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Comparative analysis of IC diesel engine performance fueled with diesel/hydrogen and diesel/ammonia mixtures

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The paper presents the results of research performed on an internal combustion diesel engine operating on a dual-fuel strategy. Fuels used during the study were diesel as a reference fuel, pure hydrogen, and ammonia as a hydrogen carrier. Fuel mixtures of diesel/hydrogen and diesel/ammonia were added to the engine, at energetic shares of 8, 12, 22, and 32% to maintain the constant engine load compared to the engine fueled with diesel. A comparison of the engine performance and the combustion process was made. Under the test conditions for all mixtures, the indication of the combustion pressure was made. The indication results were used to calculate engine performance parameters, such as IMEP, engine power, COV_{IMEP}, and thermal efficiency. They were also used to describe the combustion process, including the determination of parameters like combustion stages, 50% MFB position, and HRR. The analysis of the results shows that adding hydrogen or ammonia as a main fuel causes a much faster combustion process compared to a diesel fuel engine. With an increase in energetic share, the combustion characteristic of the diesel/hydrogen engine shows two maxima in the HRR curve, besides 32% where there is only one peak. In contrast, for the diesel/ammonia engine, there is only one maximum, where an increase in the value of the kinetic combustion phase and the delay of ignition occur.

Key words: *hydrogen, ammonia, co-combustion, combustion analysis, diesel engine*

1. Introduction

The global industrialization based on fossil fuel energy production negatively influences living organisms on our planet [17, 25]. The negative causes of industrialization are related to the emission of toxic and harmful gas compounds, greenhouse gases (GHG), which significantly influence climate change [4, 10]. One of the reasons for this is the use of worldwide transport, which is based on vehicles equipped with internal combustion (IC) engines. To prevent an increase in emissions and reduce them to zero, the European Union has introduced the Fit for 55 program [21]. To respond to that program, it is necessary to switch the technology of the IC engine, which is based on the combustion of fossil fuels, to a clean combustion technology that uses alternative fuels, such as carbon-free fuels. Carbon-free fuels can be characterized as those for which combustion doesn't lead to carbon dioxide (CO_2) emission [3], for this group of fuels, hydrogen and ammonia can be considered.

The hydrogen (H_2) as an energy carrier can be used as a fuel in pure form for IC engines, both spark-ignited (SI) and compressed-ignition (CI), and in fuel cells [8, 9, 22, 26]. Hydrogen as a fuel for IC engines has specific properties that distinguish it from fossil fuels. An important difference is in its atomic structure, where there is no carbon atom, which benefits the decrease in GHG emissions. For the mass unit, it has a very high lower heating value (LHV), which is 120 MJ/kg, which is close to three times higher than for petroleum fuels. From that point of view, it can be recognized as a fuel with big potential as a substitute for conventional fuels. However, as a gaseous fuel, it has a very low density, which implies that for a volume unit, the LHV is around 10.3 MJ/m^3, and if this value is compared with, for example, natural gas (NG), whose LHV is equal to 34.5 MJ/m^3, it is around three times lower. Hydrogen also has a very high laminar flame speed (LFS), which in normal conditions and in stoichiometric mixtures is 2.5 m/s [24]. For natural gas, the LHV value is 0.4 m/s [6], and it is nearly six times lower compared with hydrogen. Hydrogen can be a catalyst that increases the chemical reactivity of co-combusted fuels, and also because of its wide flammability limits, which are from 5 to 75% of volume, it promotes combustion of very lean co-combusted mixtures of fuels. Also, high penetration of the hydrogen flame increases the efficiency of the combustion process itself and leads to more complete combustion. However, there are several problems and challenges with hydrogen use as a fuel for IC engines, which are related to such topics as storage, transportation, and production. The production of hydrogen is generally costly and mostly done by electrolysis or by steam methane reforming. The storage technology used in transportation nowadays is based on high-pressure (350 to 700 bar) or cryogenics (1 atm; –240°C) tanks, which are expensive and advanced technologies [8]. And from that point of view, an interesting alternative can be ammonia (NH_3).

Ammonia, as a hydrogen carrier that can be treated as a fuel, also does not have a carbon atom in its structure. Thanks to that property, it can also be considered as a fuel which leads to a decrease in the emission of one of the GHGs, namely CO_2. But unfortunately, additional nitrogen in the structure promotes the emission of nitrogen oxides (NO_x). The energetic density of ammonia is higher than that of hydrogen. Ammonia compressed to 1 MPa has a volumetric energy density equal to 13.6 GJ/m^3, which is a higher value than methane, which has 10.4 GJ/m^3 at 25 MPa. Laminar flame speed for ammonia is 0.07 m/s, which is very low, and determines the character of the combustion process; it causes an increase in ignition delay [13]. The combustion process of ammonia is prolonged, which causes, in many cases, increased emission of unburnt NH_3 [20, 31], which is also recognized as a greenhouse gas [30]. The flammability limits by volume for ammonia are lower than for pure hydrogen, and they are respectively 15 to 28%, which is wider than for petroleum fuels.

As it was mentioned, most alternative and all petroleum fuels have carbon in their structure, which means that during their combustion, as a result, we get the emission of carbon dioxide, which is responsible, among others, for greenhouse gas emissions (GHGs) [19]. Based on the analysis of hydrogen and ammonia properties, it can be stated that they can be an alternative to carbon-based fuels [16]. Those fuels can be used both in spark-ignition and compression-ignition engines. However, because of the high-octane number, which leads to a low cetane number, they are difficult to ignite in CI engines, and to do that, the pilot injection of high-reactivity fuel, such as diesel oil, should be used. The hydrogen addition increases combustion speed, which is decreased by ammonia, which has, as mentioned earlier, a low LFS compared to other fuels.

Those combined properties cause blends of diesel/hydrogen and diesel/ammonia to be very interesting as an alternative energy source for IC compression-ignition engines. The research related to the combustion of the aforementioned fuel blends is the subject of research performed by many institutions around the world. The results presented by Kanth and Debbarma [11] determined the increase in engine efficiency by 2.5% when it was fueled by a mixture of diesel and hydrogen and 1.6% when it was a mixture of diesel and biodiesel (10% v/v). They also noted a decrease in CO and UHC emissions, but accompanied by an increase in NO_x. Investigation of a dual-fuel engine powered by diesel and hydrogen was presented by Gomes Antunes et al. [7]. In this case, both fuels were directly injected into the engine cylinder. They observed an increase in engine power by 14%, and to achieve stable engine operation, they heat up the intake air. An increase in pressure and efficiency was also noted during those tests. Akra et al. [1] performed the tests with an engine fueled by a biodiesel/hydrogen mixture, and they added hydrogen to get an improvement in the combustion process. They achieved an increase in engine performance and a decrease in exhaust gas emissions except for the NO_x. The co-combustion of hydrogen with diesel and biodiesel and their mixture was tested by Köse and Acaroğlu [12]. As a result, they observed that the highest increase in engine efficiency was for an engine fueled by biodiesel with a 2.5% addition of hydrogen, and for a diesel/biodiesel with a 2.5% hydrogen mixture, maximum power output was achieved. Bjørgen et al. [2] tested an engine fueled with a mixture of diesel and NH_3, where those fuels were separately injected into the engine cylinder. The ammonia energy share was 40, 50, and 60%. The diesel injection timing was constant with the value of 15° bTDC. Advancing ammonia injection 15° before diesel injection causes an increase in ignition delay. Injection of ammonia simultaneously or after diesel injection had a positive impact on the combustion process and the smooth running of the engine. Injection at the same time of diesel and ammonia resulted in the highest efficiency value, lower ammonia slip, and decreased NO_x emission. In the research presented by Nadimi et al. [18], the results of NH_3/diesel mixtures with various ratios were presented. The tests show that increasing the ammonia energetic ratio (AER) switched combustion mode from diffusive (diesel) to premixed combustion (diesel/ammonia) in a dual-fuel engine. Increase in AER changed combustion phases. Addition of NH_3 reduces the emissions of CO_2, CO, and particle matters (PM), at the same time, causes an increase in NO_x and NH_3 emissions. Also, Liu et al.[15] tested a dual-fuel diesel engine where additional fuel, ammonia, was added. The parameters that were changed during those tests were ammonia energy fraction (AEF), diesel injection pressure, and diesel injection timing. As a result of that research, they found that an increase in AEF causes a decrease in peak pressure, an increase in combustion duration, a reduction of NO_x, soot, and CO_2 emissions, which also causes an increase in emissions of unburned ammonia UNH_3 and N_2O.

Besides the aforementioned fuel mixtures, diesel/hydrogen and diesel/ammonia, there is research being performed at an IC compression ignition engine where the mixture of those three fuels, diesel/ammonia/hydrogen, is used. Zhang et al. [32] performed the tests at full load for different energetic rates for ammonia up to 50% and hydrogen up to 8%. They found that the addition of NH_3 improves engine performance, whereas the addition of H_2 improves the combustion of the diesel/ammonia mixture. The addition of NH_3 and H_2 causes a reduction of NO_x, CO_2, and soot, but causes an increase in NH_3 emissions. The research of Wang et al. [29] was performed on a single-cylinder diesel engine at a constant speed, where the energetic fraction of hydrogen was changed from 10 to 90%. The results of those tests were that the optimal hydrogen energetic fraction was 30%, which led to a drop in NH_3 emission by almost 90% but caused an increase in NO_x emission by close to 60%. The tests on a 1-cylinder diesel engine fueled with diesel/ammonia/hydrogen blends were performed by Dhas et al. [5], where ammonia and hydrogen were added to the engine with a constant flow equal to 10 LPM (liter per minute). They found that the addition of NH_3 and H_2 improved brake thermal efficiency, reduced carbon dioxide (CO_2), hydrocarbons (HC), and smoke, but increased NO_x emissions.

As shown in the literature review, the use of diesel/hydrogen and diesel/ammonia mixtures in IC compression ignition engines gives many advantages compared to the combustion of petroleum fuels. These advantages can be observed especially in terms of engine performance, efficiency, and emission of exhaust gases, especially those based on carbon fuel combustion. As a good aspect of hydrogen and ammonia mixing with diesel, it can be pointed out that, as a carbon-free fuel, it is a clean energy source in its own right. Ammonia is much safer to transport and store, and its volumetric energy index is higher than that of hydrogen itself, making it an attractive engine fuel. On the negative side, hydrogen burns very quickly and has a very low ignition energy, which creates problems with combustion control. Rapid combustion is accompanied by large temperature increases, which increase nitrogen oxide emissions. Ammonia, on the other hand, burns slowly, which can be problematic for high-speed engines. The aggressive nature of ammonia requires the use of materials resistant to its effects.

The research presented in this paper was performed on 1-cylinder diesel engines. The engine was running with constant speed and load. The addition of hydrogen and ammonia to the engine was in that manner to ensure the constant load level equal to that of a diesel-fueled engine. During the test, the engine's indication was done, and 100 consecutive cycles for each measured point were stored for combustion analysis. On the basis of measurement, the analysis of the combustion process was done, and the pressure, heat release rate, and normalized heat release curves were presented. The combustion phases were determined for each case, and the engine performance parameters, such as power, indicated efficiency, and COV_{IMEP}, were calculated and presented. The research performed in this paper presents the comparison analysis of combustion parameters between an engine fueled with diesel/hydrogen and a diesel/ammonia mixture. The research was done at the same load conditions and energetic shares of gas fuels, which gives interesting information about the combustion process of those gaseous fuels in a dual fuel engine, especially from fuel chemical properties.

2. Materials and methods

The tests were carried out on a 1-cylinder industrial dual-fuel compression-ignition engine 1CA90 powered by diesel oil and hydrogen or ammonia. The engine was air-cooled and the fuel system provided a constant injection timing of 20°CA bTDC. The engine operated at a constant rotational speed of 1500 rpm.

The engine was equipped with an indication system. An encoder with a resolution of 1 °CA was mounted on the engine crankshaft; this signal was used in the data acquisition system. Data obtained from the measuring track equipped with a pressure transducer (Kistler 6118C) and a charge amplifier (Kistler 5018A) were transferred to a PC using an A/C converter (USB-1608HS). The author's computer program allowed for real-time observation of pressure courses, released heat, HRR, temperature, combustion phases, and IMEP. The engine intake system was equipped with an air filter, a rotary flow meter (CGR-01), and a pressure pulsation damper. Hydrogen/ammonia was fed to the intake manifold in the gas phase, and its volume flow was controlled by a needle valve at the gas reducer. The gas flow was measured by a flow meter (RTU-10-300). A detailed description of the research test stand is presented in studies [27, 28].

The aim of the research was to conduct a comparative analysis of the co-combustion process of H_2/NH_3 with diesel fuel in an industrial compression-ignition engine. The research was carried out for a constant engine load defined as IMEP, whose value was 0.77 MPa. The energy share of fuels was determined from the relationship:

$$\mathrm{HEF/AEF} = \frac{d_{H2/NH3} \cdot LHV_{H2/NH3}}{d_{D100} \cdot LHV_{D100} + d_{H2/NH3} \cdot LHV_{H2/NH3}} \tag{1}$$

where: AEF – ammonia energy fraction, HEF – hydrogen energy fraction, d_{D100} – mass of diesel fuel dose per cycle, $d_{H2/NH3}$ – mass of hydrogen/ammonia dose per cycle, $LHV_{H2/NH3}$ – lower heat value of hydrogen/ammonia, MJ/kg.

IMEP was calculated based on following equation:

$$\mathrm{IMEP} = \frac{1}{V_d}\int_0^{720} p\,\frac{dV}{d\varphi} \tag{2}$$

where: V_d – displacement volume [m^3].

The indicated thermal efficiency can be defined as the ratio of the indicated work in the cylinder volume to the amount of heat supplied to the cylinder.

$$\mathrm{ITE} = \frac{\mathrm{IMEP} \cdot V_d}{Q_{cycle}} \cdot 100\% \tag{3}$$

where: IMEP – indicted mean effective pressure [Pa], Q_{cycle} – total heat supplied to the engine [J].

The heat release rate is one of the parameters that analysis allows to describe the combustion process in an IC engine. This parameter is determined on the basis of the in-cylinder pressure measurement. It can be calculated on the basis of the first law of thermodynamics and the equation of state. A simplification of those equations allows us to calculate the net heat release rate. A net heat release is calculated based on the following equation:

$$\frac{dQ}{d\varphi} = \frac{1}{\kappa-1}\left[\kappa p \cdot \frac{dV}{d\varphi} + V \cdot \frac{dp}{d\varphi}\right] \tag{4}$$

where: κ – the ratio of specific heats [–], V – cylinder volume [m^3], p – in cylinder pressure [Pa].

The indicator of unrepeatability of IMEP was determined based on the following equation:

$$COV_{IMEP} = \frac{\sigma_{IMEP}}{(\mathrm{IMEP})_{mean}} \cdot 100\% \tag{5}$$

where: σ_{IMEP} – standard deviation of IMEP.

Table 1 presents the mass and energy doses of fuels supplied to the engine for the individual analyzed measurement points.

The results show that with the increase in the share of H_2 or NH_3, the share of diesel fuel decreased so that the power generated by the engine remained at a constant level. The increase in the share of H_2/NH_3 caused the engine to require a smaller dose of energy contained in the fuel to generate the same power.

Table 1. Mass and energy doses of fuels

HEF	D100	H_2	$Q_{en(H2)}$	q_{NH2}	Q_{cycle}
	kg/h	kg/h	MJ/h	J/cycle	J/cycle
0%	1.202	0.000	0.00	0.00	1122
8%	1.018	0.0293	3.52	78.72	1029
12%	0.966	0.0467	5.60	124.54	1027
22%	0.827	0.0811	9.73	245.06	1015
32%	0.738	0.1209	14.51	322.50	1012

AEF	D100	NH_3	$Q_{en(NH3)}$	q_{NH3}	Q_{cycle}
	kg/h	kg/h	MJ/h	J/cycle	J/cycle
0%	1.202	0.000	0.00	0.00	1122
8%	1.087	0.219	4.12	91.49	1106
12%	1.044	0.319	6.01	133.58	1108
22%	0.883	0.552	10.38	256.18	1080
32%	0.759	0.788	14.82	329.34	1038

3. Results

The basis for assessing the combustion process in a piston engine is the knowledge of the pressure course in the cylinder. As part of the research, 100 consecutive engine work cycles were recorded, and the average course from the entire set of realizations was analyzed. The engine was characterized by high repeatability of subsequent cycles observed in real time during the research. Each measurement point was repeated 3 times while maintaining thermal stabilization of the engine for each change in the fuel share.

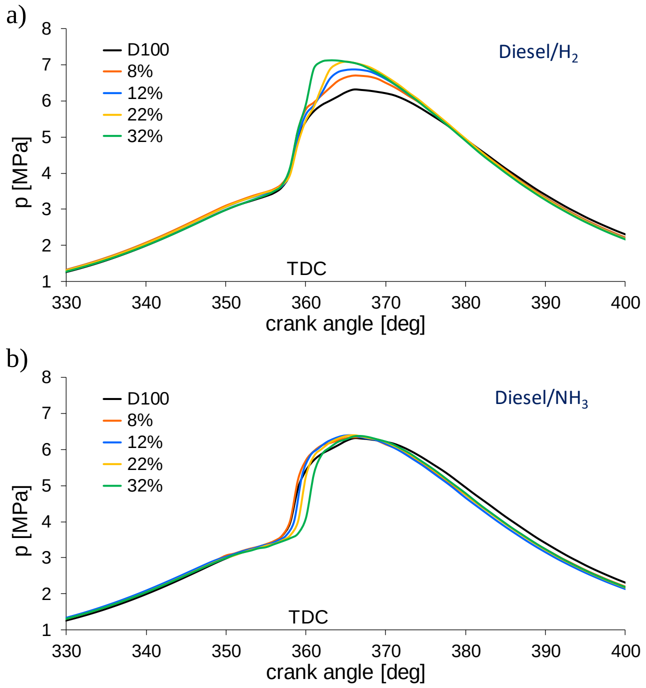

Fig. 1. Pressure curves in the engine cylinder for different H_2 (a) and NH_3 (b) proportions

Figure 1a presents a comparison of pressure courses for a dual-fuel engine fueled with diesel oil and hydrogen, and Fig. 1b with diesel oil and ammonia. It can be seen that with the increase in the share of hydrogen, the peak pressure value also increases, and this value also occurs closer to TDC. That change, because of the hydrogen properties, such as high LFS and diffusion, was expected. For the engine fueled with hydrogen, with its highest energy share, there was an increase in p_{max} by 0.76 MPa compared to the engine fueled with diesel fuel. When fueled with ammonia, the maximum pressure increase was noted for a 12% share of NH_3 and amounted to 0.05 MPa. For the ammonia fuel, a significant delay in the pressure courses is visible compared to the reference case, which is not the case for the hydrogen fuel, and it was caused by the low value of LFS of this fuel. Figure 2 presents the heat release rate courses that are the basis for assessing the combustion process.

From the HRR courses for the hydrogen-fueled engine, it can be seen that with the increase of the H_2 share, up to 22%, the combustion process takes place in two stages, after the first peak on the HRR curve, a second maximum begins to appear. Exceeding this H_2 share caused the combustion to pass into one stage of kinetic combustion with a noticeable disappearance of the diffusion part. Such behavior of a diesel/hydrogen mixture can be caused by an increase in the energetic share of the fuel, which has a low ignition energy value. Consequently, in certain cylinder conditions, the ignition of the hydrogen-air mixture occurs. The increase in HRR_{max} did not exceed 15 J/deg. For the engine fueled with ammonia, the kinetic combustion phase increased by 12 J/deg, which is similar to that of hydrogen. In this case, the nature of the HRR courses is similar for all NH_3 shares. There was a clear reduction in the diffusion combustion phase with the increase of the NH_3 share. Similarly to the pressure courses, a delay in combustion was visible with the increase of the NH_3 share. This was caused, among others, by the increase in the share of fuel with low laminar combustion speed (0.07 m/s) and high heat of vaporization (1370 kJ/kg).

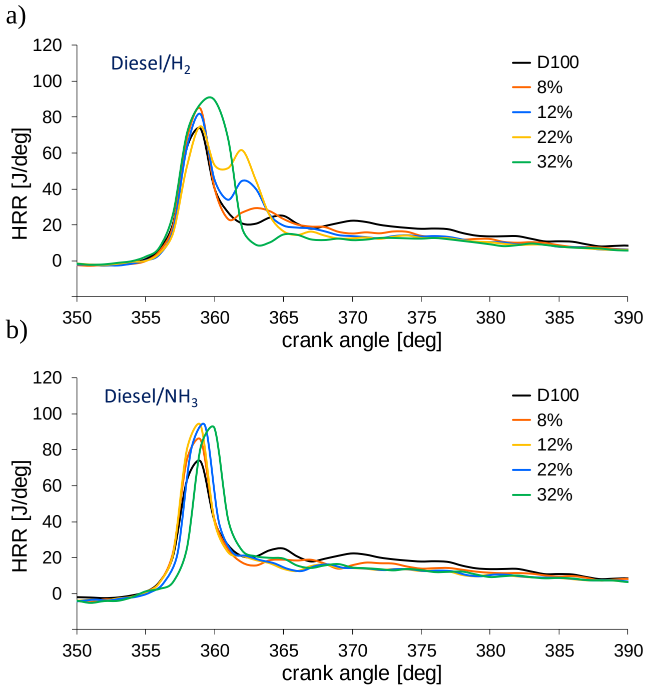

Fig. 2. Heat release rate curves for different H_2 (a) and NH_3 (b) fractions

Based on the heat release rate curves, the heat release curves were determined, which, after normalization, were used to determine characteristic combustion stages. The ignition delay time (CA10) was determined, defined as the time from the beginning of liquid fuel injection to the moment of 10% heat release. The combustion duration (CA10–90) was also determined, defined as the time from the beginning of combustion (CA10) to the moment of 90% heat release (CA90).

The determined heat release courses after normalization are shown in Fig. 3. Fueling the engine with hydrogen clearly affects the heat release course. With an increase in the H_2 share, a clear increase in heat release is visible in the initial combustion phase. Hydrogen, thanks to its high LFS value, intensifies the combustion process. For the engine fueled with NH_3, a change in the nature of heat release is also visible, but here, after the initial intensification of this process, it is clearly slowed down due to the limited flammability of ammonia.

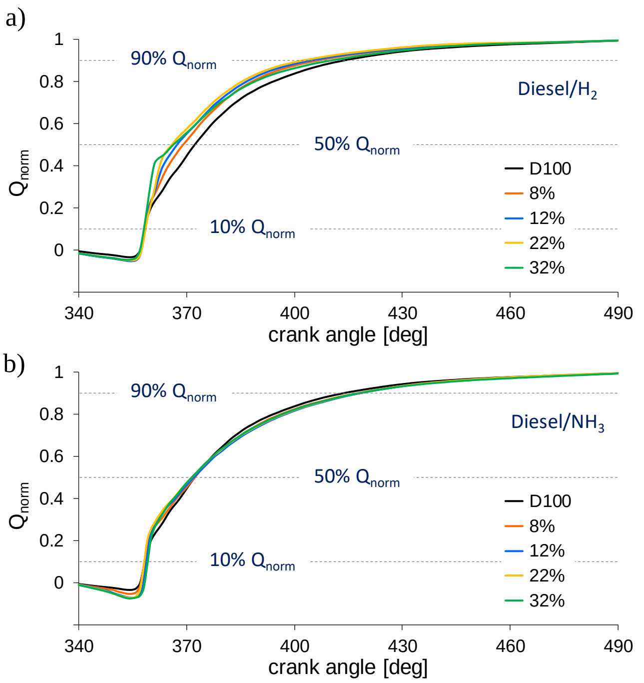

Fig. 3. Normalized heat release curves for different H_2 (a) and NH_3 (b) fractions

In Figure 3 (a and b) characteristic combustion stages are marked, 10% Q_{norm} is the ignition delay time (CA10) and 90% Q_{norm} is the conventional end of combustion time (CA90). The type of fuel is crucial for the initiation and combustion process in both spark ignition and compression ignition engines. It influences characteristic combustion phases, such as ignition delay and combustion duration. For a diesel engine, these phases are most often determined based on the curves of the total released heat.

Figure 4 shows the changes of CA10 and CA10-90 during co-combustion of diesel fuel with hydrogen and ammonia in a dual fuel engine for the analyzed ranges of H_2 and NH_3 shares. The addition of hydrogen to diesel fuel, in comparison with the addition of ammonia, caused a smaller ignition delay and shortened the combustion duration. The smaller CA10 for the diesel/hydrogen engine resulted from providing better conditions for ignition in the engine cylinder due to the improved homogeneity of the combustible mixture containing hydrogen.

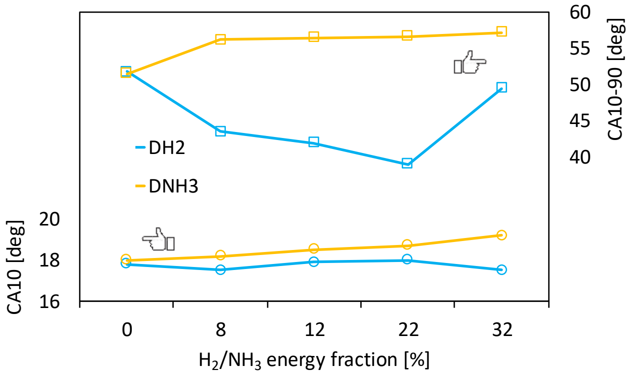

Fig. 4. Characteristic combustion stages: ignition delay (CA10) and combustion duration (CA10-90)

Easier ignition of mixtures with H_2 also resulted from the low minimum ignition energy of hydrogen (0.018 mJ), significantly lower than the minimum ignition energy of ammonia (5–8 mJ). The shorter combustion time of mixtures with H_2 added compared to NH_3 added was mainly caused by the difference in laminar flame speed values for both fuels. High peak laminar flame speed of hydrogen (2.65–3.25 m/s), which is about 40 times higher than the laminar flame speed of ammonia (0.07 m/s), intensifies combustion and helps to shorten CA10–90.

Figure 5 presents a comparison of changes in the IMEP non-repeatability coefficient for a diesel/hydrogen and diesel/ammonia dual-fuel engine. It can be seen that in both cases the COV_{IMEP} values did not exceed the 5% limit permissible for combustion engines and were comparable to the values achieved by a conventional diesel engine. For the engine combusting a mixture with the addition of NH_3, in the entire range of energy shares of this fuel, the COV_{IMEP} values were about 2%. For the engine burning a mixture with the addition of H_2, a slight increase in non-repeatability was noted for the largest 32% share of this fuel. In this case, a large amount of hydrogen in the combustible mixture caused premature ignitions in some cycles, resulting in distortion of the cycles and deepening the differences between them and the average cycle.

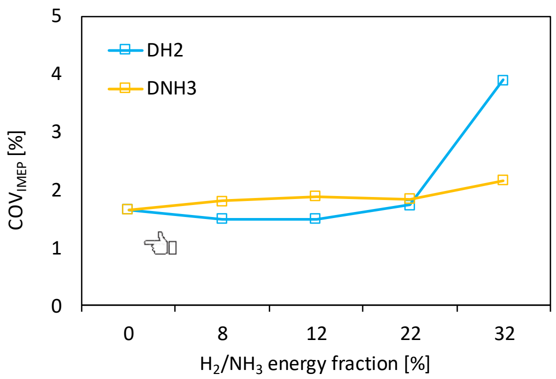

Fig. 5. Coefficient of variation for IMEP

Objective evaluation of the combustion engine operation is also possible thanks to utility indicators such as thermal efficiency (ITE) and specific energy consumption (SEC). Fig. 6 presents the curves of ITE and SEC changes obtained for the diesel/hydrogen and diesel/ammonia engines. They show that the dual-fuel engine worked more efficiently compared to the conventional diesel engine. It can also be seen that the dual-fuel engine with H_2 addition was characterized by higher efficiency and lower specific energy consumption compared to the engine fueled with NH_3. For the diesel/hydrogen engine, the shorter combustion time limited heat losses to the combustion chamber walls and contributed to reduced energy consumption and improved efficiency. For both combusted mixtures, the maximum value of ITE was observed for 12% energy share.

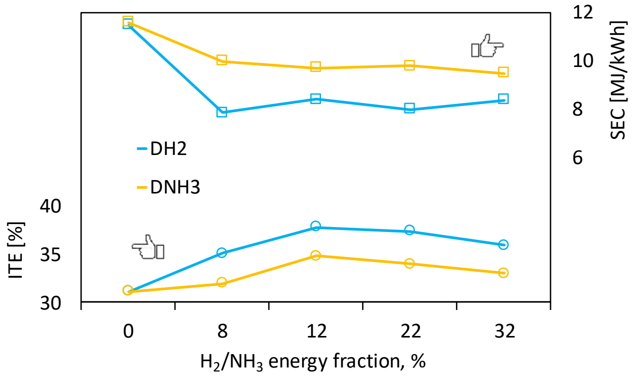

Fig. 6. Engine Thermal Efficiency (ITE) and Specific Energy Consumption (SEC)

When fueling a piston engine with conventional fuels, the basic mechanism for the formation of nitrogen oxides is the thermal mechanism. The share of hydrogen in the fuel mixture contributed to the increase in the combustion temperature and thus intensified the NO_x formation process. The data in Fig. 7 show this almost linear relationship between NO_x emissions and the share of hydrogen in combustion. For the engine fueled with the reference fuel, the NO_x concentration was 457 ppm and for a 32% share of H_2 it increased to 860 ppm, i.e. increased almost twice.

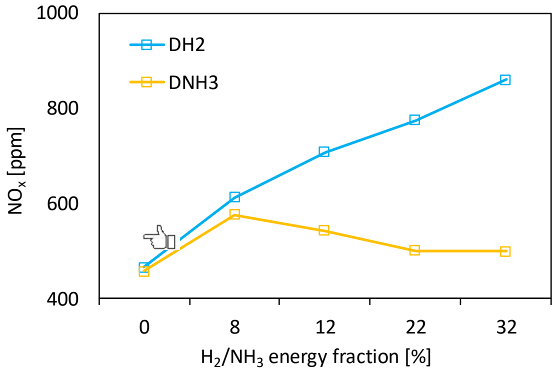

Fig. 7. NO_x concentration in exhaust gases depending on the fuel type

In the case of NH_3 engine fueling, NO_x emissions will also be affected by the thermal mechanism, but the fuel mechanism will play a significant role. Ammonia is a compound of nitrogen and hydrogen, which means that access to nitrogen in the fuel contributes to changing the nature of NO_x formation mechanisms. According to literature sources [13, 23], the deNO_x mechanism can also occur with such fuel. In the analyzed cases, for an 8% share of NH_3, the NO_x concentration increased by 120 ppm, and for a further increase in the share of NH_3, a systematic decrease in NO_x concentration was noted. One of the main benefits of using alternative fuels, such as hydrogen or ammonia, in a compression ignition engine is the reduction of carbon-based compound emissions. One of them is soot, which is a significant problem for conventional diesel engines. Figure 8 shows changes in the soot concentration in the exhaust gases of a dual-fuel engine in the analyzed range of H_2 and NH_3 shares. Emission results showed a significant reduction in soot content in the exhaust of the dual fuel engine, compared to a conventional engine fueled with diesel fuel alone. Replacing part of the diesel fuel with fuels free of elemental carbon, such as hydrogen or ammonia, in the engine cylinder reduced the possibility of incomplete fuel combustion and reduced the amount of soot in the exhaust. The diesel/hydrogen engine was characterized by lower soot emissions compared to the diesel/ammonia engine due to the better ability of H_2 to homogenize the combustible mixture and reduce fuel-rich zones in the engine cylinder.

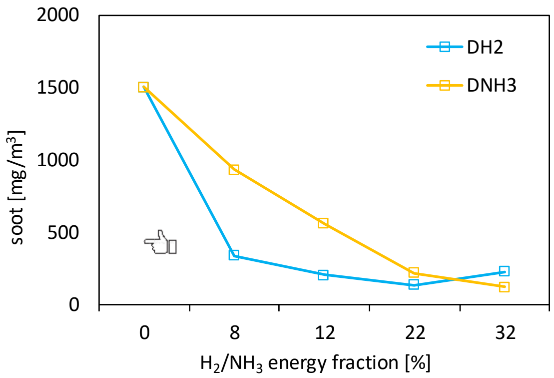

Fig. 8. The influence of H_2 and NH_3 on soot emission

4. Conclusions

As shown in this research, the addition of hydrogen or ammonia as an energy substitution for diesel fuel can be an interesting solution not only because of diesel engine performance but also the combustion process itself. On the basis of the research, the following conclusion can be drawn:

- The addition of hydrogen and ammonia to diesel fuel decreases the total energy of the cycle from 1122 kJ to 1022 kJ for diesel/hydrogen and from 1122 kJ to 1038 kJ for diesel/ammonia, to achieve the same IMEP as for diesel fuel.
- The addition of hydrogen to diesel causes an increase in the cycle's p_{max}, and that value constantly increases with the increase in hydrogen share by 0.76 MPa from 6.24 to 7.00 MPa. The ammonia addition did not cause a significant increase in p_{max}. The p_{max} was almost constant for all energy ratios, and it was around 6.24 MPa.
- The diesel/hydrogen combustion is characterized by two maxima on the HRR curve for all energy ratios except the 32% ratio, where there is only one maximum. The increase in HRR value is 15 J/deg for all cases, except for the 32% energy ratio, where HRR is 20 J/deg. The combustion of a diesel/ammonia mixture is characterized by only one maximum, indicating a combustion process closer to a kinetic one. In this case, the increase in HRR value doesn't exceed 15 J/deg for all energy ratios.
- The hydrogen addition did not affect an ignition delay (CA10), which is, for all cases, close to 17°CA, whereas the ammonia constantly increases that period from 17°CA to 19°CA, with an increase in the ammonia energetic share.
- The CA10–90 phase for hydrogen constantly decreases with an increase in energy share, and it was around

14°CA for an energy share equal to 22% from 54°CA to 40°CA. Because of an increase in unstable combustion for an energy share equal to 32% the increase in that parameter was noted to CA10–90 = 50°CA. For diesel/ammonia mixtures, there was a constant increase in the CA10–90 phase with an increase in the ammonia energy share from 54°CA to 58°CA.
- The addition of hydrogen or ammonia causes an increase in the indicated thermal efficiency, where for both combusted mixtures, the maximum value was observed for 12% energy share, and it is 38% and 34% respectively.
- The addition of ammonia did not cause any significant change in COV_{IMEP}, and the same trend was observed for all energy fractions of hydrogen, except for the last one, which was 32%. An increase in COV_{IMEP} for this energy fraction was observed. It is worth noting that the COV_{IMEP} for all tested mixtures was under 5% value.
- The emission of NO_x increases in both tested fuel mixtures. However, a more visible increase was observed for diesel/hydrogen mixtures, where the increase is from 457 ppm to 860 ppm. For diesel/ammonia mixtures, the increase in NO_x emissions was from 450 ppm to 500 ppm.
- In both cases of fuel mixtures, a decrease in soot emission was observed. In both mixtures, diesel/hydrogen and diesel/ammonia, the decrease was from a value level of 1500 ppm to 200 ppm.

The use of alternative fuel, which does not contain carbon particles in its structure, is an interesting solution to improving the combustion process and emissions of internal combustion diesel engines. Both fuels, when blended with diesel, improve ITE but also demonstrate that it is important to use their full potential, which requires controlling injection timing. The next step of the research will be the modification of the engine fuel supply system from mechanical to an electronically controlled one, which gives the possibility to change the injected fuel dose in a wider range and also provides control of the injection timing, which allows for the expansion of the field of research.

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Nomenclature

AEF	ammonia energy fraction
bTDC	before top dead center
CI	compression ignition
COV_{IMEP}	coefficient of variation for IMEP
HEF	hydrogen energy fraction
HRR	heat release rate
IC	internal combustion
IMEP	indicated mean effective pressure
ITE	indicated thermal efficiency
LFS	laminar flame speed
LHV	lower heating value
LPG	liquefied petroleum gas
SEC	specific energy consumption
σ_{IMEP}	standard deviation of IMEP

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