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An experimental investigation of the performance, emission and combustion stability of compression ignition engine powered by diesel and ammonia solution (NH_4OH)

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Abstract

This study presents experimental examinations of a stationary single-cylinder compression ignition dual fuel engine for the combustion of diesel fuel with water ammonia solution. The effect of 25% water ammonia solution on the combustion, performance, emissions and stability of the dual fuel compression ignition engine was investigated, taking into account its different operating conditions. The experiments were carried out for three modes of engine operation with three loads (35%, 60% and 100%) and a change in the water ammonia solution energy fraction at 60% load, within the range from 0% to 17%. Co-combustion of diesel fuel with water ammonia solution in the test engine contributed to an increase in the ignition delay period and combustion duration, and to an increase in the heat release rate. Compared to the combustion of diesel fuel alone, combustion involving ammonia causes deterioration in the stability of the test engine operation, yet not exceeding the permissible stability indices for reciprocating combustion engines. Addition of water ammonia solution led to reduced nitrogen oxide emissions and increasing carbon monoxide and hydrocarbon emissions and did not result in significant changes in carbon dioxide emissions.

Keywords

Diesel engine, dual fuel, ammonia solution, engine stability, ignition delay, combustion duration, emission

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Introduction

In the era of dwindling fossil fuel resources and the negative impact of the products of their combustion on the environment, alternative fuels provide the opportunities for further uninterrupted development of reciprocating engines and their adaptation to modern environmental protection requirements, both in transport and in the energy sector.[1,2] The explorations of new fuels for powering of combustion engines concern not only traction engines but also stationary engines that drive electric generators, which are often designed as co-generation systems that generate both electricity and heat.[3,4] Stationary engines used in the power industry are most often diesel engines operating in a diesel cycle and are powered by fuels obtained from crude oil processing, whose combustion contributes to air pollution, emissions of significant amounts of soot, carbon dioxide (CO_2) and nitrogen oxides (NO_x), thus contributing to the greenhouse effect.

One way to reduce the consumption of crude oil in internal combustion engines is to use ammonia (NH_3) as an alternative fuel.[5,6] Ammonia is a waste product of many technological processes (e.g. cooling processes), so its combustion in the engine may also be an effective way of its disposal. According to Ubowska,[7] the use of ammonia as a fuel not only eliminates the problem of CO_2 emissions but also the problem of soot. The main products of ammonia combustion in the air are nitrogen, steam and NO_x.[8] According to Grzesiak et al.,[9] ammonia could become one of the fuels of the future. Due to the fact that ammonia contains hydrogen and is

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a carbon-free energy source, it is a potential candidate for the replacement of conventional fuels used in engines, although the heating value (in other words, the energy density of ammonia), is lower than, for example, petrol or diesel fuel (DF).[10] However, using ammonia as a fuel in a combustion engine is not easy, as its combustion is incomplete, which affects the process efficiency and is a source of waste ammonia gas. Ammonia is difficult to use directly in internal combustion engines due to the insufficiently fast flame spread, but it is possible.[11,12]

In a compression ignition (CI) engine, ammonia may be co-combusted with conventional engine fuel (DF). Two supply systems are used in this solution, one for DF supplied to the engine cylinder and the other for alternative fuel, supplied most often to the intake manifold.[13,14] Using ammonia as fuel is not a new idea.[5] Research on its use as a direct source of fuel and as a source of hydrogen, which can be then used for fuel cell operation, has been carried out for more than 30 years.[15–17]

Despite the promising results of these studies, the risks associated with the use of ammonia must be taken into account. It is a toxic and flammable compound which, when mixed with air, forms an explosive mixture. However, it is much safer than other fuels. For example, the CI temperature of ammonia is 651 °C compared to 230 °C for diesel and 300 °C for petrol. In contrast to hydrogen, the ammonia leakage is accompanied by a strong characteristic smell that can be felt even at low concentrations. An unquestionable advantage of ammonia is the high octane number of 110–130, which, especially in spark ignition engines, may improve the combustion process and reduce undesirable effects such as knocking combustion.[18]

Nowadays, research into co-combustion processes of pure ammonia or its solutions with DF in CI engines attracts more and more attention worldwide. Reiter and Kong[19] conducted research on combustion and emissions of a dual fuel CI engine fuelled with ammonia and diesel. In their research, evaporated ammonia was supplied to the intake manifold, whereas diesel was injected into the engine cylinder to initiate combustion. The research was conducted, among others, at constant engine power and for various diesel-to-ammonia ratios. The most economical operation was ensured by the 40%–60% energy diesel fraction and 60%–40% ammonia energy fraction (AEF). Emissions of carbon monoxide (CO) and hydrocarbons (HC) in the exhaust gas using the dual fuel method were generally higher than those of pure DF. Below the 40% of ammonia, a reduction in NO_x emissions caused by lower combustion temperature was achieved. If ammonia constituted the majority of the fuel energy, NO_x emissions increased significantly due to the nitrogen in the fuel. The use of a significant amount of ammonia (ammonia energy more than 40%) resulted in a reduction in soot emissions due to the lack of carbon in the fuel structure. Analysis of the pressure variations in the engine cylinder showed that with the increase in the ammonia fraction, the ignition delay (ID) increased and the maximum combustion pressure decreased due to slower combustion. Research has shown that ammonia can be used as an alternative to fossil fuels in diesel engines.

Gill et al.[20] investigated the co-combustion of alternative fuels such as pure ammonia, dissociated ammonia (a mixture of H_2 and N_2) and pure hydrogen with diesel in the CI engine. The alternative fuel, which accounted for 3% of the total mass of air sucked in by the engine, was supplied to the intake manifold, whereas DF was injected into the cylinder to initiate combustion. The tests were conducted for two engine loads: 3 bar indicated mean effective pressure (IMEP) and 5 bar IMEP. Co-combustion of each alternative fuel with diesel resulted in a reduction in CO_2 emissions compared to diesel alone. Under high load conditions, the use of NH_3 alone has proven to be more advantageous in terms of stability and thermal efficiency of the engine compared to dissociated ammonia and hydrogen. At low loads, they behaved similarly.

Sahin et al.[6] conducted experimental studies on the effect of the fraction of ammonia solution (25% ammonia + 75% water) used for co-firing with DF on the performance and emissions in a small diesel engine. The 2%, 4%, 6%, 8% and 10% volumetric percentages of ammonia solution in the dose of the fuel supplied to the engine were analysed. The solution of ammonia in water was supplied through a carburettor to the intake air in the engine manifold. The results of the experiments showed that the addition of ammonia increases specific fuel consumption at all loads at 2200 and 3000 r/min. A reduction in specific fuel consumption was achieved at 2600 r/min, which is the speed designed for the engine. On one hand, the addition of ammonia solution to the intake manifold improved the effective efficiency of the engine for all operating conditions. On the other hand, it increased emissions and the concentrations of CO, HC and NO_x in the exhaust gas. Only CO_2 emissions were reduced with the increase in the ammonia fraction.

Another way of co-combustion of ammonia with another fuel is to burn a previously prepared blend of these fuels and deliver it with one injection system to the engine cylinder. The combustion and emission processes in a single-cylinder diesel engine using a blend of ammonia and dimethyl ether (DME) were investigated by Gross and Kong[21] and Ryu et al.[22] The tests were carried out under various operating conditions for DF only, DME only and three blends. The first contained 20% NH_3 and 80% DME, the second: 40% NH_3 and 60% DME and the third: 60% NH_3 and 40% DME. The results obtained for different blends of ammonia and DME show that ammonia causes a greater ID and reduces the engine load conditions due to its high compression temperature and low combustion rate. The research demonstrated that engine performance decreases with increasing ammonia concentration in the fuel mixture. The presence of ammonia in the fuel blend with DME reduces the pressure and combustion temperature, resulting in higher CO and HC emissions. As a result of the increased content of nitrogen in the fuel, NO_x emissions also increase. The main benefit of using ammonia was the reduction of soot in all investigated cases. Furthermore, research results have shown that an increase in injection pressure by 30 bar can allow for using blends with higher ammonia content and lead to improved combustion and improved emissions compared to the combustion of DME alone.[21]

In the study, Boretti[23] conducted numerical research on combustion in a dual fuel turbocharged diesel engine fuelled with diesel and ammonia. In the engine, both diesel and ammonia were injected directly into the cylinder. The proposed solution, compared to diesel engines with the injection of the DF into the cylinders and NH_3 injection into the manifold, resulted in higher power density and improved efficiency, similar to those achieved when burning DF alone. The simulations showed that using direct injection of both fuels, precise control of engine load is possible.

Another example of numerical modelling of the process of co-combustion of ammonia with other fuels in a reciprocating engine is the research carried out by Tay et al.[24] Simulations were performed to demonstrate the effect of the use of ammonia as the main fuel on the performance and emissions for a dual fuel CI engine. Initiation of the combustion process in the engine cylinder occurred by injection of the pilot dose of three fuels such as the diesel itself, a mixture of kerosene and diesel or kerosene alone. The research has shown that the use of ammonia to fuel the CI engine reduces the emissions of CO and CO_2 due to the lack of carbon, which, unlike ammonia, occurs in conventional fuel. Nitric oxide emissions are reduced with a small fraction of NH_3, but increases with its significant fractions of above 60%. Analysis of the effect of pilot dose injection timing on the combustion process showed that with the right injection acceleration, it is possible to achieve total combustion of the ammonia-based blend, despite the low combustion speed of this fuel. The simulations also provided information on the temporal and spatial course of the combustion process and spatial concentration of exhaust gas components in the cylinder of a dual fuel engine.

This study presents experimental examinations of a stationary single-cylinder CI dual fuel engine for co-combustion of DF with water ammonia solution (WAS), also termed ammonia water. The effect of ammonia water with 25% concentration on the combustion, performance, emissions and stability of the dual fuel CI engine was examined, taking into account its different operating conditions. The experiments were conducted for three modes of engine operation, covering three different loads (35%, 60% and 100%) and taking into account, at 60% load, the change in the energy fraction of WAS co-burnt with DF ranging from 0% to 17%. The analysis of engine operation stability was also carried based on the variation of the maximum combustion pressure in 200 consecutive cycles of its operation.[25]

Experimental methodology

Research engine and apparatus

The study was conducted on a single-cylinder four-stroke naturally aspirated Andoria Type S320 diesel engine with a horizontal cylinder system and two valves in the head. In order to carry out tests of this engine on a dynamometer with an asynchronous electric machine, modifications were made into its design. The evaporative engine cooling system was replaced by a closed water cooling system. The direct mechanical fuel injection system was replaced by an electronically controlled injection system with an electromagnetic common rail (CR) injector. Control of the CR injector allows for a temporal change in the dose, injection timing and injection pressure.

Furthermore, an electromagnetic low-pressure liquid fuel injector is mounted in the engine intake system, also with independent control of time, pressure, dose and injection timing. Co-combustion of DF with ammonia water was performed using a CR injector to inject DF directly into the engine combustion chamber and a low-pressure injector to inject ammonia water into the suction manifold. The test stand was also equipped with elements for the measurement of cooling water temperature, suction air temperature, fuel temperature and pressure and parameters of electric current of asynchronous machine loading the combustion engine. Table 1 presents the basic technical data for this engine.

Figure 1 shows a diagram of the test stand that allows dosing control of both fuels and measurements:

- Consumption of DF delivered to the cylinder;
- Consumption of WAS injected into the inlet channel;
- The concentrations of the toxic exhaust gas components;
- Rotational speed;

Table 1. Engine main specifications.

Parameter	
Type of engine	Four-stroke, CI
Number of cylinders	1
Displacement volume	1810 cm^3
Engine rotational speed	1000 r/min
Bore × stroke	120 × 160 mm
Compression ratio	17:1
Injection timing	30 bTDC
Injection pressure CR max	150 MPa
Rated power	10 kW
Engine torque rated power	84.4 N m

CR: common rail; bTDC: before top dead centre; CI: compression ignition.

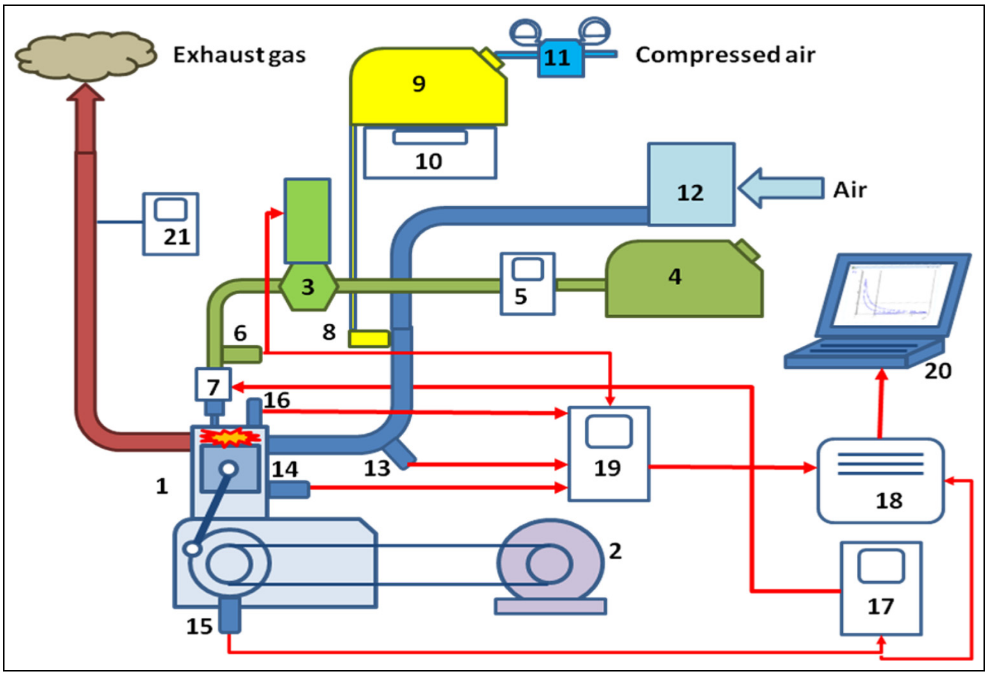

Figure 1. Schematic diagram of the experimental setup.
1 – CI engine; 2 – electric generator; 3 – DF pump with control system; .4 – DF tank; 5 – DF flow metre; 6 – DF pressure sensor; 7 – DF common rail injector; 8 – ammonia fuel electric injector; 9 – ammonia fuel tank; 10 – measuring weight; 11 – air pressure regulator; 12 – air filter; 13 – inlet air temperature sensor; 14 – engine temperature sensor; 15 – CA encoder sensor; 16 – in-cylinder pressure sensor; 17 – fuel injection controller; 18 – data acquisition converter; 19 – signal amplifier; 20 – data recorder PC-SAWIR; 21 – emission analyser.

- Indicated power of the engine and recording and analysing the in-cylinder pressure data.

A measuring system for engine indication consisting of

- Piezo-ceramic pressure transducer, Kistler 6061;
- Charge amplifier, Kistler 5011;
- Crankshaft rotation angle transmitter with 360 pulses/rev resolution;
- 16 bit digital registration module Measurement Computing USB-1608HS;
- PC-SAWIR software.

The concentrations of the exhaust gas components were measured with the Bosch BEA 350 analyser (CO, CO_2 and total hydrocarbons (THC) by nondispersive infrared (NDIR), oxygen (O_2) electrochemical cell).

Methodology

The combustion process, engine performance and emissions and stability of the dual fuel engine were examined. The energy fraction of WAS in the fuel is abbreviated as AEF. In each case, the measurements were made after the thermal state of the engine stabilized at a constant speed of 970 r/min. The scope of the tests is presented in Table 2. The initial examinations (Test 0) consisted in testing an engine fuelled only with diesel as a reference fuel at three different loads of 35%, 60% and 100%, corresponding to three doses of DF (1.58, 2.28 and 3.68 kJ/cycle). The injection angle (30° before top dead centre (bTDC)) used in all tests resulted from preliminary tests aimed at choosing its optimal value due to the efficiency of the engine with constant participation of AEF. These results were the reference point for the results of subsequent studies.

The main research consisted of two tests. The purpose of Test 1 was to verify the effect of supplying a constant highest possible dose of WAS to the cylinder (380 J/cycle) on engine performance and emissions at three loads (35%, 60% and 100%) corresponding to three doses of DF in Test 0. The maximum dose of ammonia solution resulted from its lower heating value (LHV_{WAS}) and limited efficiency of the WAS injector. It was not possible to inject more WAS during the engine cycle to get a higher WAS energy share at a dose of 1.9 kJ/cycle in DF. The water AEF in the dose of fuel supplied to the engine for individual loads was 24.6%, 17.0% and 10.5%, respectively. Test 2 was used to evaluate the effect of AEF on engine performance and emissions. AEF varied from 0% to 17%, which corresponded to a change in the dose of WAS from 0 to 380 J/cycle.

The AEF in the supplied fuel for the measurements was calculated as a ratio according to the following equation

$$AEF = \frac{Q_{WAS}}{Q_D + Q_{WAS}} \quad (1)$$

where Q_D is the heat in the fuel supplied to the cylinder in kJ/cycle

Table 2. The scope of the tests.

Test	Load (%)	Total fuel (kJ/cycle)	Diesel (kJ/cycle)	WAS (J/cycle)	AEF (%)	Excess air ratio (λ)	Diesel SOI (degree bTDC)
0	35	1.58	1.58	0	0	4.19	30
	60	2.28	2.28			2.91	
	100	3.68	3.68			1.80	
1	35	1.58	1.2	380	24.6	3.79	30
	60	2.28	1.9		17.0	2.68	
	100	3.68	3.3		10.5	1.70	
2	60	1.9	1.9	0	0	3.49	30
		2.05		150	7.5	3.10	
		2.11		210	10	2.98	
		2.22		320	14.5	2.78	
		2.28		380	17	2.68	

WAS: water ammonia solution; AEF: ammonia energy fraction; SOI: start of injection; bTDC: before top dead centre.

$$Q_D = m_D \cdot LHV_D \quad (2)$$

where Q_{WAS} is the heat in the fuel supplied to the intake manifold

$$Q_{WAS} = m_{WAS} \cdot LHV_{WAS} \quad (3)$$

Each measurement was performed at a constant AEF. The dose of DF was determined by the set injection time in the conditions of constant pressure in the CR system. The same method was used to set the WAS dose to be injected under constant pressure into the engine inlet duct. The injection time was set independently of the diesel injection time. DF consumption was measured using a volumetric method based on a consumption time of 41 cm^3. However, the actual dose of WAS was determined by measuring the change in its mass using electronic scales over the same period of time.

During each measurement, the course of pressure in the engine cylinder was recorded during 200 consecutive cycles of its operation with the sampling step of 1 °CA (crank angle). The real-time pressure recording system analyses the pressure and presents the results. The system also determines the instantaneous engine speed by measuring the duration of each pulse generated by the CA marker. After recording each engine cycle, its analysis is performed and the current values are presented

- Maximum pressure and the angle at which it occurs;
- Maximum pressure rise rate and the angle at which it occurs;
- Rotational speed;
- Unit indicated work;
- Indicated power of the cylinder.

At the same time, the current graphs are drawn

- Cylinder pressure vs. CA;
- Cylinder pressure vs. volume function;
- Pressure or temperature derivative vs. CA;
- Heat release or mass fraction burned (MFB) vs. CA;
- Changes in rotational speed, indicated work and indicated power in successive cycles.

During the recording, the changes in speed, indicated work and indicated power in subsequent cycles are also presented. The system also records the mean temperature values for each engine cycle for inlet air, coolant, exhaust gas and fuel. The exhaust gas analysers recorded changes in concentration of components in the engine exhaust gas, such as NO_x, HC, CO and CO_2. Thermal stability was maintained in each measurement point by controlling exhaust gas temperature and exhaust gas composition.

Test fuels

Fuels used in this study included DF and 25% WAS. The DF was a commercial fuel sold by the Polish ORLEN Refinery and used commonly to fuel diesel engines in cars. The fuel is a mixture of liquid HC obtained through crude oil distillation. In the case of the DF, the most important parameter is cetane number that determines the susceptibility of the fuel to CI. Polish standards define the minimal cetane number for DFs that guarantees proper engine operation at the level of 51.[26]

The research was limited to checking the possibility of co-firing in the diesel engine, of readily available 25% WAS. This is due to the limited solubility of ammonia in water, which is 33% at 20 °C and only 28% at 30 °C. The 25% WAS used is a commercial, ready to use chemical compound for general use.

WAS is a solution of ammonia and water, with a large proportion of water, which is also not negligible for the combustion process. Water evaporates in the engine cylinder and the resulting water vapour affects not only the combustion process itself but also exhaust emissions. Due to indirect injection, the so-called

Table 3. The physicochemical properties of fuels used in this study.

	Unit	Diesel	WAS (25%)
Chemical formula		$C_{14}H_{30}$	NH_4OH
Molecular weight	g	198.4	17.04
Cetane number	–	51	–
Research octane number	–	15–25	–
Boiling point	K	453–643	310
Liquid density	kg/m^3	830	910
Lower heating value	MJ/kg	42.5	2.69
Heat of evaporation	kJ/kg	243	–
Autoignition temperature	K	503	927
Stoichiometric air–fuel ratio	–	14.6	–
Viscosity (at 25 °C)	mPa s	2.419	0.475

WAS: water ammonia solution.

crevice effect will have a significant impact on emissions, especially HC and CO_2.

WAS with chemical formula NH_4OH has been widely used in the chemical, pharmaceutical, rubber and textile industries for detergent production, flue gas desulphurization and as a laboratory reagent. WAS is produced by dissolving pure ammonia in demineralized water

$$NH_3(gas) + H_2O(liquid) \rightarrow NH_4OH(liquid)$$

The NH_4OH liquid has a characteristic sharp ammoniacal smell. At 15 °C, the 12.74% solution has a density of 0.950 g/cm^3, whereas the 35.20% solution has a density of 0.880 g/cm^3. Heating of ammonia solution causes the release of ammonia. Ammonia water is mixed in any proportion with water, alcohol, acetone and chloroform. Under normal conditions, this compound is chemically stable, but as the temperature rises, the amount of gaseous ammonia released from the solution (ammonia desorption) increases, posing a risk of fire and/or explosion. Gaseous ammonia is thermodynamically stable. WAS reacts rapidly in contact with acids, strong oxidizers, halides, acrylic acid, dimethyl sulphate, silver nitrate, silver oxide, hypochlorites, mercury, CO_2, nitrogen trichloride and ethylene oxide. The physicochemical properties of DF and ammonia water are shown in Table 3.[27–29]

In all tests, the test results obtained for the engine powered by pure DF were used as a reference. Its lower heating value is $LHV_D = 42.5$ MJ/kg and the density needed to calculate the mass fuel consumption based on time-measured fixed volume consumption is $\rho_D = 830$ kg/m^3. For 25% WAS, the LHV_{WAS} was calculated using the Mendeleev formula[30]

$$LHV_{WAS} = 339.15 \cdot C_r + 1030 \cdot H_r - 108.9 \cdot (O_r - S_r) - 25.1 \cdot W_r \tag{4}$$

where C_r, H_r, O_r and S_r are the weight contents of carbon, hydrogen, oxygen and sulphur and W_r is the water in fuel (%) in raw conditions.

For a 25% ammonia solution of which the molecular weight of NH_3 = 17.04 g/mol, the weight fractions of the above-mentioned components are C_r = 0%, H_r = 3.03/17.04·25% = 4.45%, O_r = 0%, S_r = 0% and W_r = 75%. Hence, the heating value of 25% WAS is LHV_{WAS} = 2696 kJ/kg.

WAS consumption per cycle, m_{WAS}

$$m_{WAS} = \frac{120 \cdot \Delta M_{WAS}}{n \cdot \Delta t} \tag{5}$$

where ΔM_{WAS} is the mass consumption of ammonia solution, Δt is the time of measurement of fuel consumption and n is the engine speed.

DF consumption per cycle, m_D

$$m_D = \frac{120 \cdot V_D \cdot \rho_D}{n \cdot \Delta t} \tag{6}$$

where V_D is the volume of DF supplied to the engine cylinder and ρ_D is the density of DF.

Operating parameters

For qualitative and quantitative analyses of the thermal-chemical and flow phenomena occurring in the engine, the recorded pressure changes in the cylinder were used. They were used to determine the changes in several very important engine operating parameters such as IMEP and indicated thermal efficiency (ITE) as well heat release rate (HRR) or total heat released. A detailed description of the research procedure and measurement results processing are included in our previous paper.[31]

The HRR is one of the indicators characterizing the combustion process in an internal combustion engine cylinder. HRR can be determined on the basis of registered changes in-cylinder pressure, from the cylinder pressure graph, by calculating the changes in internal energy and the indicated work factor, as follows

$$HRR = \frac{1}{\chi - 1}\left[\chi p \frac{dV}{d\phi} + V\frac{dp}{d\phi}\right] \tag{7}$$

where χ is the ratio of specific heats, V is the cylinder volume and p is the cylinder pressure.

The nature of the work of a diesel engine is significantly influenced by the value of the pressure rise rate. Rate of pressure rise $dp/d\varphi$ was determined

$$\frac{dp}{d\phi} = \frac{p_i - p_{i-1}}{\phi_i - \phi_{i-1}} \tag{8}$$

where p is the cylinder pressure and φ is the crank angle.

Based on the combustion pressure charts, characteristic values representative of many individual engine cycles, such as average maximum combustion pressure, can be determined. Changes in these quantities can be calculated using statistical analysis methods and presented as a coefficient of variation for the maximum pressure (COV_{pmax}).

The average value of maximum pressure, p_{max}, determined on the set of pressure

$$p_{max} = \frac{1}{N}\sum_{i=1}^{N} p_{max\,i} \tag{9}$$

where N is the cycle index and $p_{max\ i}$ is the maximum pressure in individual cycles.

The indicated pressure is evaluated based on the recorded changes in the cylinder pressure and represents one of the indices that characterize operation of combustion engines in terms of the opportunities to ensure high and expected functional performance.

IMEP for a single engine cycle

$$IMEP_i = \frac{1}{V_d}\int_{0}^{720} pdV \tag{10}$$

where V_d is the displaced cylinder volume.

The average value of IMEP

$$IMEP = \frac{1}{N}\sum_{i=1}^{N} IMEP_i \tag{11}$$

where $IMEP_i$ is the indicated mean effective pressure in individual cycles.

One of the most commonly used criteria for assessing the correct operation of an internal combustion engine is its cycle-by-cycle variation. As a measure of the cycle-by-cycle variation of the engine, the coefficient of variation for the maximum pressure COV_{pmax} can be taken, expressed in percentage and calculated as the ratio of the maximum standard pressure deviation to its average value over many recorded engine cycles.

Coefficient of variation of maximum pressure (COV_{pmax}) is defined as[32]

$$COV_{p\,max} = \frac{STD_{p\,max}}{p_{max}}\,100\% \tag{12}$$

The standard deviation of maximum pressure is defined as

$$STD_{p\,max} = \sqrt{\frac{1}{N}\sum_{i=1}^{N}(p_{max\,i} - p_{max})^2} \tag{13}$$

where N is the cycle index and p_{max} is the mean value of maximum pressure of N cycle $p_{max\ i}$.

The profiles of standardized heat release were used to analyse the combustion stages. The combustion process was divided into conventional stages, such as the first one being ID and the second one being combustion duration (CD). For reciprocating engines, these time instants are defined in CA degrees, which allows for the comparison of the results for different engines. The ID is affected by both physical and chemical processes. The first depends primarily on the quality of fuel atomization, the way it is supplied to the combustion chamber and the gasodynamic processes in the engine cylinder. These factors cause the physical ID. The chemical processes are depended on the fuel quality, C/H ratio and O_2 content in the molecular structure, its heat of vaporization, ignition temperature, laminar flame speed (LFS) and LHV. Both phenomena, that is, physical and chemical delays, occur simultaneously. In practice, ID in a piston engine is mostly considered as a period expressed in CA degrees, from the beginning of DF injection to the time when 10% of heat is generated. Another stage is the combustion period from the release of 10%–90% heat.

Uncertainty analysis

In practice, experimental measurement of physical properties is typically connected with a measurement error and measurement uncertainty. Engine indication is an operation that depends on the accuracy of measurements of many quantities to a varying degree, resulting in a final uncertainty of the results. Therefore, the determination of the values that characterize engine work based on the indicator diagram must include the estimation of their measurement uncertainty. The measurement uncertainties of all directly measured quantities result from the accuracy of the instruments and measurement methods used. They also influence the accuracy of determination of the other analysed quantities. A summary of the estimated uncertainties (level of confidence of 99.7%) of the parameters measured is presented in Table 4.

Results and discussion

The basis for the assessment of the combustion process in a reciprocating engine is the analysis of indicator data. The pressure profiles for the engine cylinder and heat release were analysed. The main stages of combustion were identified based on the analysis of standardized heat release. The ID was defined as the time from the beginning of diesel injection to the time of 10% heat release. CD was also determined as the period between

Table 4. Absolute error and uncertainty of measured parameters.

Measured parameters	Absolute error	Uncertainty (%)
Pressure in cylinder	0.05 MPa	1
Rate of flow of diesel	0.03 kg/h	3
Rate of flow of ammonia solution	0.06 kg/h	2
Engine speed	0.1 r/min	0.01
IMEP	0.02 MPa	4
ITE	1.6%	5
NO_x concentration	25 ppm	4
THC concentration	12 ppm	5
CO_2 concentration	0.4%	5
CO concentration	0.06%	5

IMEP: indicated mean effective pressure; ITE: indicated thermal efficiency; THC: total hydrocarbons.

the release of 10%– 90% of heat.[31] The ID has a significant effect on the subsequent stages of combustion. The increase in this stage contributes to the increase in pressure and causes higher mechanical loads on the crankshaft. This leads to the so-called hard engine operation. For this reason, it is worthwhile to analyse the profiles of the rate of pressure rise during combustion. The stage called CD has a decisive impact on engine efficiency. The long combustion stage leads to the increase in heat losses to the engine combustion chamber walls, thus reducing engine efficiency. For the characterization of reciprocating engines, these stages are normally given as CAs.

Combustion characteristics

As indicated in the literature, ammonia can be used to fuel both spark ignition and CI engines.[33,34] The use of ammonia can be a good solution to an unfavourable phenomenon of engine knocking. The risk can be its properties that accelerate corrosion of copper or bronze elements. Due to low combustion speed and high resistance to self-ignition of ammonia fuels, these fuels predisposes for co-combustion with other fuels, for example, in a dual fuel engine. The authors carried out a study of co-combustion of ammonia in the form of an aqueous solution with diesel.

In the first stage, the combustion process was analysed for the engine powered with DF. The tests were carried out for constant rotational speed and constant fuel injection timing. Next, tests were carried out at enrichment of the WAS mixture with its participation of 380 J/cycle. In both analysed cases, for pure diesel and diesel and WAS, the energy dose delivered to the combustion chamber of the engine was almost the same as 1.58, 2.28 and 3.68 kJ, respectively. The data obtained for the diesel engine powered only by DF were considered a reference for the dual fuel engine.

Figure 2 shows the pressure and pressure rise profiles for the three analysed loads obtained for the engine fuelled with DF only and with a constant WAS fraction of 380 J/cycle. These graphs presents the significant effect of WAS on the pressure profiles. At the initial part of the pressure profile, the effect of WAS on the pressure rise can be observed before ignition. Injected into the manifold, WAS changes the properties of the air/WAS mixture. Another difference can be observed after the injection of DF. In the atmosphere of air/WAS, ignition occurs much later. This analysis will be presented in detail based on the heat release analysis. Comparison of the profiles of the rate of pressure rise $dp/d\phi$ reveals that the combustion process involving ammonia is characterized by a bit higher pressure rise values, making engine operation harder.

The analysis of heat release profiles shows that the ammonia fraction has a significant effect on the HRR, especially at the kinetic combustion phase. The HRR peak value for the maximum load is reached 4° later than for a conventional mode and was by 12 J/deg higher for the engine fuelled with the addition of WAS. This is the case for all analysed loads. No significant effect of ammonia on the diffusion phase of combustion was observed.

Figure 3(a) presents the results of the division of the combustion process into conventional stages with the WAS fraction of 380 J/cycle. It was found that combustion with WAS led to the increase in ID. The same difference in ID was found for the all loads and it was 5 °CA. Therefore, it was at a similar level for virtually the whole range. In the experiment, the 50 °CA angle was changing when the load or type of fuel was changed due to constant angle of beginning of DF injection. In case of powering of engine by DF, with the decrease in load, the fuel dose is decreasing as well which causes atomization and faster evaporation and this affects the intensification of the kinetic combustion phase. As a consequence, change in the 50 °CA angle is noticed.

The analysis of the conventional CD revealed that the WAS fraction contributes to its increase. As the load increases, the difference becomes smaller and smaller. For a 35% load, the difference in ID was 8 ° CA, and for a full load, this was only 1 ° CA. It was found based on the analysis that with the increased load, WAS fraction can increase. Figure 3(b) shows the efficiency (ITE) and IMEP. For the tested engine powered with WAS fraction of 380 J/cycle, maximum efficiency of 37.2% was achieved for 60% of the load. It was interesting that a higher IMEP value was obtained for all loads with WAS. This could be due to a change in the length of combustion phases, ID and CD.

In the next stage of the research, the effect of the increasing WAS fraction on co-combustion with DF was analysed for 60% load. In terms of the indicated engine efficiency, the most advantageous was the operation at a load of 60%, corresponding to the dose of DF of 1.9 kJ/cycle. The dose of DF was constant at 1.9 kJ/cycle, whereas the dose of WAS increased from 0 to 380 J/cycle (Table 1). The tests were conducted as before at a constant rotational speed of the engine crankshaft and at constant ignition timing for DF injection of 30° bTDC.

Figure 4 presents the comparison of the pressure profiles in the combustion chamber of the engine for co-combustion of diesel and various AEF. The effect of the increasing WAS fraction on the combustion process was analysed. Up to 10% of the AEF, the combustion process was characterized only by the increasing ID, whereas no effect on the increase in the value of either the maximum pressure or the increase in the rate of pressure rise was observed. A similar pattern can be observed for heat release, where standardized heat release profiles clearly show the effect of WAS on ID.

A slight increase in the maximum combustion pressure was found after exceeding 10% of the AEF. The maximum value of combustion pressure was obtained for AEF of 14.5% and amounted to 6.64 MPa. For the engine fuelled with diesel only, the maximum pressure

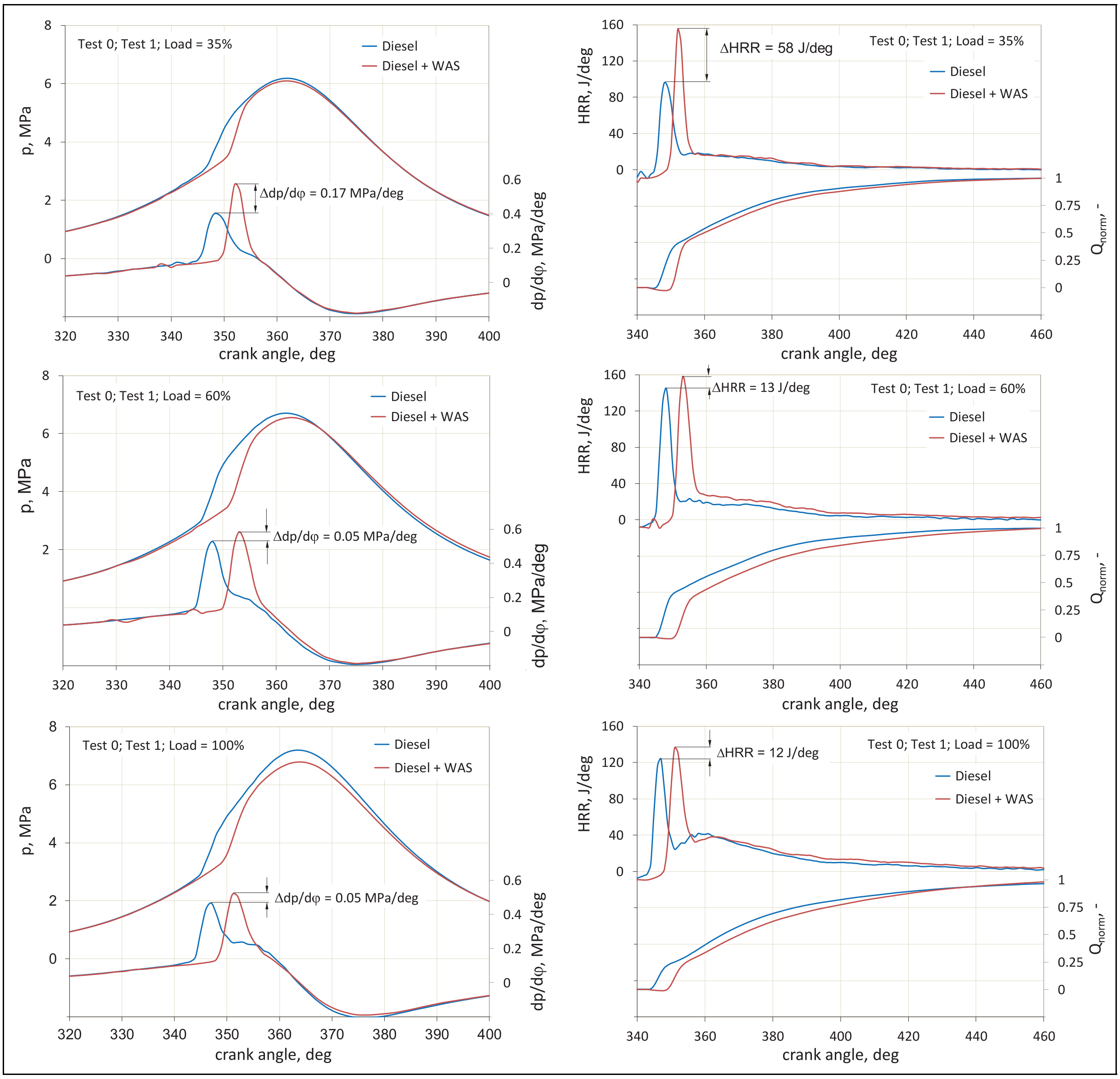

Figure 2. The pressure and pressure rise profiles for the three analysed loads obtained for the engine fuelled with diesel fuel only and with a constant WAS fraction of 380 J/cycle.

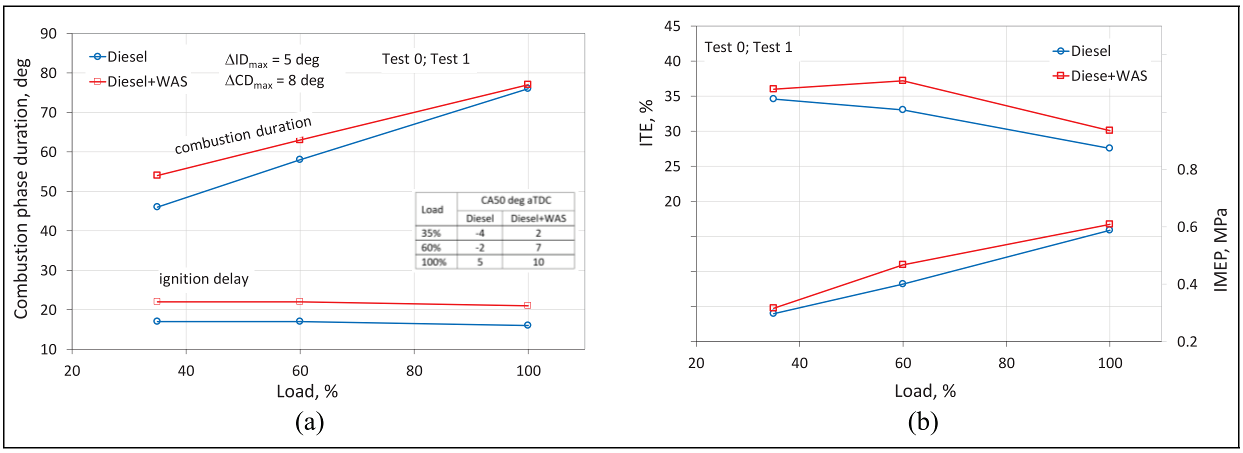

Figure 3. The stages of (a) the combustion and (b) the efficiency and indicated mean effective pressure for the diesel fuel and the dual fuel engine with the WAS fraction of 380 J/cycle.

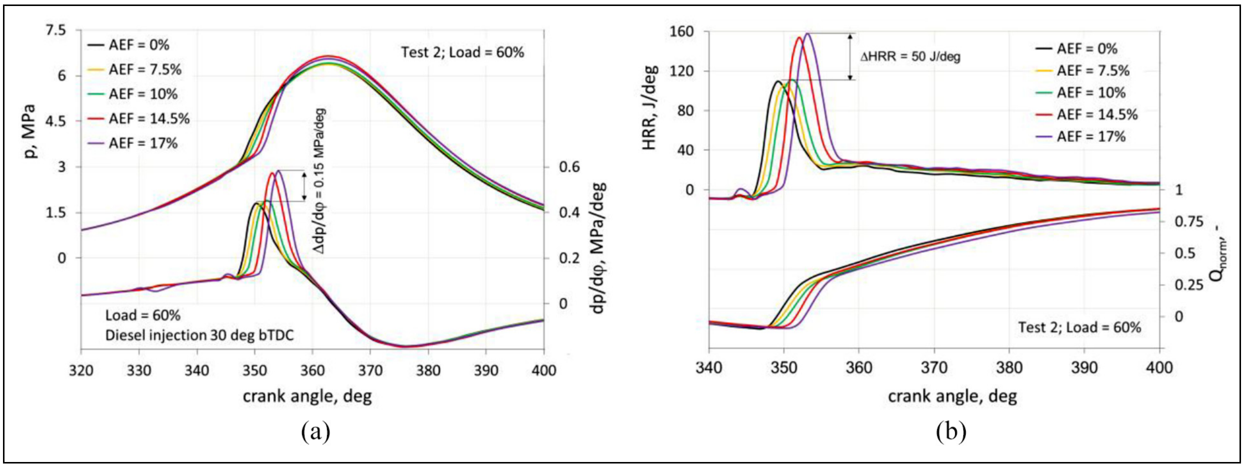

Figure 4. The pressure profiles (a) for the engine cylinder and heat release and (b) for increasing WAS fraction at 60% load on the engine.

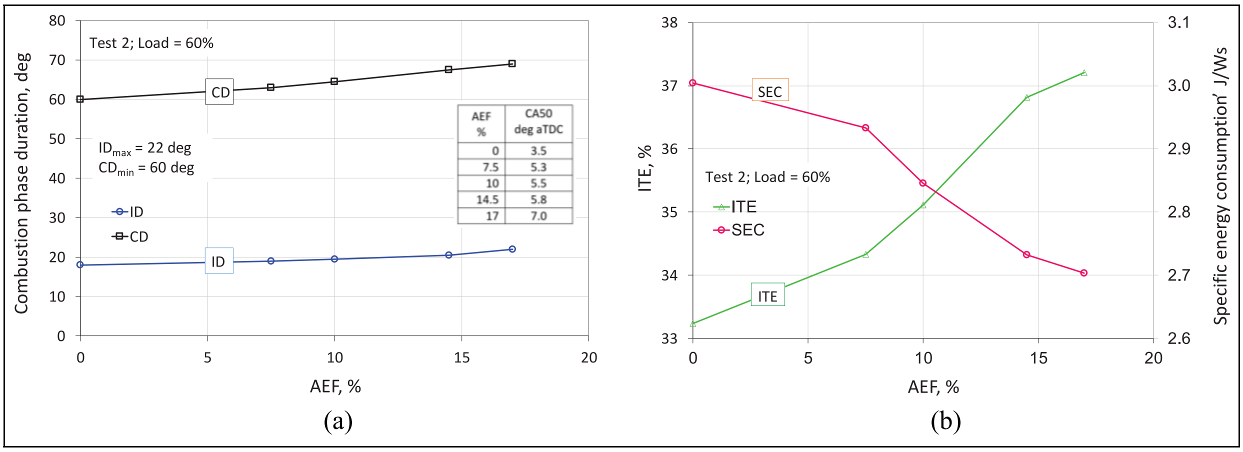

Figure 5. The effect of WAS fraction on the stages of (a) the combustion and (b) the efficiency and specific energy consumption at 60% load on the engine.

was 6.4 MPa, thus the increase in pressure was relatively small.

The increase in the AEF to 17% already caused a small increase in the maximum pressure value, which was due to the low WAS combustion rate. The maximum increase in the rate of pressure rise was 0.15 MPa/deg for 17% of AEF. The analysis of heat release shows that the highest HRR occurred also for 17% of AEF and compared to the conventional engine, it amounted to 50 J/deg. Standardized heat release processes were used to determine the combustion stages of ID and CD.

Figure 5 presents the effect of the fraction of WAS used for co-combustion with diesel in a CI engine. It can be clearly stated here that the increase in the WAS fraction was followed by an increase in ID. For the maximum WAS fraction (AEF = 17%), it increased by 4 °CA compared to that obtained for the engine fuelled only with diesel.

An increase in CD was found for the entire range of WAS energy fraction during combustion with DF. The maximum CD value was obtained for the WAS energy fraction of 17%, which was 15% higher than that obtained for an engine fuelled only by diesel. It was found that the value of engine efficiency increased with the increase in WAS energy fraction. For the maximum WAS energy fraction, the efficiency was 37.2%, that is, higher by nearly 4% compared to that of the engine fuelled only with diesel. The specific energy consumption (SEC) decreased with the increase in energy WAS fraction. For the engine fuelled with diesel, it was 3 J/W s and for a 17% fraction, it fell to 2.7 J/W s, which means that the engine energy demand was reduced by 10%.

Engine stability

The stability of an internal combustion engine operation related to the non-repeatability of the subsequent operating cycles and ignition misfiring can be assessed based on variation of IMEP[35] and also on the basis on variation of the maximal combustion pressure – COV_{pmax}.[36–38] Figure 6 presents the coefficients of

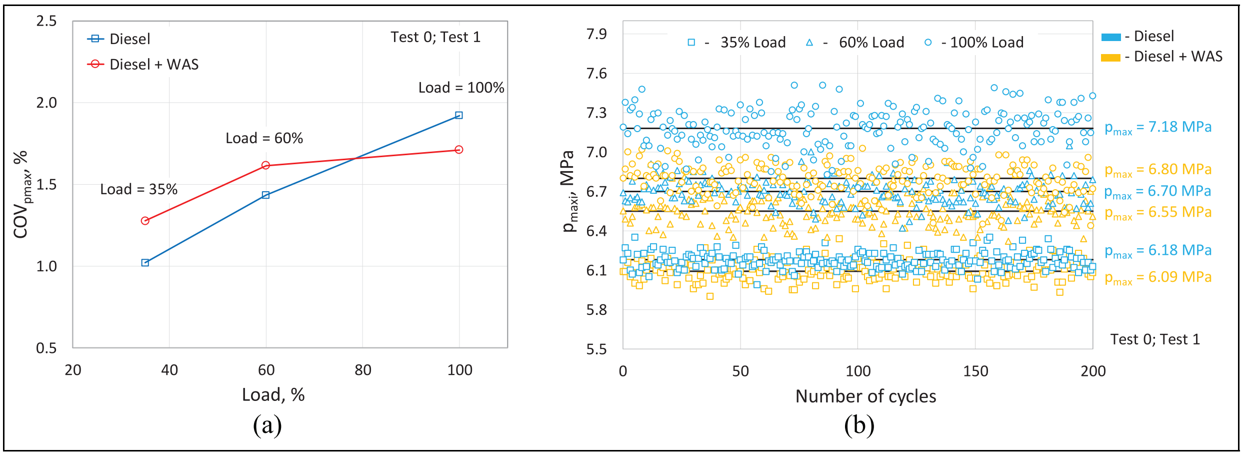

Figure 6. (a) The coefficients of variation of the maximal combustion pressure and (b) the dispersion of the maximum pressure value in individual engine operating cycles around the mean values of p_{max} determined for the analysed engine loads for the engine fuelled only with diesel and diesel fuel with WAS.

variation of the maximal combustion pressure determined for the analysed engine loads (35%, 60% and 100%) for the engine fuelled only with diesel (reference fuel) and DF with WAS (Figure 6(a)) and the dispersion of the maximum pressure value in individual engine operating cycles around the mean values of p_{max}, determined for 200 engine operating cycles (Figure 6(b)). The results correspond to the tests carried out in Test 0 and Test 1, described in Table 1.

For the dual fuel engine, the energy dose of WAS was constant and amounted to 380 J/cycle. With the COV_{pmax} coefficient adopted as an indicator of dual fuel engine operation stability, it can be seen that as the load increases, both the engine fuelled only by diesel and the engine for co-combustion of diesel with WAS were characterized by an increase in COV_{pmax}, indicating a deterioration in engine operation stability. The increase in the load from 35% to 100% resulted in an increase in COV_{pmax} from 1.02% to 1.92% for the mono-fuel engine and from 1.28% to 1.71% for the dual fuel engine.

The type of fuel used also had an effect on the engine operation stability. Under these conditions, for low (35%) and medium (60%) loads, the values of the p_{max} coefficient of variation were higher for the dual fuel engine, indicating worse stability of its operation. When working with maximum load, the effect of WAS fraction on the repeatability of subsequent operating cycles was reverse. Dual fuel engine worked more stable.

Figure 7 illustrates the coefficient of variation of maximum pressure (Figure 7(a)) and the dispersion of the maximum pressure value in individual engine operating cycles around the mean values (Figure 7(b)) determined for the analysed energy fractions of WAS at 60% load on the engine. The results correspond to the tests conducted in Test 2 and Test 1, described in Table 2.

Figure 7(a) shows the effect of the WAS energy fraction on the repeatability of the maximum combustion pressure in the cylinder of the test engine. It can be seen that co-combustion of DF with WAS slightly deteriorates the engine operation stability measured by the value of the coefficient of variation of maximum pressure. At AEF = 14.5%, the highest COV_{pmax} value of 1.64% was obtained. For the engine powered only by DF, this ratio was lower and amounted to 1.19%.

It can be concluded that the test engine worked correctly under all analysed conditions and did not exceed the permissible stability indices for internal combustion reciprocating engines. According to Nicolici et al.,[39] the acceptable COV_{pmax} limit for internal combustion engines is 10% and is equal to the limit that is acceptable until recently for COV_{IMEP}.[40] Currently, the COV_{IMEP} limit for internal combustion engines defined by Heywood is 5%.[35]

Exhaust emissions

An important part of the study was the assessment of the effect of the energy fraction of WAS burned with diesel oil on the concentration of toxic components in engine exhaust gases: NO_x, HC, CO and CO_2. Individual components of engine exhaust gas depend mainly on combustion process temperature and composition of the air–fuel mixture.

Figure 8(a) shows the maximum combustion temperatures and the results of the measurements of NO_x for three analysed loads obtained for the engine powered only by DF and with a constant WAS fraction of 380 J/cycle. As the load increased, the maximum temperature in the engine's combustion chamber increased. It can also be seen (Figure 10(a)) that WAS contributes to lowering the maximum temperature in the engine over all loads. For the maximum load, a decrease in the maximum combustion temperature by 94 K was noted. The effect of the WAS burned with DF in the reduction of NO_x emissions is noticeable. The addition of WAS reduces NO_x emissions by 40% (from 880 to 530 ppm) at 35% of engine load and up to 54% (from

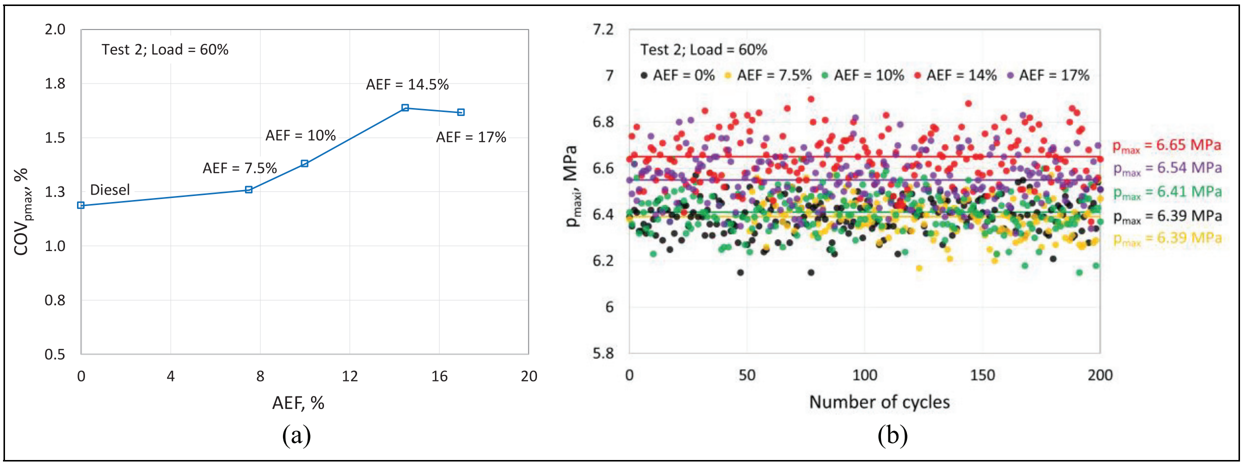

Figure 7. (a) The coefficient of variation of maximum pressure and (b) the dispersion of the maximum pressure value in individual engine operating cycles around the mean values determined for the analysed energy fractions of WAS at 60% load on the engine.

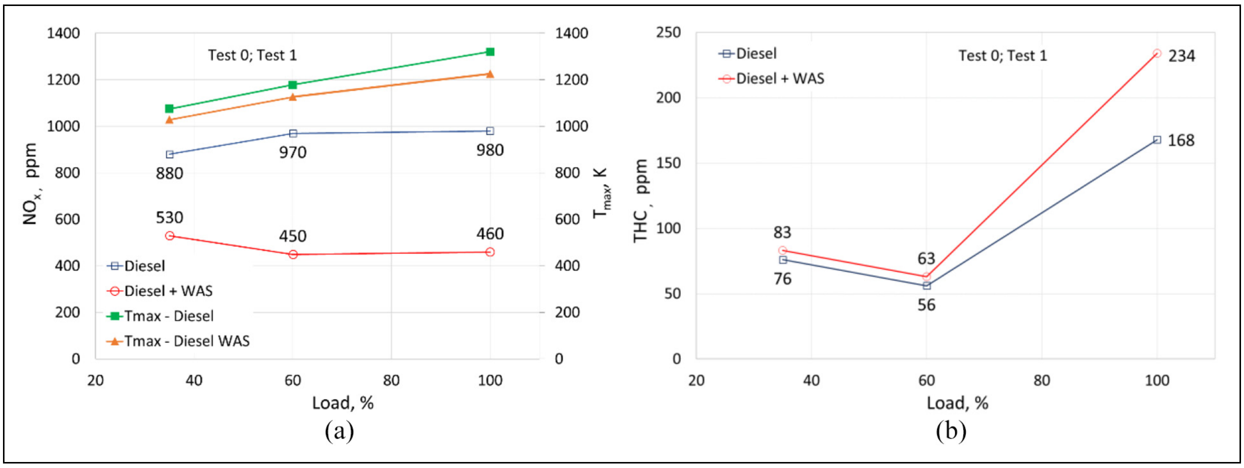

Figure 8. (a) Emissions of NO_x and maximum combustion temperatures and (b) emissions of THC (the test engine fuelled by diesel and diesel–WAS).

970 to 450 ppm) at 60% of engine load compared to a reference fuel engine. Even at 100% engine load, the decrease in NO_x emissions was at a similar level (53%). Reduction of NO_x is related to the reduction high temperature due to vaporized water during combustion and reduced O_2 availability.

Figure 8(b) illustrates the THC emissions measurement results for the above-mentioned engine operating conditions. At the lowest load, no effect on THC emissions was found for the WAS energy fraction burned with DF. At 60% load, THC emissions were at their lowest level, although the addition of WAS caused their increase by 13%, but it was still the lowest compared to other engine loads. A noticeable increase in THC emissions (by 39%) took place at full engine load.

Figure 9 illustrates the measurements of CO and CO_2 for the above-mentioned engine operating conditions. A noticeable effect of WAS burned with DF on the increased CO emissions is observed. The WAS addition leads to the increase in CO emissions by 65%–165% over the entire engine load range compared to the engine fuelled only with diesel. For both fuels, the engine quickly increases CO emissions with the increase in the load. The presence of ammonia in the fuel reduces the pressure and combustion temperature, resulting in higher CO and HC emissions. In the case of CO_2, the effect of WAS on CO_2 emissions was found to be insignificant. Changes in the concentration of CO_2 are not greater than $\pm$18%. For both fuels, the engine decreases CO_2 emissions as the load increases.

Figure 10 illustrates the results of concentrations of toxic components in the engine exhaust gas, where the WAS dose increased from 0 to 380 J/cycle at a constant diesel dose of 1900 J/cycle and the maximum combustion temperature. In this case, there was no significant impact of the WAS contribution on the maximum temperature in the engine cylinder. The peak temperature was at a level 1130 K. The impact of WAS on the reduction of NO_x emissions was confirmed. Increasing the WAS addition results in a regular reduction of NO_x

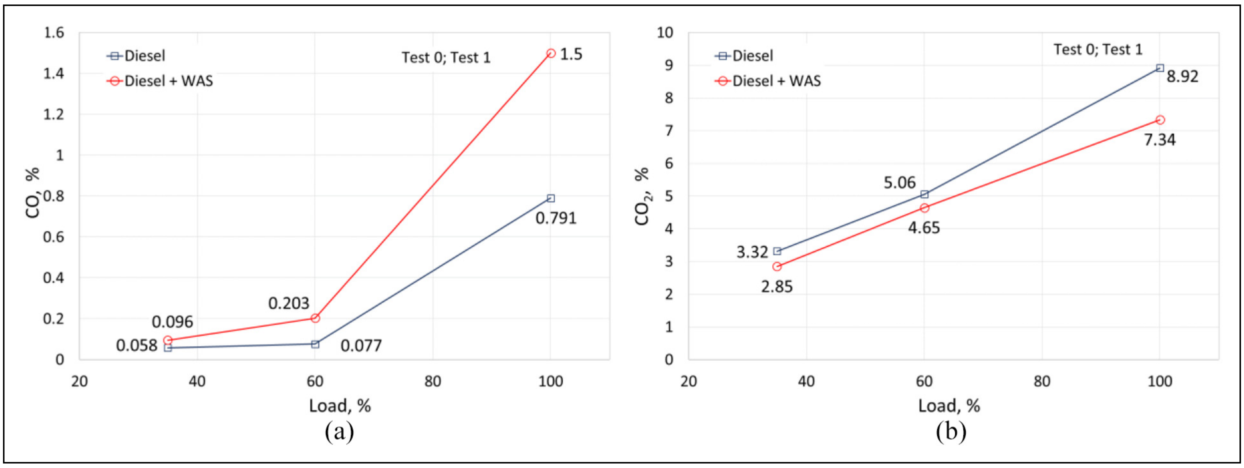

Figure 9. Emissions of (a) CO and (b) CO_2 (the test engine fuelled by diesel and diesel–WAS).

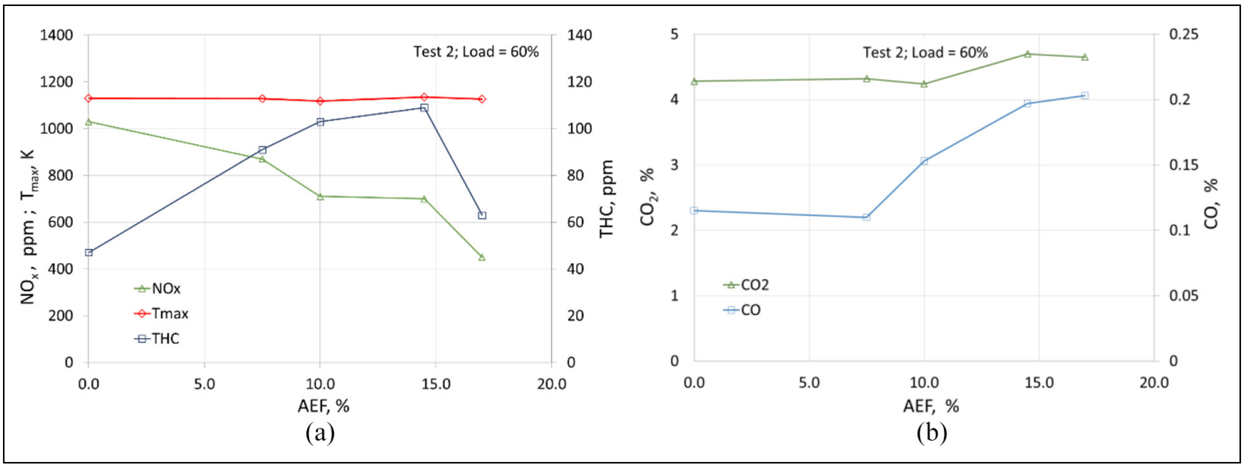

Figure 10. The effect of WAS fraction on concentrations of (a) NO_x and THC and (b) CO and CO_2 and the maximum combustion temperatures at 60% load on the engine.

emissions. At the maximum WAS fraction (AEF = 17%), NO_x concentration was reduced by 56%. Furthermore, increasing the WAS fraction in fuel caused an increase in THC emissions, which was the highest (130%) with WAS fraction of AEF = 14.5%. Adding WAS to DF also resulted in a continuous increase in CO emissions, with the highest level of 76% obtained for WAS fraction of AEF = 17%. However, increasing the WAS fraction in fuel did not cause any significant changes in the concentration of CO_2– the changes were not greater than 10%.

Conclusion

The use of ammonia to fuel a CI diesel engine is one of the possibilities to manage a harmful emission of combustion processes and limiting the consumption of fossil DF, considered to be one of the main causes of the greenhouse effect. The article presents experimental studies of a dual fuel CI engine fuelled with diesel injected directly into the cylinder and WAS injected into the intake manifold. The energy fraction of WAS used for co-combustion with diesel ranged from 0% to 17%. Analysis of the results obtained in the study leads to the following conclusions related to the combustion process, engine stability and emissions:

Combustion process:

1. Co-combustion of DF with WAS in a test engine contributes an increase in ID and CD compared to the sole DF combustion. For the test engine, a change in load does not cause a significant change in ID and causes a significant increase in the CD.
2. At a partial load of 60% of the engine load, the increase in WAS increases the ID and CD. The highest values were achieved for the maximum AEF = 17%.
3. The process of combustion with ammonia is characterized by higher values of pressure rise in comparison to the combustion of sole diesel, making engine operation harder, regardless of its load.
4. A significant increase in the in-cylinder pressure and HRR occurs only when the energy fraction of

WAS exceeds 10%, at the partial load of 60% of the engine load.

5. The presence of ammonia in the combustible mixture has a significant effect on the HRR of a dual fuel mode, affecting mainly the kinetic combustion phase. At the maximum load, the HRR peak value for a dual fuel mode was nearly 100% higher than for the conventional mono-fuel mode.
6. At a partial engine load of 60%, the value of the engine efficiency increases with the increase in the WAS fraction at a 17% fraction, the efficiency was 37.2%, that is, it increased by 4% compared to the conventional engine.

Engine stability:

1. The co-combustion of WAS with diesel compared to DF alone contributes to a slight deterioration of engine operation stability, expressed with COV_{pmax}.
2. As the load increases, the variation of the maximum pressure in subsequent engine cycles increases, indicating the deterioration in the engine stability.
3. Despite a slight deterioration of stability, the test engine within the performed test conditions did not exceed the permissible stability indices for internal combustion reciprocating engines.

Emissions:

1. The addition of WAS to the dual fuel engine reduces the emissions of NO_x compared to a reference fuel engine for all loads analysed.
2. The dual fuel engine, at partial load, is characterized by the HC emissions similar to that of the mono-fuel engine. A noticeable increase in THC emissions (by 39%) occurs at full engine load.
3. The WAS addition leads to the increase in CO emissions over the entire engine load range compared to the diesel-only operation. In the case of CO_2, the effect of WAS on CO_2 emissions was found to be insignificant.
4. Increase in the share of WAS in co-combustion with DF results in a regular reduction of NO_x emissions and an increase in CO and THC emissions and it does not cause significant changes in CO_2 concentration.

Declaration of conflicting interests

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Appendix I

Notation

COV_{pmax}	coefficient of variation of maximum pressure (%)
LHV_D	lower heating value of diesel fuel (MJ/kg)
LHV_{WAS}	lower heating value of WAS (MJ/kg)
$\dot{m}_D$	diesel fuel consumption per cycle (mg/cycle)
$\dot{m}_{WAS}$	WAS consumption per cycle (mg/cycle)
n	engine speed (r/min)
N	cycle index
p	pressure (MPa)
p_{max}	maximum pressure (MPa)
STD_{pmax}	standard deviation of maximum pressure (MPa)
T	temperature (K)
V	cylinder volume (m^3)
V_d	displaced cylinder volume (m^3)
λ	excess air ratio
χ	ratio of specific heats