# COMBUSTION OF RME – DIESEL AND NExBTL – DIESEL BLENDS WITH HYDROGEN IN THE COMPRESSION IGNITION ENGINE

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*Abstract*

*The article presents the test results of the single cylinder compression ignition engine with common rail injection system operating on biofuels and conventional diesel blends with hydrogen. Two types of liquid fuels were tested: blend of the 7% Rapeseed Methyl Ester (RME) with conventional diesel fuel and Neste Pro Diesel – blend of the 15% Hydrotreated Vegetable Oil (HVO), produced by Neste Oil Corporation with conventional diesel fuel. The purpose of this investigation was to examine the influence of the hydrogen addition to biofuels and diesel blends on combustion phases, autoignition delay, engine performance efficiency and exhaust emissions. Hydrogen fraction was changed within the range from 0 to 43% by energy. Hydrogen was injected into the intake manifold, where it created homogeneous mixture with air. Tests were performed at both fixed and optimal injection timings at low, medium, and nominal engine load. After analysis of the engine bench tests and simulation with AVL BOOST software, it was observed that increasing hydrogen fraction shortened the fuel ignition delay phase and it affected the main combustion phase. Moreover, decrease of carbon monoxide (CO), carbon dioxide ($CO_2$) and smoke opacity was observed with increase of hydrogen amounts to the engine. However, increase of the nitrogen oxide (NOx) concentration in the engine exhaust gases was observed.*

*Keywords: hydrogen, RME, HVO, NExBTL, PRO Diesel, diesel fuel, CI engine, combustion, emission*

## 1. Introduction

The warming of the climate over the last decades defined as the result of the human activity on ecosystems, natural resources, agriculture, and countryside. The International and local authorities urges to introduce stronger regulations because of continuous global warming, influenced by the combustion of oil, coal and gas. However, these regulations can affect the oil production with

consequences of the economy of the World, because there is a link between oil production and global economic growth; if oil production cannot grow, the implication is that the economy cannot grow either [1]. Recently the industry of the World reached the oil production peak, which means that the supply of oil is no longer able to meet the demand of oil price. "We are not running out of oil, but we are running out of oil that can be produced easily and cheaply," claims the comment of J. Murray and D. King [1].

The recent revolutions in Arab countries, continuous civil wars in the North Africa and Middle East, Iran nuclear crisis, sanctions of US on Russia make concerns about the stability of the energy sources and force the industry to look for sustainable energy sources supplied by reliable domestic suppliers and thus avoid the costly delivery. The tide relation between oil production and global growth of economy or in other words the impact of oil to the economy, and location of the reserves in the certain regions of the World, implicates the research of alternative energy resources.

In 2015, global $CO_2$ emissions reached 32.3 $GtCO_2$, while all kinds of the transport accounted for 7.75 $GtCO_2$. With increased emissions of the road transport by 68% since 1990, it accounts almost for the quarter of the global $CO_2$ emissions – 5.8 $GtCO_2$ [2]. There are few routes to ease the burden of road transport emissions, to add renewable fuel to the fossil fuel or, substitute the fossil fuel with renewable fuel. The use of renewable fuels has the potential to reduce the emissions and, thus mitigate the effects of the environmental crisis of climate change.
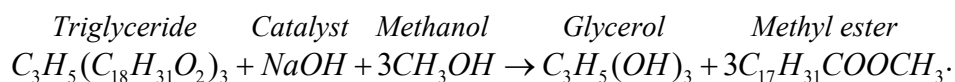
Among the current renewable fuels are alternative biomass based biofuels. The physical properties and the availability in the globe make the hydrogen the other quite attractive alternative fuel for road transport. Although the use of sole hydrogen for combustion engines is hardly possible, the co-combustion with various fuels including renewable fuels makes it the subject of research interest. Moreover, it is widely known, that there is no single fuel solution for the future transport because the availability and cost of alternative fuels differ between the modes. The discussion in this study will be focused on the co-combustion of hydrogen with blends of rapeseed methyl ester (RME) with fossil diesel fuel and hydro treated vegetable oil (HVO) with fossil diesel.

## 2. RME and HVO blends with fossil diesel fuel

Use of biofuels may cause the reduction of $CO_2$ emission, because of the low C/H ratio. Most biofuels today produced from commonly available, edible feedstock and classified as first-generation. They can offer some $CO_2$ benefits and can help improve domestic energy security [4, 5]. Biofuels produced from second-generation biomass does not compete with food production, however high raw material costs are still an issue in making it processes economically attractive [3].

Biomass (vegetable oils and animal fats) consist of a mixture of triglycerides, i.e. esters of glycerol and unsaturated fatty acids. Esterification of triglycerides with alcohol (methanol) in presence of catalyst, gives a mixture of fatty acid methyl esters (FAME) and glycerol. The alternative to the esterification of triglycerides is the hydro treating of triglycerides. During this process, the HVO produced of the same triglyceride, feedstock used to produce FAME or Rapeseed Methyl Ester. However, FAME and HVO are the different products, with different chemical structure and physical properties.

The transesterification of rapeseed biomass oil containing the triglyceride ($C_3H_5(C_{18}H_{31}O_2)_3$), involves it reaction with alcohol ($CH_3OH$) in presence of a catalyst ($NaOH$). The final products of this reaction are glycerol ($C_3H_5(OH)_3$) and rapeseed (RME) methyl ester ($C_{17}H_{31}COOCH_3$). During this reaction, the fatty acid in the vegetable oil converted into methyl ester and the by-product, glycerol, is separated:

$$\text{Triglyceride} \quad \text{Catalyst} \quad \text{Methanol} \quad \text{Glycerol} \quad \text{Methyl ester}$$
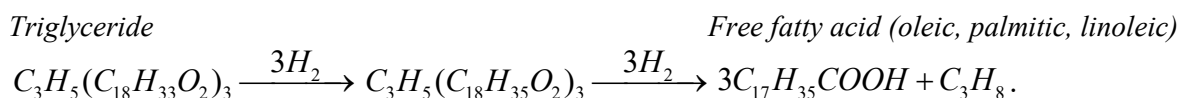$$C_3H_5(C_{18}H_{31}O_2)_3 + NaOH + 3CH_3OH \rightarrow C_3H_5(OH)_3 + 3C_{17}H_{31}COOCH_3.$$

For the engine tests usually are used pure FAME or it blends with a fossil diesel fuel. 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. For the tests of this study there was used the biofuel purchased from Orlen Południe S.A. (Poland) in accordance to the standard EN 590:2013 and defined at this study as RME7. The properties of RME7 according to the source [11] and Orlen product data sheet presented at the Tab. 1.

*Tab. 1. Comparison of the fuel properties [11, 13, 30]*

| Properties | | Test method | Biodiesel RME7 | PRO Diesel | Hydrogen |
|---|---|---|---|---|---|
| Chemical formula | | – | $C_{14}H_{24.2}O_{0.8}$ | $C_{15}H_{32}$-$C_{18}H_{38}$ | $H_2$ |
| Composition, %wt | | – | 81.9 C, 11.9 H, 6.2 O | 84.8 C, 15.2 H | 100 |
| Density, kg/m³ at 15ºC and 1.01 bar | | EN ISO12185 | 838.7 | 826 | 0.08985 |
| Molar mass, g/mol | | – | 205.3 | 212.4-254.5 | 2.016 |
| Lower heating value, MJ/kg | | ASTM D4809 | 42.14 | 43.2 | 120 |
| Lower heating value, MJ/Nm³ | | ASTM D4809 | 36095 | 35750 | 10.7 |
| Stoichiometric air-fuel ratio, kg/kg | | – | 14.35 | 14.56 | 34.3 |
| Stoichiometric air-fuel ratio, Nm³/Nm³ | | – | 10032 | 9577 | 2.6 |
| Heating value of stoichiometric mixture, | MJ/kg | – | 2.74 | 2.78 | 3.40 |
| | MJ/Nm³ | – | 3.60 | 3.73 | 3.00 |
| Auto ignition temperature at STP, ºC | | – | ~260 | ~210 | 585 |
| Flammability limits at NTP, %vol | | – | 0.6-7.5 | – | 4-77 |
| Laminar flame speed, m/s at NTP | | – | 0.3 | – | 2.65-3.25 |
| Kinematic viscosity, mm²/s at 40ºC | | EN ISO3104 | 3.5-5 | 3 | 118 |
| Aromatics, %wt | | EN 12916 | 23.2 | 13.6 | – |
| Cetane number (CN) | | ASTM D6890 | 51.7 | 60 | 5-10 |
| Carbon to hydrogen ratio (C/H) | | – | 6.9 | 5.6 | – |

HVO produced by catalytic hydro processing of the same triglyceride feedstocks used to produce RME [6]. During this process, an alcohol is not required, the products are hydrocarbons rather than fatty acid alkyl esters, and no glycerol by-product is formed [7]. During the catalytic hydro processing of triglycerides at the first step, free fatty acids are formed from the triglyceride molecules in presence of hydrogen. One mole of propane and three moles of oleic, palmitic, and linoleic fatty acid are formed, respectively:

*Triglyceride*                                           *Free fatty acid (oleic, palmitic, linoleic)*

$$C_3H_5(C_{18}H_{33}O_2)_3 \xrightarrow{3H_2} C_3H_5(C_{18}H_{35}O_2)_3 \xrightarrow{3H_2} 3C_{17}H_{35}COOH + C_3H_8 .$$

In the second step, hydrogenation takes place to saturate the oleic and linoleic acids, because the side chain of palmitic acid is already completely saturated. Then, the three common reported reactions to eliminate oxygen may occur:

– Decarboxylation: $\quad C_{17}H_{35}COOH \rightarrow C_{17}H_{36} + CO_2$,

– Decarbonylation: $\quad C_{17}H_{35}COOH + H_2 \rightarrow C_{17}H_{36} + H_2O + CO$,

– Hydrodeoxygenation: $\quad C_{17}H_{35}COOH + 3H_2 \rightarrow C_{18}H_{38} + 3H_2O$.

Decarbonylation and decarboxylation form hydrocarbons having one carbon atom less than the parent free fatty acid (FFA) whereas hydrodeoxygenation removes the oxygen atom keeping the same carbon atoms as in the original FFA. In this way, the fully saturated hydrocarbon linear paraffins are comprised in the range of $C_{15}$-$C_{18}$.

The Finish scientists together with Neste Corporation developed the modern, renewable, oxygen-free, low-emission diesel fuel NExBTL (Nest's trademark for HVO process) produced from second-generation feedstocks: palm, soybean, rapeseed oils, as well as waste and residue fat fractions coming from food, fish and slaughterhouse industries [8].

Unblended NExBTL meets EN 15940:2016 requirements for paraffinic diesel fuels and European diesel fuel standard EN 590:2013 in all respects except density, which is below the lower limit of 820 kg/m$^3$ [8]. Too high CN, low cold flow property and low lubricity are the properties of NExBTL that limits its capability to replace fossil DF. Blending of the NExBTL with fossil DF is the solution to decrease the CN and raise the lubricity so that it can be used with no engine modifications [27]. For the tests of this study there was used the PRO Diesel (Nest's trademark) purchased from NESTE petrol station in Lithuania. PRO Diesel is blend of 15% (vol.) NExBTL with fossil diesel fuel. The properties of hydrogen according to the source [30] and PRO Diesel according to the source [13] and NESTE product data sheet presented at the Tab. 1.

During the tests [9] of RME20 at low to moderate engine speed, the BSFC was higher by 11.9% in compare to fossil DF. The BTE at the low speed of 1400 rpm was 0.392 with RME20, in compare to 0.373 with DF. However, with increase of speed fuel energy conversion efficiency slightly is reduced to the value of 0.354. The NO$_x$ emissions generated from engine running on the RME20 at the speed 1400-2200 rpm increased from 1.4%, to 44.7%. The CO at a fully opened throttle was lower on 29.2-33.6%.

Lebedevas et al. [10] revealed that the performance and emission indicators of the diesel engine running on blends of biofuel RME30 with fossil DF was negligible. However, the properties of fuel containing *Camelina sativa* oil methyl esters makes considerably worse than RME in the course of the storage. Labeckas et al. [11] conducted experimental study with CI engine with 5-15 vol% amount of ethanol blend with DF, and ethanol (15 vol%), DF (80 vol%), RME (5 vol%) blend (E15B). They revealed that the oxygen mass content of the mixture reflects changes of the auto ignition delay time caused by the use of E15B blend more predictably than the CN (cetane number) does. The auto ignition delay for mostly oxygenated blend E15B was 15.4% longer than DF (0.78 ms) and the BSFC was higher. The lower emissions of NO$_x$, CO, HC emissions along with positive changes in and opacities of the exhaust revealed an importance of the fuel oxygen bound.

When using the 15% NExBTL additive to DF (PRO Diesel) and with the advanced start of injection to the maximum engine efficiency [12], smokiness decreased by 40%, and the HC and CO$_2$ decreased as well; however, the NO$_x$ increased by 80%. However, test results of Ewphun et al. [17] showed that further increase of HVO fraction at the fuel content decreasing NO$_x$ and as well the soot concentration, because of shorter auto ignition delay and lower flame temperature. During the tests performed by Pirjola et al. [23], the substitution of DF with HVO, the emissions of NO$_x$ reduced by 20% and PM by 44%. Bhardwaj et al. [16] noted that the HVO fuelling resulted in about 50% reduction in smoke emissions and 43% reduction in gravimetric PM flow. These and other studies [13, 18-22] conducted with the CI engines fuelled with HVO showed that HVO has potential to reduce NO$_x$, soot emissions and deposit formation in the cylinder.

Experiments carried out on the CI engine [14] with fossil DF blended with 20-vol% of RME and HES of 0–5%, revealed the lower engine performance, efficiency, and emissions except the NO$_x$, which slightly increased. Addition of hydrogen to the fuel blend, the CO emissions, smoke, and total unburned hydrocarbon emissions (THC) decreased, while the NO$_x$ kept the same increasing trend. The addition of hydrogen has not a significant effect on auto ignition delay.

During the investigation [24] performed on a diesel – generator, hydrogen was supplied with HES of 5-24% to the diesel – biodiesel (7%) blend (B7). With increase of the HES, CO$_2$, CO and HC emissions decreased. However, the NO$_x$ increased due to the increase of the in-cylinder temperature. There was also noticed an increase of the peak pressure and heat release rate, since B7 ignition delay was reduced due to increase of the HES.

No significant increase of $NO_x$ was observed during test conditions of the CI engine with EGR [15] running with fossil DF blend with 7% of FAME. However, HES of 25% caused the reduction of $p_{max}$ and decrease of $CO_2$ emission by 22% at the lowest load and middle speed. Tests carried out with amounts of HES = 5% [26] fossil DF, shortens the CI engine ignition lag and, decrease the rate of pressure rise. With a hydrogen share of 5–15%, the entire combustion duration does not change significantly, but with hydrogen share of 15% $p_{max}$ increased. With HES of 17%, the combustion knock starts and with HES of 25%, the fast combustion accompanied by combustion knock.

These reviewed experiments either performed on of the CI engine, with sole HVO, PRO Diesel or with addition of hydrogen to the fossil DF or FAME fuel blends. There are gaps in knowledge dealing with hydrogen co-combustion with RME7 and PRO Diesel in the CI engine. The hydrogen co-combustion with RME7 and PRO Diesel were investigated with fixed start of DF injection, to find the influence of hydrogen presence and ways to reduce the emissions, whilst maintain the engine efficiency.

## 3. Purpose of the Research

The purpose of this research is to conduct the analysis of the effect of neat hydrogen co-combustion with two biomass based fuel blends of RME7 and PRO Diesel on the performance and emissions parameters of a CI engine operating at a Low, Medium, Nominal Loads and with fixed start of DF injection. The objectives of this research may be stated as follows:
–   determine the effect of hydrogen fraction on co-combustion with RME7 and PRO Diesel on the in-cylinder pressure, *BTE* when running with HES form 0% to 30%,
–   evaluate the effect of HES on co-combustion with RME7 and PRO Diesel on the changes of emissions, including the NO, CO, $CO_2$, smokiness,
–   determine the influence of hydrogen fraction on co-combustion with RME7 and PRO Diesel on the autoignition delay (lag) expressed by the initial combustion duration of CA0-10 and the main combustion duration CA10-90.

## 4. Experimental Set-up and Procedure

Tests have been performed at the Institute of Thermal Machinery of Czestochowa University of Technology, Al. Armii Krajowej 21, Czestochowa in Poland. The single cylinder water cooling stationary CI engine Andoria S320 equipped with high pressure Bosch common rail fuel pump driven by an electric motor. Displacement of the engine – 1810 $cm^3$, compression ratio – 17. After CI engine starts to run, it delivers energy by driving belts to generator. The engine was set to operate at the constant speed of 965 rpm $\pm$ 0.83%. The experimental set-up presented at the Fig. 1.

Each experiment conducted at the various *IMEP* (Indicated Mean Effective Pressure). The *IMEP* managed by changing the liquid and gaseous fuel ($H_2$) supply to the combustion chamber. The hydrogen supplied together with air into the intake manifold. In cylinder, air – hydrogen mixture under the elevated heat and pressure self-ignited by injected diesel fuel. The diesel fuel consumption was measured by stopwatch. Hydrogen was supplied into the engine intake manifold out of the balloon with a one-stage pressure regulator to reduce its pressure to 1 bar, which was the pressure of the hydrogen gas supply line. A firebreak valve was installed just upstream of the air intake manifold to prevent flashback phenomenon.

Pollutants in the exhaust gas were analysed using *Bosch* and *Maha* (smoke) analysers. In-cylinder pressure (*p)* fixed by piezo sensor *Kistler* 6061B installed instead of the preheating plug. The crank angle (CA) fixed by encoder *Kistler* type 2612C. The data acquisition converter *Measurement Computing Corporation* PCI-DAS 6036 was used in line with PC software *SAWIR* – System of the Indicator Chart on Real Time Analysis). Software *InstalCal* was installed to the PCI-DAS 6036 converter for calibration and test as well as self-calibration of the analogue input

and analogue output lines is available. The *SAWIR* data acquisition application was developed using programming environment Delphi 6.0 of Windows operating system.
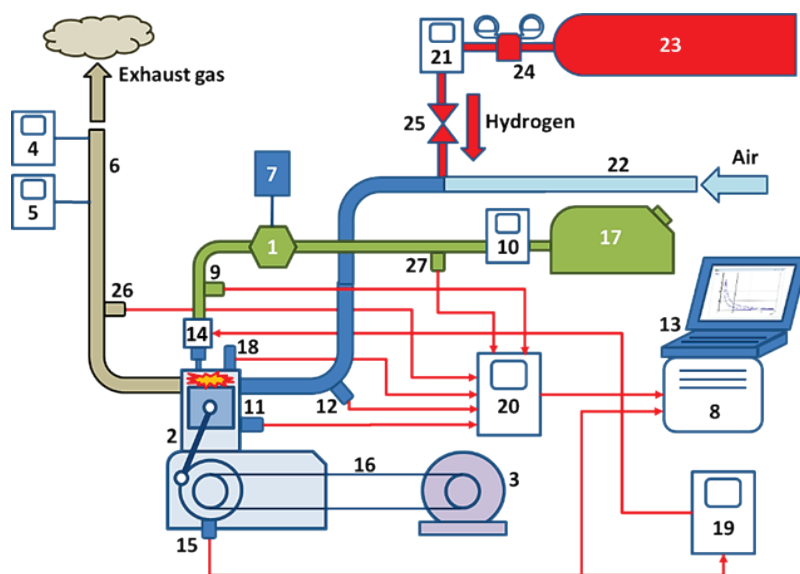


*Fig. 1. Experimental set-up: 1 – DF pump, 2 – CI engine, 3 – generator, 4 – smoke analyser, 5 – emission analyser 6 – exhaust pipe, 7 – DF pump drive el. engine, 8 – data acquisition system, 9 – DF pressure sensor, 10 – DF flow meter, 11 – engine temp. sensor, 12 – inlet air temperature sensor, 13 – PC – SAWIR, 14 – DF common rail injector, 15 – CA 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 balloon 24 – hydrogen one-stage pressure regulator, 25 – hydrogen firebreak arrestor, 26 – exhaust gas temperature sensor 27 – DF temperature sensor*

Two types of pure liquid fuels tested for this experimental study: RME7 and PRO Diesel. Tests of the RME7 were performed under the Low Load (LL) of *IMEP* = 259.4-288.3 kPa (lean mixtures with $\lambda$ = 3.74-4.05), Medium Load (ML) of *IMEP* = 422.3-468 kPa ($\lambda$ = 2.28-2.48) and Nominal Load (NL) of *IMEP* = 576.4-622.5 kPa (close to the rich mixtures of $\lambda$ = 1.57-1.72). While tests of the PRO Diesel were performed under LL of *IMEP* = 367.3-418.7 kPa ($\lambda$ = 2.99-3.37), ML of *IMEP* = 535.8-623.6 kPa ($\lambda$ = 1.88-1.99) and NL of *IMEP* = 592.8-679.3 kPa ($\lambda$ = 1.28-1.41). The max. value of *IMEP* at NL can be considered as the rated value, because the torque at that *IMEP* was 90-95 Nm, 7-13% over the peak torque of 84.4 Nm @ 1200 rpm. As the presence of hydrogen effects the combustion duration, the start of diesel injection timing $\varphi$, during tests was set at the fixed position, enabling to compare and analyse several MFB profiles at various HES.

The injection timing $\varphi_1$ for RME7 operation (Tab. 2, test No. 1) was determined at the position of 50% MFB, which corresponds to the peak of indicative pressure in cylinder. This position was set within the 8-12 deg CA. Other injection timing $\varphi_2$ (Tab. 2, test No. 2) was determined with the lowest HES, again at the combustion of 50% MFB within the range of 8-12 deg CA. Moreover, the same injection timing $\varphi_2$ was used for the rest of hydrogen fractions of 16-29%. At the ML operation $\varphi_1$ = 24 deg (Tab. 2, test No. 3) was determined at the HES = 0%, and injection timing was fixed at $\varphi_2$ = 22 deg with change of the HES 15-29%. At the NL operation $\varphi_1$ = 30 deg was determined at the HES = 0%, and $\varphi_2$ = 26 deg was fixed with change of the HES 14-28%.

The similar procedure with targeting of 50% MFB within the range of 8-12 deg CA was repeated with PRO Diesel. During the operation with PRO Diesel injection, timings $\varphi_1$ and $\varphi_2$ were determined again. Incidentally $\varphi_1 = \varphi_2$, with pure biofuel (HES = 0%) and with supply of hydrogen. This circumstance enables to compare emissions and performance parameters, within the whole range of HES; starting form HES = 0% to the highest values of HES tested during the experiment.

*Tab. 2. Injection timing at various engine loads (IMEP) and composition of RME7, PRO Diesel and $H_2$*

| Test No. | Composition of combustible mixture | Injection timing φ, CAD BTDC | Loads |
|---|---|---|---|
| 1. | RME7+$H_2$ 0% | 18º | Low Load |
| 2. | RME7+$H_2$ (16-29%) | 16º | |
| 3. | RME7+$H_2$ 0% | 24º | Medium Load |
| 4. | RME7+$H_2$ (15-29%) | 22º | |
| 5. | RME7+$H_2$ 0% | 30º | Nominal Load |
| 6. | RME7+$H_2$ (14-28%) | 26º | |
| 7. | PRO Diesel+$H_2$ 0% | 18º | Low Load |
| 8. | PRO Diesel+$H_2$ (17-27%) | 18º | |
| 9. | PRO Diesel+$H_2$ 0% | 26º | Medium Load |
| 10. | PRO Diesel+$H_2$ (13-26%) | 26º | |
| 11. | PRO Diesel+$H_2$ 0% | 28º | Nominal Load |
| 12. | PRO Diesel+$H_2$ (13-24%) | 28º | |

## 5. Analysis of the results and discussion

The analysis of the experiments presented in the study based on in-cylinder pressure data acquisition. The 200 consecutive engine-working cycles of each combustible mixture and HES collected for the analysis. The impact of HES on combustion properties and combustion duration of the CI engine operating with the RME7 and PRO Diesel under fixed injection timing at three engine loads were studied.

In-cylinder maximum pressure curves for various hydrogen fractions provided at the Fig. 2. The increase trend of in-cylinder maximum pressure ($p_{max}$) was noticed only with increase of HES more than 20% for RME7 with all three-engine loads. While for PRO Diesel, in-cylinder maximum pressure increased steadily and $p_{max}$ was higher than with RME7 with increase of HES. The $p_{max}$ with PRO Diesel was higher than that with RME7 at the whole test range of hydrogen fraction starting from 0%, because the start of combustion with PRO Diesel takes place earlier than with RME7 and pressure rise faster (Fig. 14-15). The other reason of that phenomenon is that heating value of PRO Diesel stoichiometric mixture (on both volume and weight basis) is higher than that of RME7 (Tab. 2).
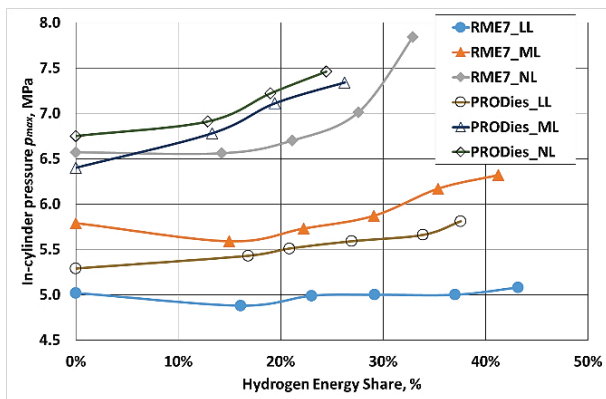


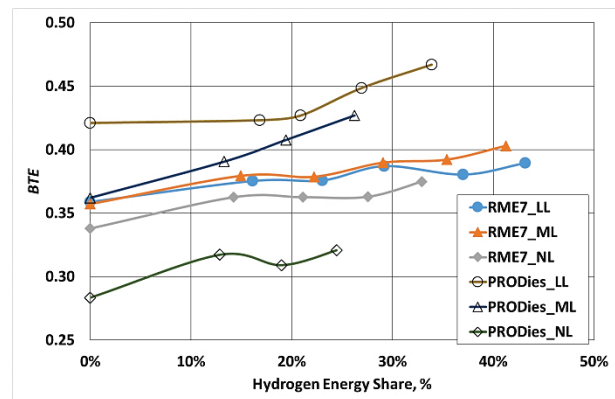| Fig. 2. In-cylinder max pressure at Low, Medium and Nominal Loads and HES | Fig. 3. The dependence of BTE on fuel used loads and HES |
|---|---|

The *BTE* increased steadily with increase of the HES with both tested fuels (Fig. 3), and increase that is more significant with PRO Diesel noticed. The increase of *BTE* was 8-11% with RME7 and 12-19% with PRO Diesel. The efficiency of the engine was higher with PRO Diesel.

The possible increase in engine efficiency might be because of the higher cetane number (Tab. 1) of the PRO Diesel and therefore shorter ignition delay [13]. The significantly low efficiency at NL is the result of the measurement uncertainties, as well as the position of efficiency curve of ML (between the LL and NL) of tests with RME7. The lower auto ignition temperature of PRO Diesel (210ºC), than RME7 (260ºC) influenced the earlier SOC and shorter auto ignition delay. The lower heating value of the fuel mixture did not changed with increased engine load, and therefore did not make any significant effect on *BTE*. The biggest influence on the *BTE* was made by increased mass flow rate of the hydrogen as the response to the decreased total fuel mass flow rate by 5% at the LL and by 14% at the NL.

In-cylinder pressure curves for PRO Diesel+$H_2$0%, PRO Diesel+$H_2$27% at LL and PRO Diesel+$H_2$0%, PRO Diesel+$H_2$24% at NL are provided at the Fig. 4 with the position of SOC (start of combustion) of corresponding mixtures. The SOC was taken at crank angle at which the curve of the ROHR (Rate of heat release) changes its value from the minus side to plus one. For the accuracy of the evaluation, 200 single-cycle in-cylinder pressure diagrams were recorded. Therefore, the SOC was averaged over 200 combustion cycles. The position of the SOC was verified after AVL simulation (Fig. 16-17). In-cylinder pressure curves for RME7+$H_2$0%, RME7+$H_2$29% at LL and RME7+$H_2$0%, RME7+$H_2$24% at NL are provided at the Fig. 5 with the position of SOC.
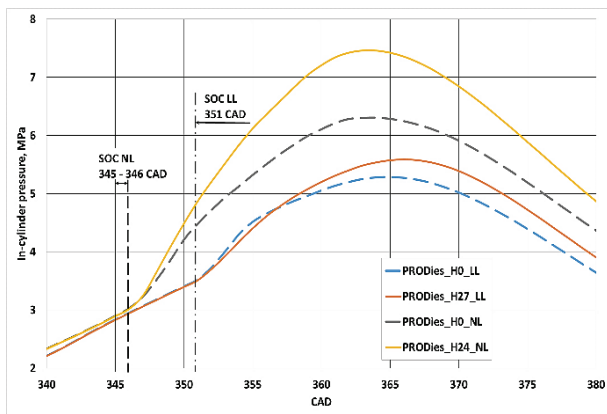


Fig. 4. In-cylinder pressure and SOC at Low and Nominal Loads with PRO Diesel
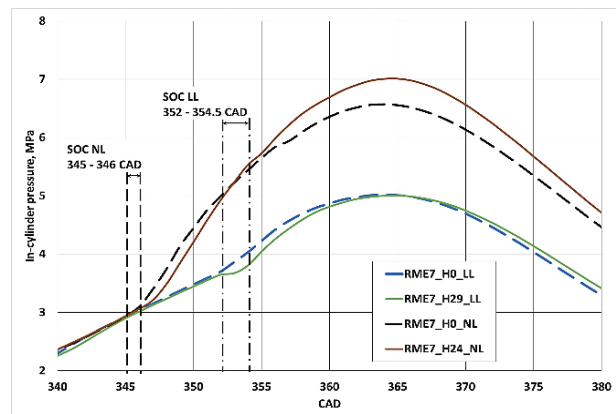
Fig. 5. In-cylinder pressure and SOC at Low and Nominal Loads with RME7

The SOC of 351 CAD (Fig. 4) was set at the LL with PRO Diesel+$H_2$0% and PRO Diesel+$H_2$27% operation, 346 CAD at the NL with PRO Diesel+$H_2$0% and 345 CAD at the NL with PRO Diesel+$H_2$24%. At the LL, the SOC of 352 CAD (Fig. 5) was set with RME7+$H_2$0% and SOC of 354 CAD was set with RME7+$H_2$29%. At the NL, the SOC of 345 CAD was defined with RME7+$H_2$0% and 346 CAD with RME7+$H_2$24%. The SOC of RME7 at both loads without presence of hydrogen takes place earlier as it can be seen at the Fig. 5, because SOI was earlier too (Tab. 2, test No. 1, 2, 5 and 6).

The increase of in-cylinder maximum pressure with RME7 at LL was negligible in compare to the co-combustion with hydrogen mode RME7+$H_2$29% (Fig. 5). The negligible influence of hydrogen fraction at the LL and partially at the ML can be explain by the low volume fraction of the hydrogen in the combustion chamber. The volume fraction of $H_2$ at LL tests are within the range of 1.47-2.79%, which is below the lower flammability limit of hydrogen – 4% (Tab. 1) and the combustion was not intensified by presence of the hydrogen. The lean hydrogen – air mixture does not support the flame propagation and results in rather low hydrogen combustion efficiency [25]. The boundary of 4% was exceeded, with HES of 29% at ML and 21% at NL. The combustion became more intensive, especially at the premixed phase and in-cylinder $p_{max}$ increased.

AVL BOOST simulation [31] revealed that the hydrogen fraction at the LL do not affects significantly the combustion intensity of the fuel mixture. The lean hydrogen – air mixture does not support the flame propagation and results the slow increase of the pressure-rise, plotted at the Fig. 6 and 7. The more significant increase of the pressure-rise because of HES noticed at the NL. The combustion became more intensive at the premixed phase and *dp/dφ* increased from 0.26 MPa/deg to 0.29 MPa/deg with RME7 and from 0.24 MPa/deg to 0.32 MPa/deg with PRO Diesel.
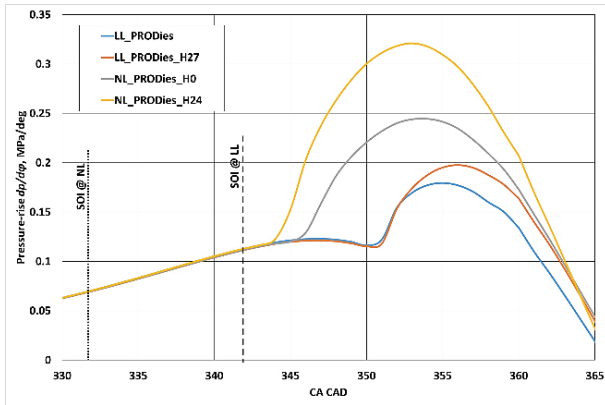


| *Fig. 6. The pressure-rise history at LL and NL* | *Fig. 7. The pressure-rise history at LL and NL* |
| *with PRO Diesel (AVL BOOST simulation)* | *with RME7 (AVL BOOST simulation)* |
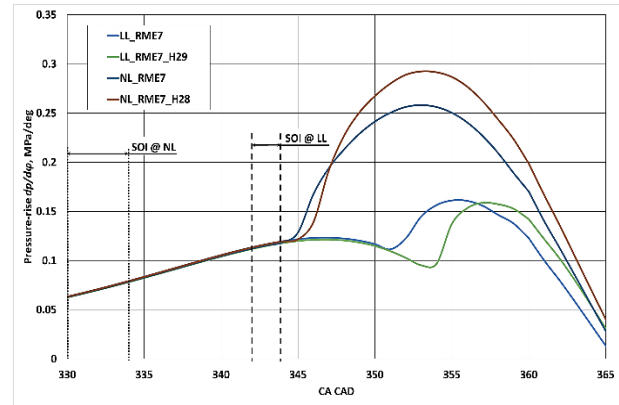
The *p_max* increased with PRO Diesel by 5.4% in compare to the co-combustion with hydrogen mode PRO Diesel+H$_2$27% at LL (Fig. 4). The volume fraction of H$_2$ at LL tests with PRO Diesel is below the lower flammability limit of hydrogen; it was within the range of 1.59-3.19%. However, due to the lower auto ignition temperature of PRO Diesel (210ºC) than RME7 (260ºC) the SOC occurs earlier and the auto ignition delay makes shorter with PRO Diesel.

The auto ignition delay is amount of physical and chemical delay, related to the physical and chemical properties of the fuel. The physical delay is the time required for fuel droplets to penetrate into the compressed air of combustion chamber and evaporate. The chemical delay is the time required for chemical reactions [11]. The longer auto ignition delay and slow increase of the pressure-rise for the RME7 could be caused by the physical delay of the fuel, i.e. higher density, viscosity, and slower evaporation of RME7 droplets. HVO is not oxygenated fuel with the lower density and kinematic viscosity than that of RME7 and fossil DF [27].

The PRO Diesel used during the test, has 23.2% lower C/H ratio in compare to the RME7 (Tab. 1). The increased hydrogen increment rate caused the decrease of C/H ratio and that causes the reduction of CO, CO$_2$ emission (Fig. 8-9) as well as reduction of smokiness [14, 15, 24].
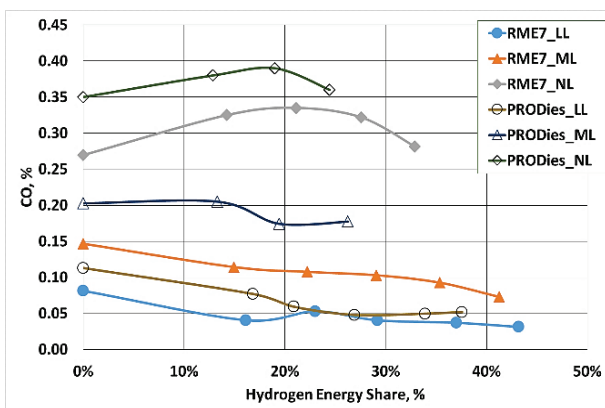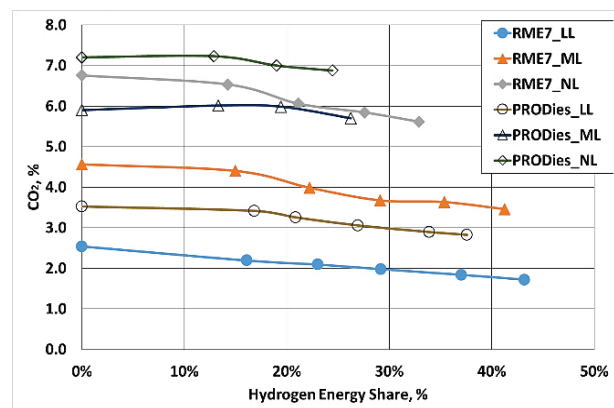


| *Fig. 8. The CO emission vs. HES* | *Fig. 9. The CO$_2$ emission vs. HES* |

The amount of smokiness depends on the CN, soot particles formation and burning rate set up by diffusion phase at the final combustion stages depending on the chemical structure and amount of the fuel injected [11]. However, with increase of hydrogen fraction, the combustion became more intensive at the premixed phase, the burning rate at the diffusion phase as well as the combustion duration (CD) makes shorter and that can be related to the decrease by 44-66% of smokiness with increase of HES (Fig. 11). The smokiness measured for the PRO Diesel was lower than for RME7, because of chemical structure and physical properties of the fuel, i.e. lower density and kinematic viscosity than that of RME7. According to investigations [28], the NExBTL presents smaller SMD (Sauter mean diameter) than fossil DF and biofuels. The PRO Diesel being the blend of NExBTL and fossil DF has smaller SMD than RME7 and better-homogenized mixture guiding to the lower smokiness.

Such significant decrease of smokiness confirms the statement that with increase of HES the air and fuel mixture makes better premixed. The increase of hydrogen fraction in the mixture enhances the ROHR, temperature and thus contributes the conditions for the formation of the NO (Fig. 10).
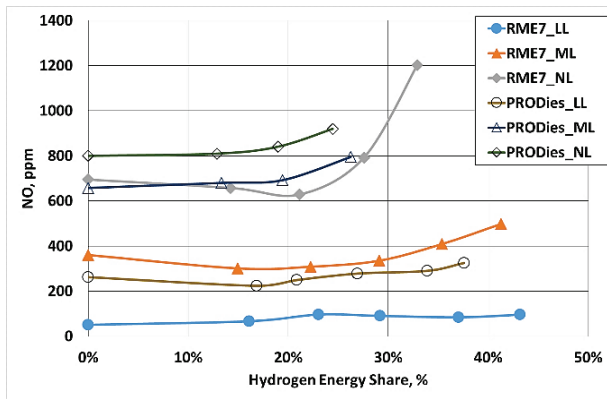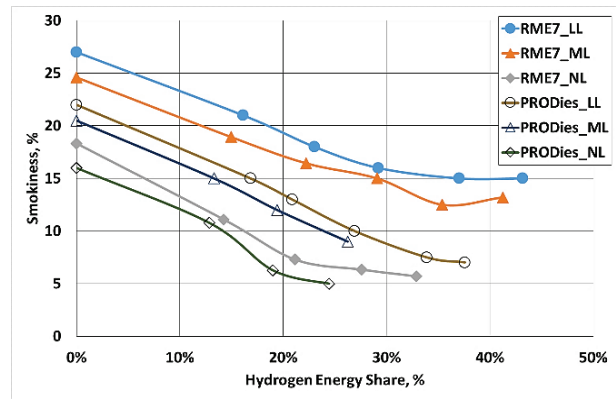


*Fig. 10. The NO emission vs. HES*



*Fig. 11. The smokiness vs. HES*

The ROHR as pressure-rise over CA was determined by AVL BOOST simulation software of engine cycles and gas exchanges [31]. The low HES at the low loads do not enhance the NO formation. The increment of hydrogen fraction generated the increase of maximum ROHR by 17.6-20% with RME7 and 18.8-24% with PRO Diesel (Fig. 12-13). This gain of ROHR determines the increase of NO from 657 ppm with HES = 14% to 792 ppm with HES of 28% at the NL with RME7 (Fig. 10) and from 810 ppm with HES = 13% to 920 ppm with HES of 24% at the NL with PRO Diesel blend was noticed [15, 24].
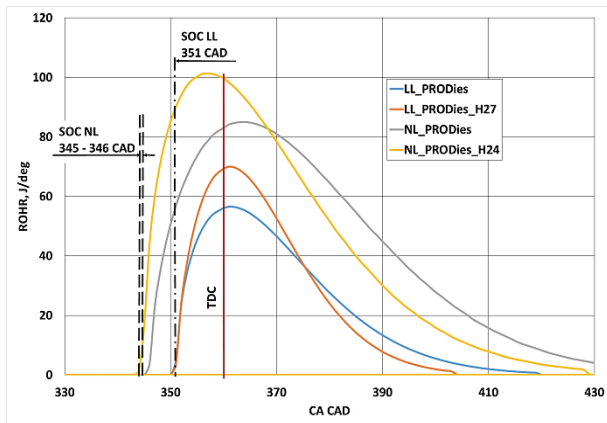


*Fig. 12. The ROHR history at LL and NL with*
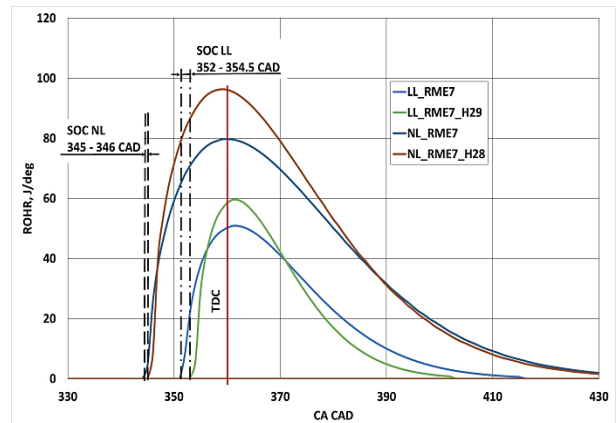*PRO Diesel (AVL BOOST simulation)*



*Fig. 13. The ROHR history at LL and NL with*
*RME7 (AVL BOOST simulation)*

The possible reason of higher NO emissions of RME7 in compare to PRO Diesel can be related to the earlier SOC that may occur because of its higher CN (Tab. 1) and more advanced injection timing (Tab. 2). According to well-known chain reactions mechanism developed by Zeldovich, the NO built up beyond the flame from free atoms of $N_2$ and $O_2$ at the temperatures above the 2200 K. The start of autoignition with PRO Diesel occurs earlier than that of RME7. The increase of hydrogen fraction enhance the advancement of autoignition and the maximum temperature of burned gases increases and consequently the higher levels of NO emission can be expected [29].

The MFB was calculated as the cumulative heat release. The combustion duration of CA0-10 and CA10-90 was calculated at each *IMEP* and with each hydrogen fraction for RME7 and PRO Diesel according to the MFB profiles.

The increase of HES shortens the ignition delay (lag) expressed by the initial combustion duration of CA0-10 (Fig. 14 and 15) due to the high premixed combustion rate and impact of higher laminar speed of hydrogen flame. Increase of hydrogen fraction also reduces the main combustion duration CA10-90 (Fig. 16 and 17) which was accelerated by the first combustion phase CA0-10.
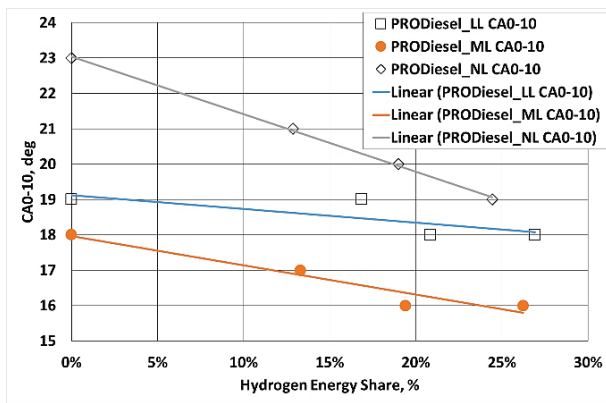


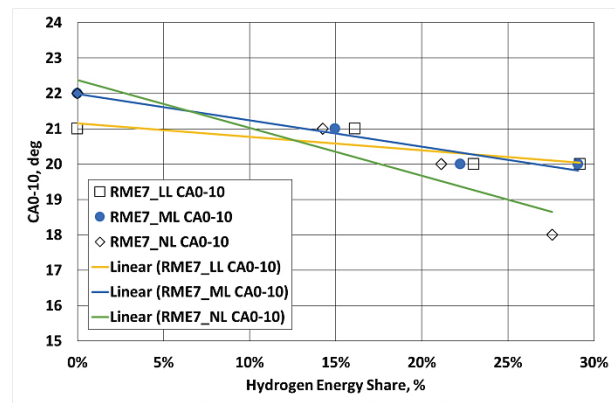*Fig. 14. The combustion delay 0-10% MFB at tested HES of PRO Diesel*



*Fig. 15. The combustion delay 0-10% MFB at tested HES of RME7*
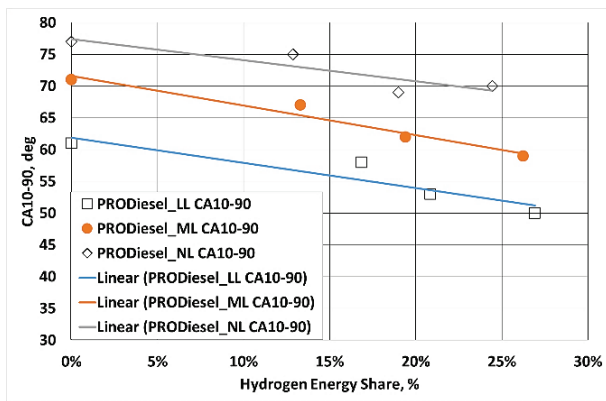


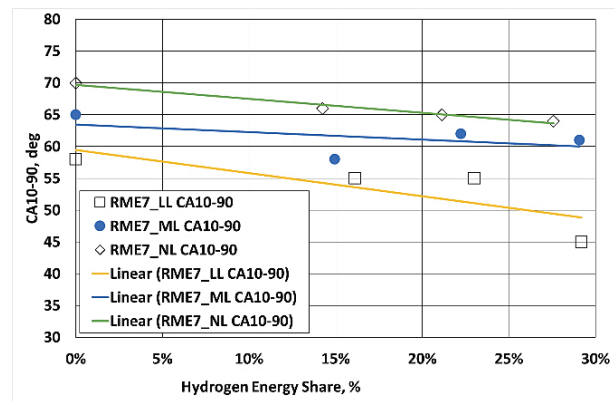*Fig. 16. The combustion delay 10-90% MFB at tested HES of PRO Diesel*



*Fig. 17. The combustion delay 10-90% MFB at tested HES of RME7*

The range of increment of the CA0-10 with PRO Diesel embraces wider range of CA than that of RME7. The combustion duration of CA0-10 varying with PRO Diesel within the range of 16-23 CAD, while with RME7 it was shorter: 18-22 CAD. The duration of CA10-90 takes almost the same 25-27 CAD with both tested fuels. The higher CN and smaller SMD of the PRO Diesel leads to the better homogenized mixture and more intensive combustion at the premixed phase of MFB at the CA0-10. The further combustion process takes almost the same duration

of CA10-90 with both fuels tested and influence of hydrogen fraction is negligible as it was combusted at the CA0-10.

## 6. Conclusions

The main following conclusions can be drawn from the presented analysis:

1. The increase trend of $p_{max}$ was noticed with increase of HES for RME7 and PRO Diesel. The $p_{max}$ with PRO Diesel was higher than that with RME7 at the whole test range of HES because the start of combustion with PRO Diesel takes place earlier than with RME7 and pressure rise faster;

2. The *BTE* at the LL and with HES = 33% reaches the top efficiency value of 0.467 for PRO Diesel fuel. As soon as the engine load increases to the rated value, the fuel energy conversion efficiency reduced to the maximum value of 0.32 at NL and HES = 24%;

3. The increased hydrogen increment rate caused the reduction of CO, $CO_2$ emissions and reduction of smokiness. The increase of HES enhances the temperature, ROHR and contributes the increase of the NO;

4. The HES from 10 to 20% was found as the optimal percentage with respect to overall toxic emissions. However, *BTE* increases remarkably, while HES exceeds 20%;

5. The fuel hydrogen mass content reflects the changes of the combustion phases: the main combustion phase and auto-ignition delay time. As observed, ignition delay expressed by CA0-10 changed from 23 to 19 CAD with PRO Diesel and from 22 to 18 CAD at NL. This is significant difference, which requires injection timing to be modified to optimal value. CA10-90 was also shorter with increase of HES that makes combustion closer to constant volume conditions, what leads to increase in engine thermal efficiency;

6. AVL BOOST simulation confirmed the usability of AVL software for verifying several hypothetical assumptions to theoretical analysis;

7. Supply of hydrogen by port fuelled strategy to the CI engine with both tested fuels was found as the effective way for improvement of the engine performance and reduce toxic exhaust emissions.

The test results show that hydrogen fraction could be used for the enhancement of the CI engine. The addition of hydrogen to the commercially used DF blends RME7 and PRO Diesel suggests ecological advantages and increases the role of the renewable biofuel that is one of the targets recommended by the Directives of EC.

## Acknowledgments

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