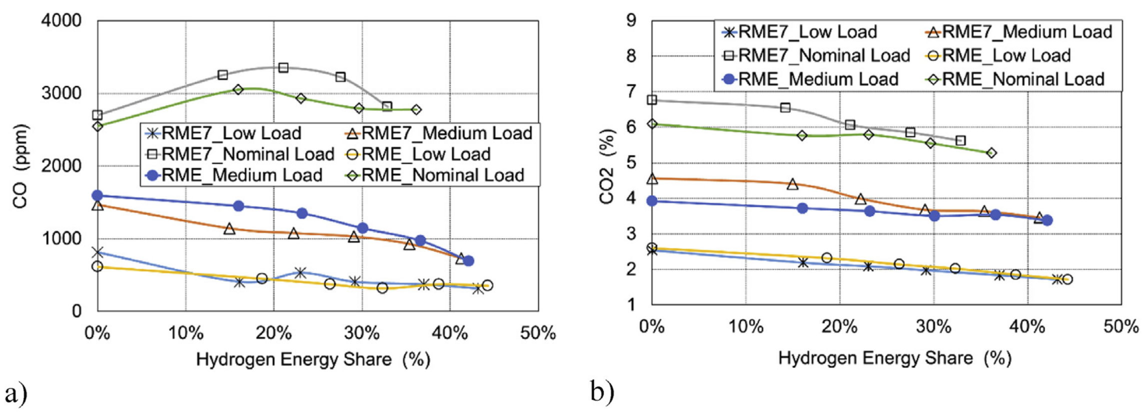

Fig. 9 – a) CO b) CO2 vs. *HES* at various engine loads.

smoke. Unburnt hydrocarbon based molecules and soot are usually inline with each other and they are mostly formed as result of both local oxygen deficiency and short time for complete combustion as it is observed for HC and CO emissions. As seen in Fig. 10, smokiness is in negative trend with HES. It means, that hydrogen assisted diesel fuels provides unfavorable conditions for soot formation. Among all the exhaust emissions tested, smokiness is the parameter which significantly decreases with increase in HES.

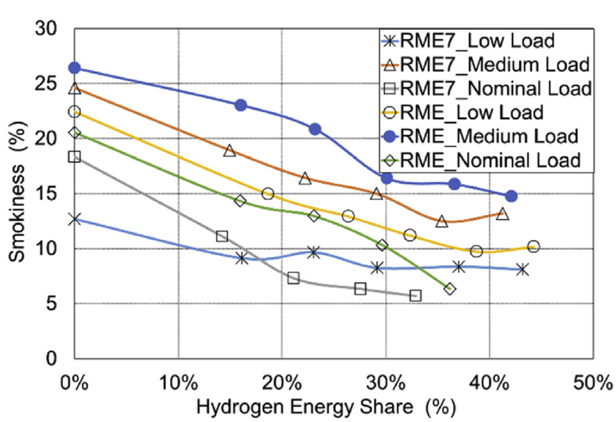

Fig. 10 – The smokiness vs. *HES* at various engine loads.

Conclusion

Investigation presented here deals with impact of hydrogen addition to diesel based fuels RME and RME7. Hydrogen addition to these fuels increases the LHV of the entire combustible fuel charge trapped in the engine cylinder. Higher fuel's LHV usually provides better conditions to obtain higher combustion temperature, hence, it affects other combustion parameters and exhaust emissions. The conclusions from the investigation are as follows:

- In-cylinder peak combustion pressure increases significantly by approximately 15% at medium and nominal loads with hydrogen increase from 20 to 33%.
- The combustion duration CA0–10 (considered as ignition lag) and CA10–90 (main combustion phase) shorten with increase of HES.
- Presence of hydrogen also contributes to decrease in ISFC due to higher LVH for the total in-cylinder charge due to high LHV of hydrogen. Increase in ISFC does not affect brake thermal engine efficiency, which is at stable level of 0.35 ± 0.015.
- HES of less than 15% decreases the NO emissions, but higher hydrogen dose increases it significantly at nominal load.

- Negative correlation of hydrogen addition on unburnt HC and CO is observed only at partial loads. The engine working at full load generates maximal emissions of HC and CO while HES is around 20%.
- In all tests NO, HC and CO emissions from RME7 combustion were higher in comparison to pure RME.
- The smokiness decreased steadily with increase of HES.
- Amounts of hydrogen addition by energy share were limited with combustion knock occurring at nominal load with HES of nearly 35%.

Future works will concentrate on reducing knocking combustion in tests with hydrogen amounts over 50% by energy.

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Nomenclature

List of variables

$dp/d\phi$: pressure rise
p_{max}: in-cylinder maximum pressure
φ: injection timing
λ: relative equivalence air-fuel ratio - lambda

List of abbreviations

aBDC: after bottom dead center
bBDC: before bottom dead center
aTDC: after top dead center
bTDC: before top dead center
ISFC: indicated specific fuel consumption
ITE: indicated thermal efficiency
CA: crank angle
CAD: crank angle degree
CA 0–10: initial combustion duration measured by CAD and determined by positions from SOC to 10% MFB
CA 10–90: main combustion duration measured by CAD and determined by positions from 10% MFB to 90% MFB
C/H: carbon to hydrogen ratio
CI: compression ignition
CO: carbon monoxide
CO_2: carbon dioxide
CN: cetane number
DF: diesel fuel
FAME: fatty acid methyl ester
HES: hydrogen energy share
B/IMEP: brake/indicated mean effective pressure
LL: low load
MFB: mass fraction burned
ML: medium load
NL: nominal load
NO: nitrogen monoxide
NTP: normal temperature and pressure - defined at 20 °C and 1 atm (101 325 Pa)
RME: rapeseed methyl ester
RME+H_2*0%*: RME − hydrogen mixture with HES = 0%
RME+H_2*16%*: RME − hydrogen mixture with HES = 16%
RME7: mixture of 7% (vol.) RME with fossil diesel fuel
RME7+H_2*0%*: RME7 − hydrogen mixture with HES = 0%
RME7+H_2*16%RME7 −*: hydrogen mixture with HES = 16%
SOC: start of combustion
STP: standard temperature and pressure - defined at 0 °C and 1 bar (100 000 Pa)

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