Chapter 4

Combustion of Gaseous Alternative Fuels in Compression Ignition Engines

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Additional information is available at the end of the chapter

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Abstract

The problem of alternative fuels for combustion engines has been growing in importance recently. This is connected not only with decreasing fossil fuel resources, but also with the growing concern for the natural environment and the fight against global warming. This paper discusses the possibility of utilizing alternative gaseous fuels in compression-ignition engines, using dual-fuel, gas-liquid operation strategy. Current state of the art of this technology had been introduced, along with its benefits and challenges to be countered. The discussion had been supported by authors own research experience on dual-fuel engines. The latest results of research on the impact of gas composition on combustion process in the Common Rail dual fuel engine had been presented, at the same illustrating the environmental benefits of using gaseous fuels. The Utilization of gaseous fuels with varying composition was illustrated systematically, starting with natural gas. The possibility of using fuels with lower content of methane (the so-called low-calorie gases) was shown by the impact of depleting natural gas with carbon dioxide. Industrial gases, such as syngas contain a large amount of hydrogen, carbon monoxide or higher hydrocarbons (ethane, propane). The possibility of fueling CI engines with these gasses was presented by the influence of enriching natural gas with mentioned components. The results cover engine dynamometer tests for different operating conditions with the analysis of the combustion process and detailed emission measurements discussion. The results of experimental studies were supplemented by simulation results, using mathematical models, developed by the authors for multi-fuel enginesr.

Keywords: Dual-fuel, Compression Ignition, Common-Rail, Gaseous Fuels, Natural Gas, Emissions, Performance

1. Introduction

Currently, over 20% of global demand for energy is managed by transport [3], which to a significant extent, has been dominated by internal combustion engines. Heavy duty road transport, maritime transport and almost half of passenger cars are powered by compression-ignition (CI) engines. Increasingly, internal combustion engines have been used for energy generation.

Higher efficiency compared to spark ignition engines and the ability to operate on a wide range of fuels predisposes the modern CI engine to be considered as one of the powertrains of the future. It should be noted that modern combustion technologies available with the dissemination of electronic fuel injection process control and a range of new concepts for exhaust aftertreatment control have made CI engines become more and more environmentally friendly. Simultaneously, wide spectrum of regulations results in clear and efficient conversion of chemical energy contained in the fuel, provided for different carriers of this energy. This flexibility allow modern CI engine to operate on liquid fuels with extremely different physicochemical parameters without the need for structural modifications. The optimization of engine operation can be done at the level of control algorithms [2].

Recent years there has been a significant increase in interest in gaseous fuels. Their share in the global energy balance will soon exceed 21% of the total energy produced [4]. This is due to the fact that regardless of the type of gas and the applied combustion technology, these fuels allow for a significant reduction in toxic emissions. On the other hand, very high availability (deposits of natural gas, shale gas, ability to obtain bio-methane from organic matter, syngas and coal gas from industrial processes) and low price cause that, in the coming years, gaseous fuels will significantly diversify all energy sectors.

Currently, many research centers are developing the concept of powering CI engines with gaseous fuels in a dual-fuel mode. A relatively small dose of liquid fuel, injected into the cylinder, acts as an ignition inhibitor and stabilizes the combustion of gaseous fuel which provides major part of energy. As in the case of mono-operation, the gaseous fuel supplied to the engine can change. Adaptation to the changes in fuel composition is reached by regulating the fuel injection parameters of initial dosage and many other regulation parameters like air amount and temperature, exhaust recirculation, etc.

Appropriately designed CI engine with complex regulation algorithms (often based on artificial neural networks and advanced mathematical models) can essentially be called a multi-fuel engine which can be powered at any time with any liquid or gaseous fuel depending on availability.

For more than 7 years, our team has been carrying out projects in the field of utilization of gaseous fuels in compression ignition engines. In the following chapter the results of dose works had been summarized introducing the reader to the topic of multi-fuel, gas –liquid, engines.

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1.1. Gaseous fuels for CI engines

Limited supplies of fossil fuels and the fight against global warming and greenhouse gas emissions all contribute to the search for new types of fuels that could be effectively used in modern compression-ignition engines [7, 38, 40, 43, 50].

Some properties of the flammable gases that are the basic constituents of gaseous fuels are shown in table 1 [7, 38, 40].

Component name	Calorific value $[MJ/m^3{}_n]$	Density at normal conditions $[kg/m^3]$	Lower ignition limit (% of gas in air)	Upper ignition limit (% of gas in air)	Speed of combustion [m/s]	Auto-ignition temperature [K]	Theoretical air requisition $[m^3/m^3]$
Hydrogen H_2	10,78	0,0899	4	75	0,302	807	2,38
Methane CH_4	35,89	0,717	5	15	0,338-0,67	923	9,54
Ethane C_2H_6	63,77	1,34	3	12,4	0,43-0,856	793	16,7
Propane C_3H_8	91,28	1,97	2,1	9,5	0,384-0,821	783	23,8
Carbon monoxide CO	12,6	1,25	12,5	74	0,024	881	2,38
Hydrogen sulfide H_2S		1,54	4,3	45	-	563	7,14

Table 1. The basic properties of the individual components of gaseous fuels [7, 38, 40].

Of all the above-mentioned flammable gases only methane is a fossil gas. The remaining gases are obtained as a result of the processing of minerals, or via autogenous, natural processes.

Methane is the main component of natural gas. Its content varies depending on the gas source, and is usually between 90 and 99%. Hard coal mines are another source of methane, where significant amounts of this gas are obtained via the so-called de-methanising process of the mines. The main advantage of using methane as combustion engine fuel is its low coal content, which means that its combustion results in much lower carbon dioxide emission levels. Moreover, it is characterised by a very high octane number (125-130), which allows it to be used in engines with a higher compression ratio, and ensures improved general engine efficiency [40, 50]. Natural gas is usually stored in compressed form as CNG (Compressed Natural Gas). It is also possible to store this gas in liquefied form as LNG (Liquefied Natural Gas), but this requires very low storage temperatures of the order of -163 deg. C [43].

Of the remaining gas fuels available, of significance currently are liquefied petroleum-derivative LPG (Liquefied Petroleum Gas) gases, which are a mixture of propane and butane.

Currently, however, particular attention is given to gas fuels obtained from renewable energy sources. Gas fuels obtained from various sorts of biomass, animal excrement, communal waste, sewage, and agricultural/food industry waste are defined as biogas [7, 43, 50].

The composition of biogas is not permanent, and depends not only on the raw materials that it is made of, but also on the technology used for its production. The estimated chemical composition of biogas made according to different methods and materials is shown in table 2 [7].

Component	Content		
	Agricultural biogas	**Treatment plant biogas**	**Landfill biogas**
Methane CH_4	45-75 %	57-62 %	37-67 %
Carbon Dioxide CO_2	25-55 %	33-38 %	24-40 %
Oxygen O_2	0,01-2,0-2,1 %	0-0,5 %	1-5%
Nitrogen N_2	0,01-5,0 %	3,4-8,1 %	10-25 %
Hydrogen sulfide H_2S	10-30 000 ppm	24-8 000 ppm	15-427 ppm

Table 2. Composition of biogas, depending on its origin [7].

During the analysis of the properties of the aforementioned flammable gases in view of their use in self-ignition engines, above all attention must be paid to their relatively high self-ignition temperatures. Another downside of the use of these fuels in self-ignition engines is the relatively narrow limit of their flammability, which in combination with the relatively high global excess air coefficient for these engines restricts gas combustion in the combustion chamber [40, 43].

Unprocessed, directly drawn biogas is heavily contaminated, and contains significant amounts of non-flammable components. Moreover, the hydrogen sulfide that naturally resides in biogas has highly corrosive properties. The methods of treatment of biogas have been well recognised [3, 21], and the need for its purification does not represent a significant restraint to its use on a mass scale. As a result of the treatment process, proper engine fuel, called biomethane, is obtained. After treatment, biogas has a similar composition to natural gas, and therefore has very similar physicochemical properties.

Hydrogen has a very small molecular weight and a high gross calorific value. It has the highest energy-to-mass ratio among the known hydrocarbon fuels, maintaining at the same time beneficial physicochemical parameters in view of fuel combustion in gas engines. The main advantages of this fuel include the lack of emission of carbon dioxide during combustion. The use of hydrogen as an alternative gas fuel is currently restricted by the low efficiency of the process of its extraction. Currently, the production of hydrogen via electrolysis is 3.5 times more expensive than its extraction using catalytic steam reforming of fossil fuels [44]. In molecular form, hydrogen can also be manufactured using the natural gas reforming process. The direct combustion of natural gas is still more effective than its conversion into hydrogen fuel. An extensive analysis of the methods of production of hydrogen has been shown in [13, 30, 44]. The use of hydrogen as fuel for engines is also limited by the high reactivity of the gas, which restricts its storage and transport capabilities, and enforces the use of special non-reactive alloys. Additional constraints are related to the low density of the gas, which imposes

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special requirements regarding the sealing in engine components. Due to the above, clean molecular hydrogen is not a realistic alternative to petroleum-derivative fuels as engine fuel for common use. It must be stressed that there are new concepts for the production of molecular hydrogen, and the highest expectations are currently invested in the use of biological processes - anaerobic fermentation [5, 46] and photolysis of water [31, 37], which are still not fully developed as of today. However, disregarding the development of the technology for the production of hydrogen in the future, it is estimated [4] that the main use of this fuel will be limited to the storage of energy in the form of fuel cells, and as additives to other gas fuels.

Currently, a new technology for gas fuels, HCNG, is being developed. The abbreviation stands for natural gas (or biomethane), enriched with hydrogen to the level of 4-9% of the total volume. Research on these kinds of compositions is being carried out mainly in the United States. The main advantage of HCNG is the ability to significantly reduce the toxicity of emissions in relation to pure CNG. Karner and Frnackfort tests [17] for using this fuel to power spark ignition engines adapted to the use of CNG have shown that compounds below 20% do not require additional adapting or changing of the control programme. It was also proved that a 50% hydrogen content level does not increase the probability of explosion as compared to pure CNG, and can be used with appropriately regulated engine parameters. Adding hydrogen to fuel increases its calorific value. During the aforementioned tests, the engine achieved 5% more power than when powered by pure CNG.

1.2. Advanced combustion and control technologies for alternative-fuel CI engines

Along with the search for new, cleaner fuels for combustion engines there is a need to increase the efficiency of conversion of chemical energy contained in the fuel, with particular emphasis on real engine’s emissions.

A milestone for improving the combustion of liquid fuels in CI engines was the introduction of electronic injection process control. Common Rail Direct Injection (CRDI) gradually replaces traditional, mechanically controlled injection systems in all CI Engine developments, from light-duty to heavy-duty stationary and marine engines. In the electronic Common Rail (CR) injection systems that are commonly used nowadays, it is possible to precisely adjust the injection timing, pressure and duration. Furthermore, CR systems also have the option of dividing the fuel into several doses, injected sequentially, depending on the engine operation conditions. In such systems, usually a very small portion of diesel (pilot dose) is injected early in the compression phase to improve the conditions for the main dose combustion, which is injected some time after the ignition of the pilot dose. Divided injection enables additional control over the combustion process, reduces toxic emissions and noise levels [29].

New control possibilities provided by the introduction of CR injection systems do not cover all potential of CI engines, related to alternative fuels utilization. New modes of combustion are being sought in order to reduce emission levels and fossil fuel consumption. A potential candidate is the Homogeneous Charge Compression Ignition (HCCI) engine concept. In the HCCI engine a premixed air/fuel mixture is introduced to the cylinder. Combustion takes place spontaneously when the homogenous fuel mixture has reached its chemical activation energy and therefore is controlled by chemical reaction kinetics only [18].

The HCCI technology isn't mature enough to be commercially available yet, but it is being intensively developed [54]. Diesel fuel alone is no suitable for HCCI operation due to its low volatility and high propensity. Gasoline HCCI engine concepts have been studied by many researchers [10, 33, 49, 57], showing the possibility to cut fuel consumption by a maximum of 15% while virtually eliminating NOx emissions. Still, stable high load operation is the main issue to be solved for gasoline HCCI engines. One of the possible solutions is to blend gasoline with fuels with high knock resistance properties (diesel or liquid biofuels) [6].

The combination of different gaseous fuels (natural gas, hydrogen, propane) with diesel in HCCI mode is reported to yield low emissions and, to some extent, increase engine efficiency [36, 47]. The same time it is possible to increase alternative fuel energy share even further by substituting diesel fuel with liquid biofuels. The combination of both high cetane number liquid fuel and high octane – gaseous fuel can provide a soft engine run while operating at high compression ratios [11]. It was found that high compression ratio (up to 18:1) engine has an advantage of producing ultralow NOx emissions and high thermal efficiency at steady state operation [15].

The problems with proper control of the combustion process, had caused that the HCCI gas-diesel operation, have consequently evolved to another dual-fuel concept called PCCI (Premixed Charge Compression Ignition) or RCCI (Reactivity Controlled Compression Ignition) [8, 12, 22]. The concept combines the advantages of HCCI combustion while introducing CR direct injection of liquid fuel. It is assumed that small amount of diesel fuel is injected early during the compression phase and therefore can be completely premixed before the auto ignition starts. This keeps the advantages of HCCI operation while, by in cylinder blending, showing good potential in terms of controllability and high load operation. The research on RCCI has focused mostly on the combination of gasoline-like and diesel-like fuels. For the combination of natural gas and diesel limited studies are available. Especially full scale engine test results with such operation are lacking.

It is worth to notice that this technology has a good potential for using a wide spectrum of fuels with significantly different composition and thermal/energy properties. This leads to the concept of a multi-fuel gaseous engine. In multi-fuel engines, currently recognised fuelling methods should be subject to modification. The share of individual fuel types in total energy content should be defined in order to ensure maximum reduction of toxicity of emissions and achievement of maximum engine efficiency. To ensure optimum utilisation of the engine's capabilities, these parameters should be regulated depending on engine speed and load. With such a defined fuelling strategy the selection of the right control algorithm is not a simple task. Modern self-ignition engines with electronic control systems ensure simpler regulation. Therefore, it is believed that further development of multi-fuel engines should be based on Common-Rail type systems [48, 55, 56].

Regulation of the quantity of supplied fuel, as well as start of ignition angle of the pilot dose should be assured. The publications of Stelmasiak [39, 40, 41, 42], Kowalewicz [19, 20], Mikulski, Wierzbicki and others [27, 28, 32, 52] have provided significant input into research on regulation parameters in multi-fuel engines. Moreover, there are also concepts for the introduction of new engine operation-control parameters. Many researchers report on the

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possibility to improve the efficiency of multi-fuel engines by controlling (throttling) quantity [41] and temperature [34, 35] of supplied air. In view of the reduction of the pilot dose to a minimum, the assurance of multi-fuel capability must bring about the necessity to take into consideration the composition of gaseous fuel in control algorithms [9, 16]. Recently, Liu and Fei [23] used neural networks technology to implement complex control algorithms in a dual-fuel engine.

1.3. Motivation and scope of the present research

The above analysis shows that in the upcoming years alternative gaseous fuels will significantly diversify all energy sectors. That is why research effort should be directed towards optimizing the chemical energy conversion processes for those fuels. Particular emphasis should be made on dual-fuel PCCI engines, as they show good potential in terms of fuel flexibility and clean and efficient combustion. As it has been discussed the concept of dual-fuel engines is still a new subject, and the development of optimum control algorithms requires further theoretical and experimental research. Main challenges to be conquered in terms of the combustion process control are: enabling efficient partial load operation with liquid dose minimisation, while keeping beneficial NOX and HC emissions levels and assuring stable transient operation. Even more complicated scientific problem remains to manage all of the mentioned issues for changing gaseous fuel composition.

The goal of the presented research was to discuss the possibility of utilizing alternative gaseous fuels in compression-ignition engines, using dual-fuel, gas-liquid operation strategy and contribute to research the problems in this topic with emphasis on fuel - flexible operation. The latest results of research on the impact of gas composition on combustion process had been presented, at the same illustrating the environmental benefits of using gaseous fuels. The Utilization of gaseous fuels with varying composition was illustrated systematically, starting with natural gas. The possibility of using low-calorie gases was shown by the impact of depleting natural gas with carbon dioxide. Industrial gases, such as syngas contain a large amount of hydrogen, carbon monoxide or higher hydrocarbons (ethane, propane). The possibility of fuelling CI engines with these gasses was presented by the influence of enriching natural gas with mentioned components.

The presented research cover engine dynamometer tests for different operating conditions with the analysis of the combustion process and detailed emission measurements discussion. The results of experimental studies were supplemented by simulation results, using mathematical models, developed by the authors for multi-fuel engines.

2. Engine test stand setup and methodology

The object used for the tests was a four-cylinder, turbocharged and intercooled CI engine, manufactured by Andoria-Mot, with a code mark ADCR. Basic technical parameters of the engine have been presented in Table 3. The engine has been equipped with a Common-Rail fuel injection system with piezoelectric injectors and was controlled by an EDC16C39 electronic controller with factory (optimized for diesel fuel operation) engine maps. The basic input signals introduced to the controller were engine speed and accelerator pedal position. The controller carries out two different fuel injection strategies, depending on the operation parameters. At low rpm and under small loads in the medium speed range, divided injection was performed. In the remaining range, a single fuel charge was injected.

Engine	**ADCR**
Type	diesel, 4-stroke, turbocharged with intercooler
Fuel injection	Common Rail fuel accumulator system
Engine layout	4 cylinder inline, vertical
Cylinder diameter / piston travel	94 / 95 mm
Piston displacement volume	2636 cm^3
Compression ratio	17,5 : 1
Rated Power / rotational speed	85 kW / 3700 rpm
Max. Torque / rotational speed	250 Nm / 1800-2200 rpm
Min. Idle rotational speed	750 rpm
Fuel consumption at torque peak	210 g/kWh
Injection system (Bosch)	accumulator injection system (Common Rail) CR2.0
Turbocharger	radial, with exhaust extraction valve
EGR system	pneumatic EGR valve with exhaust cooler

Table 3. Technical data of the ACDR engine.

The engine was installed on a test bed at the Department of Mechatronics and IT Education of the University of Warmia and Mazury in Olsztyn. The test stand in the configuration used in the research have been shown on Figure 1.

During the tests, engine speed and dynometer Torque were set as demanded values for each test point. The test bench automatics used dynamic dynometer operation and acceleration pedal operation to maintain the set values with accuracy of +-10 rpm and +- 5 Nm for speed and torque respectively. The cooling water and lube oil temperatures were kept at a constant 85/95 deg. C ± 1. After stabilization, steady state measurements were performed, including basic operation parameters of the engine, in-cylinder pressure, injector coil current, opacity and emissions. A time period of 120 s was set for data acquisition in every test run.

A piezoelectric pressure sensor (Type 6056A by Kistler), installed in the first cylinder through the heater plug adapter, was used for recording pressure signal. The sensor, combined with a Type 5018A charge amplifier, was connected via a DAQ card to a PC. The association of the pressure signal with respective rotation angle values was provided by an optical encoder mounted on the engine’s crankshaft. The recording was performed every 1 CA deg., in the full range of the engine’s work cycle.

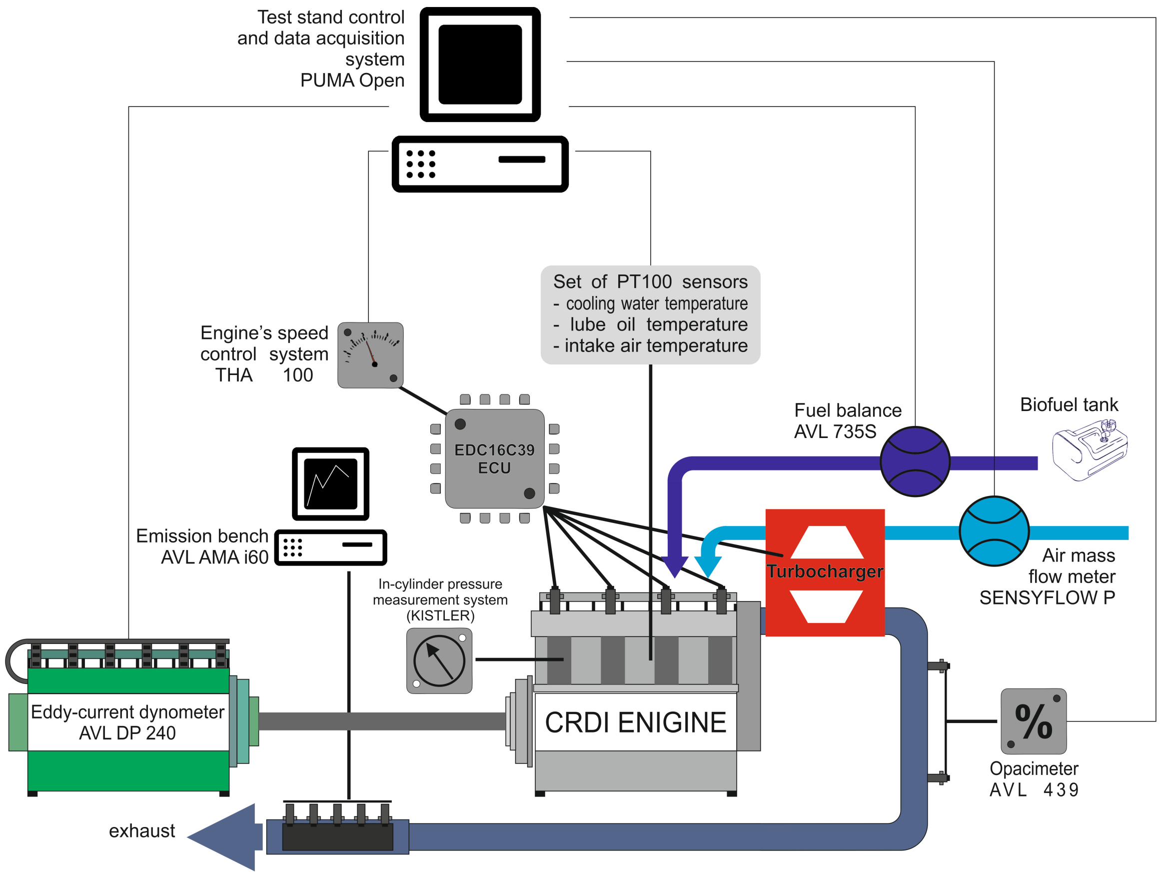

Figure 1. The scheme of the engine test bench used for research.

For each measurement point, after stabilization of the engine's operation parameters, pressure versus crank angle (α) was recorded for 100 cycles. Pressure measurement results were then cycle-averaged, giving pavg (α). Standard deviation was used to calculate the average error of pressure measurement for each crank angle Δpavg (α). The average relative error was then calculated:

$$\Delta p_{avg\_r}(\alpha) = \frac{\Delta p_{avg\_r}}{p_{avg\_r}} \cdot 100 \tag{1}$$

as an estimate of the engine's operation repeatability. Additionally, to determine the moment of injection, a current clamp was mounted on the injector of the indicated cylinder, which allowed recording the current changes on the injector coil. The injector coil current signal was analyzed in the same way as described for pressure signal and was used to determine the start of injection angle (SOI).

The mass of air aspired by the engine (G_{air}), fuel consumption (G_{fuel}) and air temperature at the inlet manifold (T_{air}) were recorded during the tests, and time averaged for each test run. Standard deviation was used to calculate the accuracy of the measurements results. A similar methodology was used to determine the measurement uncertainty of concentration levels of exhaust components.

With the use of the measured values, a number of parameters were calculated. The method of second pressure derivative analysis [32], was used in order to designate the start of combustion (SOC) and to calculate the ignition delay angle (α_{id}). Brake-specific fuel consumption (BSFC) and brake-fuel conversion efficiency (BFCE) were also calculated according to the following formulas:

$$BSFC = \frac{G_{fuel}}{P_e} \tag{2}$$

$$BFCE = \frac{3600}{BSFC \cdot Q_{fuel}} \tag{3}$$

where P_e – engine power, and Q_{fuel} – gross fuel heating value. For the calculated values, the measurement uncertainty was designated using the Kline and McClintock method [14]. The list of parameters recorded directly or indirectly during the test runs, along with maximum uncertainty for all measurement points, is presented in Table 4.

For the second phase of the research the ADCR engine described above was reconfigured, for the dual-fuel operation. The standard EDC16C39 engine controller was dismounted and replaced with a custom made liquid fuel injection control system. The system allowed for adjusting the injection angle and injection timing with the resolution of ±0,5 CA deg. and 1 μs respectively. During the experiments, a specific amount of gaseous fuel was fed to the intake manifold; its flow rate was regulated by mass flow controllers MASS-STREAM D-6371-DR. The separate flow regulator was used for CNG as the base fuel, and another controlled the CO_2 dilution of the gas-air mixture. Both liquid and gaseous fuel injection parameters where adjusted and monitored using a single PC (the same as for in-cylinder pressure signal recording). The scheme of the dual-fuel feeding system was presented on figure 2 [45, 53].

The per-cent energy fraction of liquid fuel (U_{fuel}) introduced to the engine at the given test point (set by engine speed and torque) in dual-fuel operation was designated as follows. The liquid fuel chemical energy introduced to the engine during mono-fuel operation was compared with the total chemical energy of liquid and gaseous fuel necessary to reach the same operating point during dual-fuel operation: the flow regulators

$$U_{fuel} = \frac{G_{fuel} \cdot Q_{fuel}|_{monofuel}}{G_{fuel} \cdot Q_{fuel} + G_{CNG} \cdot Q_{CNG}|_{dualfuel}} \cdot 100\% \tag{4}$$

Where Q_{fuel} is the heating value of the liquid fuel, G_{CNG} and Q_{CNG} stand for CNG consumption and heating value.

No	Parameter	Symbol	Measurement device	Uncertainty level	Unit
Directly measured					
1	Engine rotational speed	N	AVL DP 240	± 5	RPM
2	Torque	Te		± 2	Nm
3	Generated Power	Pe		± 0,2	kW
4	Air aspired to the engine	G_{air}	SENSYFLOW P	± 0,5	kg/h
5	Fuel consumption	G_{fuel}	AVL 735S	± 0,1	kg/h
6	Intake Air temperature	T_{air}	PT 100	± 0,2	K
7	Start of injection angle	SOI	Current clamp	± 0,5	CA deg.
8	Total hydrocarbons *	THC	AVL AMA i60	± 11	ppm
9	Total nitrogen oxides *	NO_X		± 19	ppm
10	Carbon monoxide *	CO		± 13	ppm
1	Carbon dioxide *	CO_2		± 1	ppt
10	Oxygen *	O_2		± 1	ppt
13	Opacity	EGO	AVL 439	± 0,9	%
Calculated					
14	Start of combustion angle	SOC	Second derivative of pressure analysis	± 1	CA deg.
15	Brake specific fuel consumption	BSFC	Equation (2)	± 10	g/kWh
16	Brake fuel conversion efficiency	BFCE	Equation (3)	± 1	%
*- concentration of the compound in the exhaust gases					

Table 4. List of parameters recorded directly and derived indirectly from calculation, along with achieved maximum uncertainty level.

In order to estimate the influence of higher gaseous hydrocarbons addition for dual-fuel engine operation, a simulation model by Mikulski has been used for preliminary research. A detailed description of the developed mathematical model, as well as the methodology of numeric calculations, can be found in another study by the authors [26]. Provided below is a concise summary of the key elements of the model.

The model implemented detailed chemical reaction kinetics of gaseous fuel – air mixture in a dynamic volume, zero-dimensional reactor describing the engine cylinder. The model includes the phases of compression, combustion and decompression in the chamber of a dual-fuel compression-ignition engine. It is assumed that at any point in time, the charge in the cylinder is a homogeneous mixture of air, natural gas, diesel fuel and exhaust fumes. Proportions of individual components vary in the stages of injection and combustion of combustible compo-

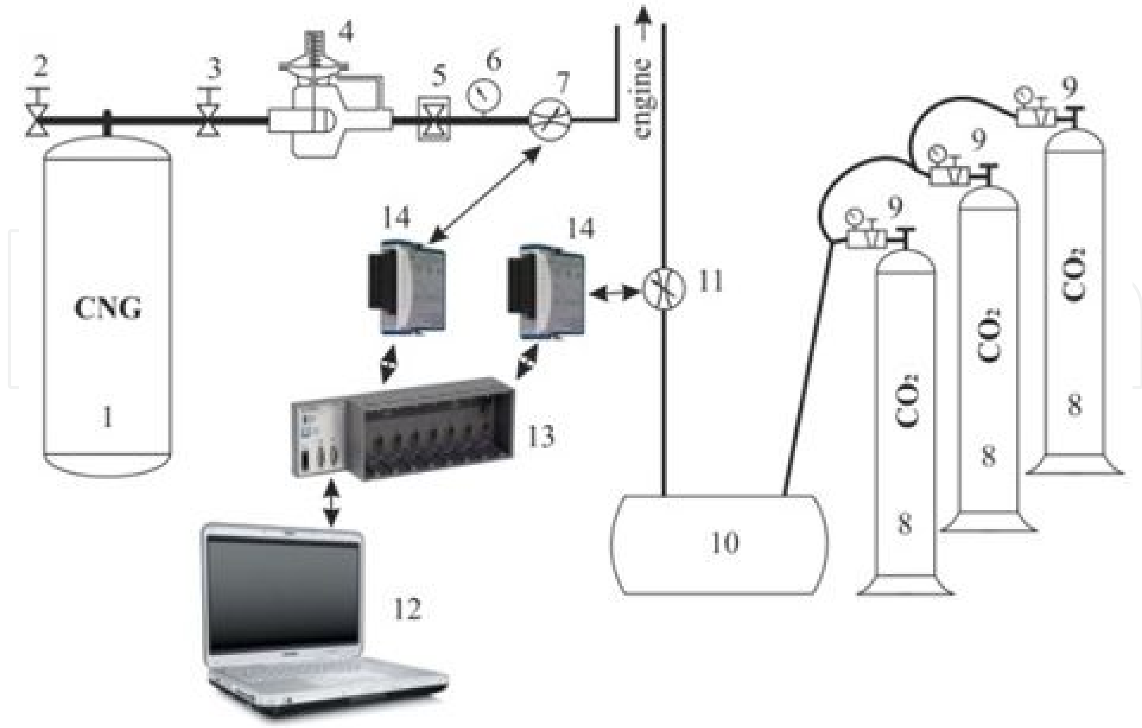

Figure 2. Block diagram of the gaseous fuel supply system: 1 - CNG cylinder, 2 - filling valve, 3 - cut-off valve, 4 - two-stage reducer, 5 - solenoid valve, 6 - pressure gauge, 7 - MasStream gas mass flow regulator, 8 - CO_2 cylinders, 9 - reducers with pressure gauges, 10 - expansion tank, 11 - mass flow regulator, 12 - PC, 13 - CompactRio programmable controller, 14 - i/o cards to control the flow regulators.

nents. The state of charge parameters in the cylinder was described with the use of the energy equation derived from the second law of thermodynamics and the equation of the ideal gas law.

The model included heat exchange with the walls of the combustion chamber as a sum of three streams passing through the cylinder wall and head and the bottom of the piston. During injection of liquid fuel, the thermodynamic parameters of the medium change. The impact of the fuel stream injected into the cylinder was simulated with the use of the authors' proprietary correlation, based on normal distribution [24].

The starting point of diesel fuel combustion is determined by the ignition delay period. The model used the equation proposed by Assanis et al. [1], due to the fact that it provided a more accurate description of the impact of the presence of gaseous fuel in the cylinder on the delay of auto-ignition of diesel fuel [32]. Diesel fuel works as an ignition inhibitor for the gaseous fuel - the ignition point is identical for both fuels. The potential ignition delay of gaseous fuel arises from the calculations of the applied gas combustion mechanism.

The diesel fuel combustion process in the model was simulated with the use of Wiebe's function, with the coefficient md = 0,4, obtained from verification tests.

This approach enables the analysis of the impact of the diesel fuel combustion process on the combustion process of the gaseous fraction. The model of gaseous fuel combustion was based on a one-step macro-reaction of direct oxidation of the main combustible components of the mixture: methane (CH_4), ethane (C_2H_6) and propane (C_3H_8), to carbon dioxide and water. The rate of heat release from the combustion of individual components is determined by the kinetics of the chemical reaction, with the following formula for global reaction rate:

$$\frac{d[C_iH_{2i+2}]}{dt} = A_n exp\left(-\frac{Ea_i}{\bar{R}T}\right)\cdot\left[C_iH_{2i+2}\right]^{a_i}\left[O_2\right]^{b_i} \tag{5}$$

The formulas in square brackets represent concentration levels of specific reagents. The solution of each of the differential equations (3) determines the course of changes of the number of moles of specific reagents (N_i (α)) and, at the same time, the heat release rate from the combustion of gaseous fuel:

$$\frac{dQ_g(\alpha)}{d\alpha} = \sum_{i=1}^{3}\frac{dQ_i(\alpha)}{d\alpha} = \sum_{i=1}^{3}H_i\frac{dN_i(\alpha)}{d\alpha} \tag{6}$$

The constant values found in equation (6) are summarized for specific gases in Table 5.

i		A	Ea	a	b	H_{mol}	H_{mas}
		[-]	[MJ/mol]	[-]	[-]	[MJ/mol]	[MJ/kg]
1	CH_4	$8,3\times10^6$	0,125	0,3	1,3	802,5	50,03
2	C_2H_6	$1,1\times10^{12}$	0,125	0,1	1,65	1423,7	47,3
3	C_3H_8	$8,6\times10^{11}$	0,125	0,1	1,65	2045,3	46,38

Table 5. Values of constants in Eq. (5).

The model was implemented in Matlab environment. The calculations were done, with the calculation step equivalent to 0,5 CA deg. The described model has been used to calculate: in-cylinder pressure, temperature and heat release rates for every CA deg.

3. Dual-fuel CNG-diesel engine test result

Numerous results available in publications dedicated to dual-fuel CI engines have proven that the effective use of such engines is possible within the maximum load range [26, 28, 40, 43].

It is related with the fact that with lower engine loads the air/gas mixture inside the combustion chamber is too lean and does not combust completely. It can be assumed that in this case only the part of the gaseous fuel, in close proximity to injected liquid fuel, is being burnt. The remaining part of the gaseous fuel (on the outskirt of the combustion chamber) does not combust, because the air/gas mixture is beneath its combustibility limits.

3.1. Standard engine controller operation

Tests on the effects of the use of dual-fuel CI engine were carried out using the engine described in the second paragraph. During the initial test phase the engine was run using the original controller - programmed for single-fuel operation, which performed different fuelling strategies depending on operation conditions. At lower rotational speeds the dose of injected liquid fuel was divided into the pilot and main doses, whereas at higher rotational speeds only one fuel dose was injected. Additionally, depending on the operating conditions, the controller changed fuel pressure in the accumulator rail and the angle of start of injection.

During this test phase a controlled dose of gaseous fuel was introduced into the inlet manifold. Inside the manifold and intake the gaseous fuel was mixed with air, creating homogenous mixture which was introduced to the combustion chamber. The liquid fuel dose was adjusted automatically by the controller, to ensure constant engine load output. Thus, as the gaseous fuel content in the total fuel dose was increased, the liquid fuel was reduced. By the change of the injected diesel fuel dose the controller simultaneously changed the start of injection angle.

Selected patterns of combustion chamber pressure registered for two rotational speed values, and recorded emission compositions for these measurements have been shown in figs. 3-5.

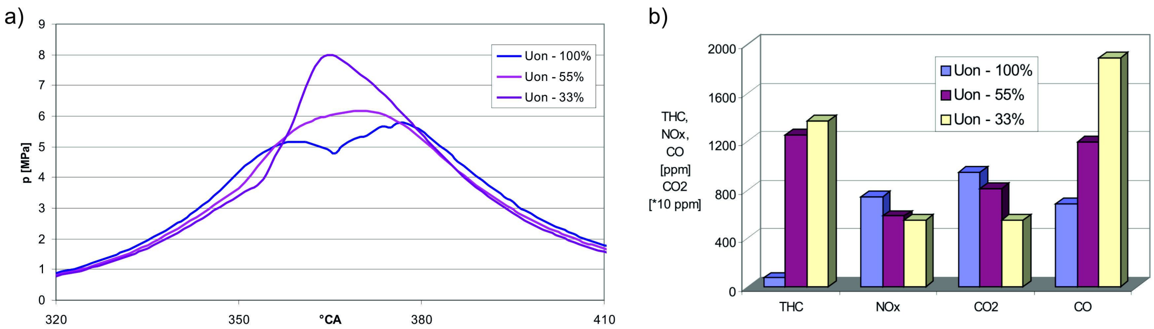

Figure 3. Combustion chamber pressure change patterns (a) and the concentration of toxic compounds emissions (b) at a rotational speed of 1500 rpm, Mo=150Nm for various shares of CNG in the fuel dose.

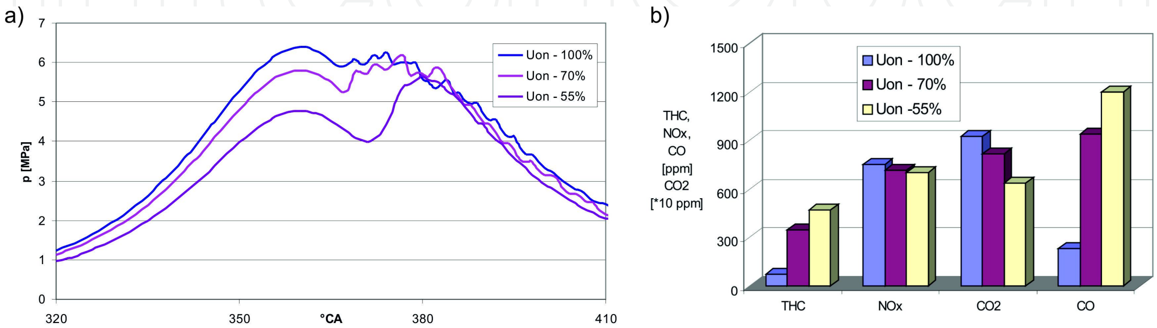

Figure 4. Combustion chamber pressure change patterns (a) and the concentration of toxic compounds emissions (b) at a rotational speed of 3000 rpm, Mo=150Nm for various shares of CNG in the fuel dose.

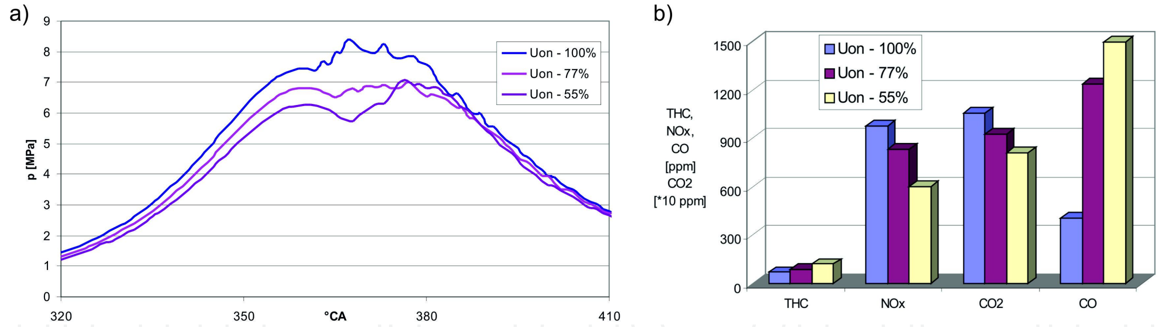

Figure 5. Combustion chamber pressure change patterns (a) and the concentration of toxic compounds emissions (b) at a rotational speed of 3000 rpm, Mo=200Nm for various shares of CNG in the fuel dose.

The presented results clearly indicate significant changes, both in the combustion process patterns and in emission levels. At lower rotational speeds (fig. 3), when a divided dose of liquid fuel is injected, the ignition of gaseous fuel is initiated by ignition of the pilot dose, which resulted in a sudden pressure rise in combustion chamber. At higher rotational speeds (figs. 4 and 5), when a single injection of liquid fuel was applied the rise of CNG content in the fuel dose delayed the pressure incensement, which was caused by delayed injection of liquid fuel. Thus, the combustion of gaseous fuel occurred after the piston reaches the TDC position. In every tested situation the increase of the CNG content resulted in a consequential increase of THC emissions, because of the incomplete combustion of methane. The increase of the gaseous part of the total fuel introduced into the combustion chamber also brings an increasement in CO emission levels, but NO_X and CO_2 emissions are slightly reduced.

With such fuelling strategy the gaseous fuel addition dose also significantly reduced the total engine efficiency, which must be related primarily to the incomplete combustion of the gaseous fuel. It must also be pointed out that the increase of gaseous fuel content in the fuel dose caused the engine to operate with more unevenness. With the content of gas fuel at the level of approx. 50% of the energetic value, engine operation became uneven and made it impossible to maintain constant engine load. It was assumed that the reason for this was the inappropriate injection of the diesel fuel.

3.2. Towards optimized dual-fuel control strategy

In order to define the effect of pilot dose parameters on the gas fuel combustion process in a CI engine, a special controller was used during the further phases of testing, allowing to control the liquid fuel dose parameters, such as injection pressure and start of injection. Control of diesel fuel pilot dose injection parameters, and above all the change of start of injection angle enabled a significant increase in the content of gaseous fuel in the fuelling dose, while maintaining smooth engine operation. Figs. 6-8 show the effect of the pilot dose injection advance angle on the combustion chamber pressure trace, as well as on the content of toxic compounds in exhaust gases, for different diesel fuel amounts.

The pressure patterns shown in the diagrams above indicate unequivocally that governance of the pilot dose injection does affect the air/gas mixture combustion process. Increasing the

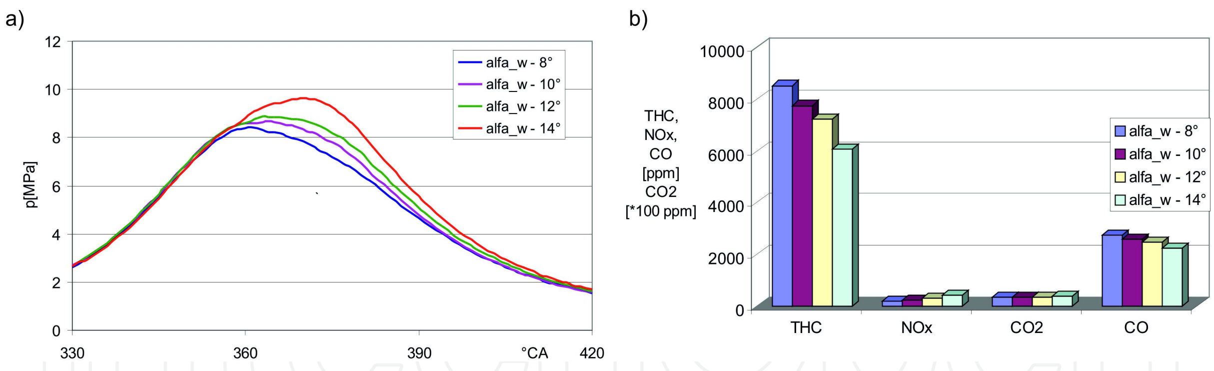

Figure 6. The effect of the pilot dose injection advance angle on the combustion chamber pressure change pattern (a) and toxic compound emission (b) at 3000 rpm, for a 24% content of diesel fuel in the fuelling dose.

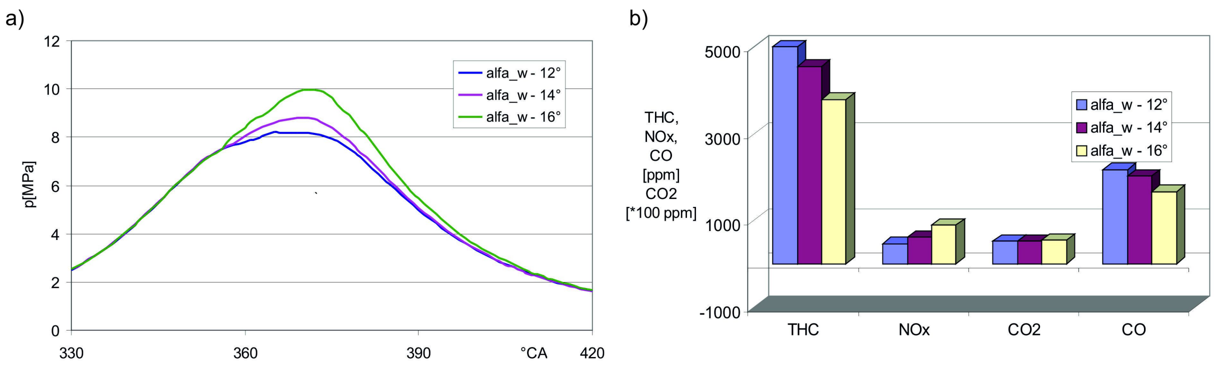

Figure 7. The effect of the pilot dose injection advance angle on the combustion chamber pressure change pattern (a) and toxic compound emission (b) at 3000 rpm, for a 20% content of diesel fuel in the fuelling dose.

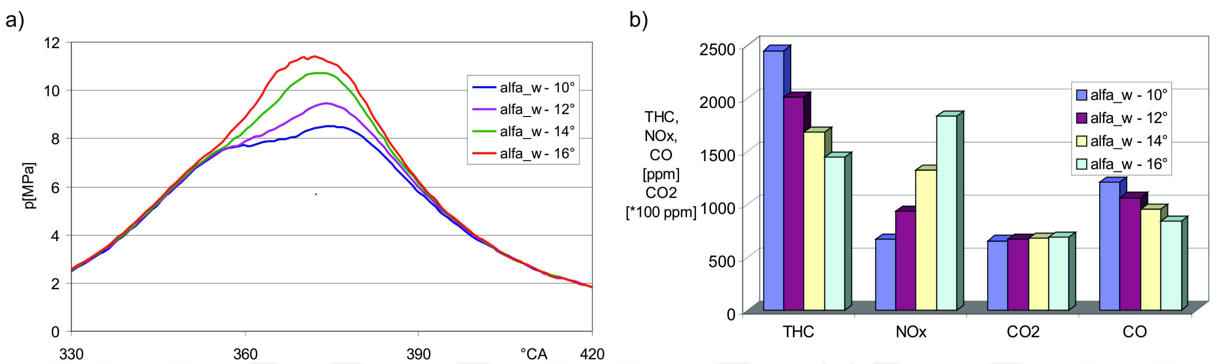

Figure 8. The effect of the pilot dose injection advance angle on the combustion chamber pressure change pattern (a) and toxic compound emission (b) at 3000 rpm, for a 15% content of diesel fuel in the fuelling dose.

injection advance angle of the diesel fuel pilot dose accelerates the combustion of gas fuel, which leads to the increase in maximum combustion chamber pressure and better combustion of gaseous fuel. Changing the injection advance angle does also affect the amount of toxic compounds in emissions. Increasing this angle causes a significant reduction of THC and CO emissions, which should contribute to better combustion of gaseous fuel. Unfortunately, alongside the increase of the injection advance angle, the amount of NOx in emissions also rises as a result of the rise in combustion chamber temperature due to better combustion of methane.

The change of the injection advance angle of the diesel fuel pilot dose also affects engine performance. Fig. 9a illustrates the effect of the pilot dose injection advance angle on the torque value, while fig. 9b illustrates the effect of this angle on engine efficiency for different values of diesel fuel content in the fuelling dose.

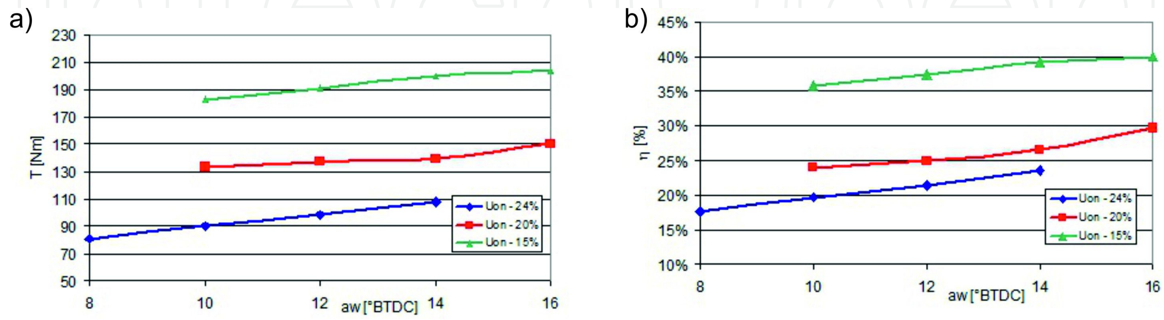

Figure 9. Effect of the injection advance angle of the pilot dose on the value of torque and general engine efficiency at 3000 rpm, and various values of diesel fuel content.

The results presented above allow us to conclude that the effective use of gaseous fuels is possible in the high engine load range. This is mainly due to the fact that at large engine load values the produced gas/air mix has better parameters that assure better combustion of gas fuel. In the case of a lesser gas fuel content the mixture is too lean and its combustion is not complete and too slow, and thus it does not guarantee the achievement of high engine efficiency.

It must be noted, however, that the presence of gaseous fuel in the fuelling dose leads to higher maximum pressure values in the engine, which in turn causes higher loads imposed on the engine's piston-crank mechanism. When the content of gaseous fuel rises, the content of THC and CO is reduced, this contributes to the reduction of the air surplus coefficient value. However, the emission value of NO_X increases due to higher combustion temperatures.

3.3. The effects of diluting CNG with CO_2 on engine performance and emissions

Due to the recent rise in interest in renewable fuels, including biogas, in the next testing phase several measurements were made in order to determine the effect of adding CO_2 to gaseous fuel.

Figs. 10 and 11 illustrate registered combustion chamber pressure change patterns and the content of toxic compounds emissions for different values of CO_2 addition in gaseous fuel. The illustrated combustion chamber pressure patterns clearly indicate that as the content of CO_2 in gas fuel rises, the ignition of gaseous fuel is delayed, which, in constant pilot dose injection conditions, leads to a reduction in combustion chamber pressure and, in effect, to the reduction of engine efficiency. Fig. 12 illustrates the effect of the content of CO_2 in gas fuel on torque and general engines' efficiency values, with constant diesel pilot dose injection parameters.

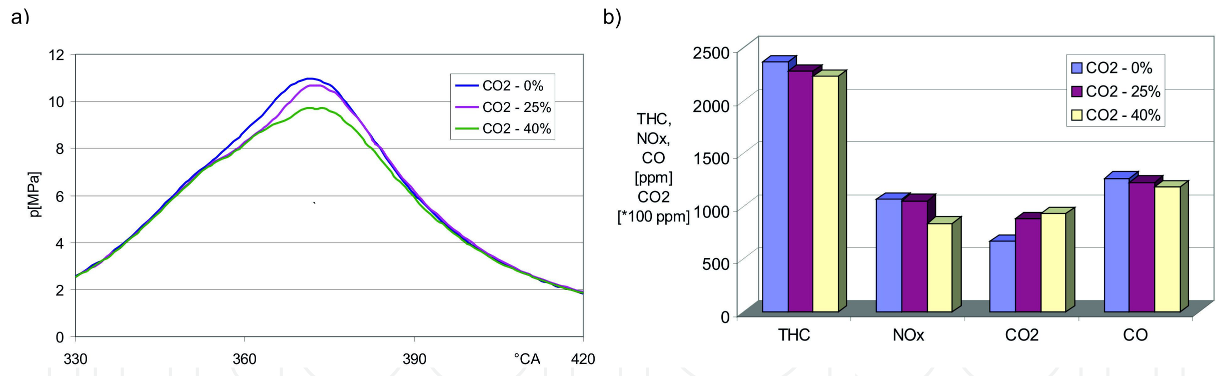

Figure 10. Combustion chamber pressure change patterns (a) and patterns of the presence of toxic compounds (b) at 3000 rpm, with diesel fuel content at 22.5%, for various CO_2 content levels.

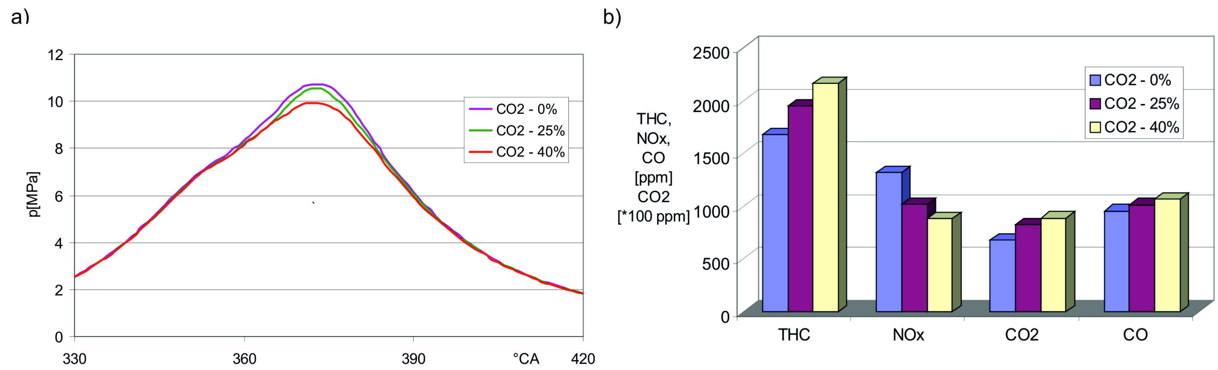

Figure 11. Combustion chamber pressure change patterns (a) and patterns of the presence of toxic compounds (b) at 3000 rpm, with diesel fuel content at 15%, for various CO_2 content levels.

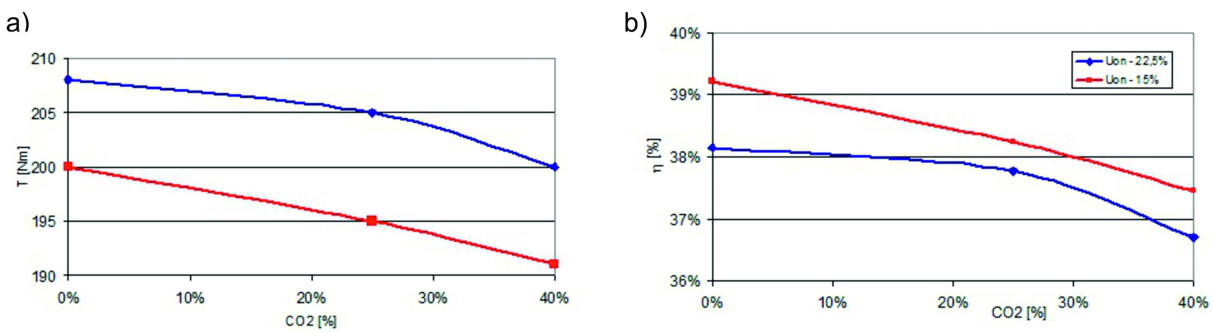

Figure 12. Effect of content value of CO_2 in gas fuel on the value of torque and general engine efficiency at 3000 rpm and various values of diesel fuel content.

During the performance tests on dual-fuel CI engines, a series of experiments was also carried out in order to determine the effect of liquid fuel injection pressure on engine's performance. Fig. 13 presents combustion chamber pressure change patterns and patterns of toxic emissions for different diesel fuel pilot dose injection pressure values. Fig. 14 illustrates the effect of pilot dose injection pressure values on the gained torque and general engine efficiency.

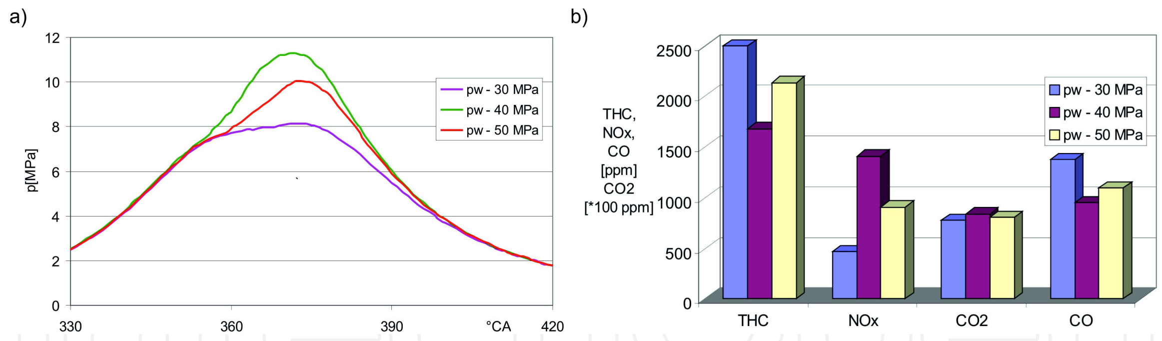

Figure 13. Combustion chamber pressure change patterns and toxic compounds patterns at 3000 rpm, with diesel fuel content at 15%, content of CO_2 in gaseous fuel at 25%, for various pilot dose injection pressure levels.

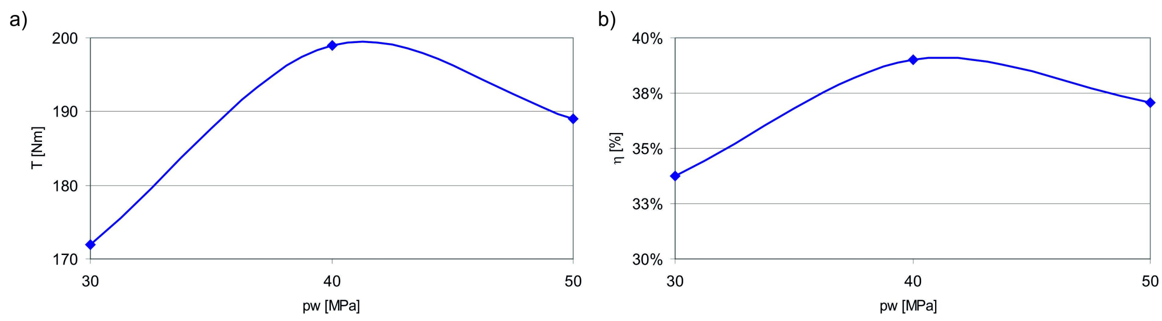

Figure 14. Engine torque and general engine efficiency patterns at 3000 rpm, with diesel fuel content at 15%, content of CO_2 in gaseous fuel at 25%, for various pilot dose injection pressure levels.

The test results presented above suggest that if the pilot dose injection pressure is too low, it causes a significant reduction in general engine efficiency, which is most probably connected with insufficient fuel spraying, which results in the insufficient speed of flame propagation within the combustion chamber and, in effect, gas fuel is not subject to complete combustion. If the injection pressure is too high, on the other hand, it causes surplus fuel spraying, which is also detrimental to the combustion process. The obtained results are compliant with the results described in [40, 43, 53],who note that the optimum results of the operational efficiency of a dual-fuel engine can be obtained with limited pilot dose injection pressure values.

3.4. The effects of C_2H_6 and C_3H_8 enrichment on engine performance and emissions

In order to show the influence of addition of higher hydrocarbons, simulation tests were performed. As the starting point for the study, the operating parameters of an ADCR engine were used, at a rotational speed of 3400 rpm and break torque of 150 Nm. The correctness of the model for these parameters was verified for a broad range of gaseous fuel proportions [25].

Calculations were done for a constant share of gaseous fuel of 70% of the total energy supplied in the fuel. The angle of start of injection of liquid fuel was common for all measuring points and amounted to 6 CA deg. before TDC. The start of ignition was calculated by the model, providing for the impact of gas composition. Similarly, the amount of injected liquid fuel (2,2 kg/h) and its combustion duration (23 CA deg.) were constant for all simulations. It was assumed that the entire volume of gas supplied to the cylinder would participate in the reaction, and the combustion of the gaseous fuel was complete. The composition of gaseous fuel, determined by the share of individual components, was the simulation parameter. The tests assumed a constant level of gas and air consumption.

We shall illustrate the effect of using higher hydrocarbons as gaseous fuel by studying large substitutions of ethane and propane (10%, 30% 80%), for which the effect is clearly visible. Since both additions have different influence on the diesel fuel combustion, also an effect of addition even amounts of those gasses have been briefly described. The results of model simulation for all those cases was shown on figs. 15-17.

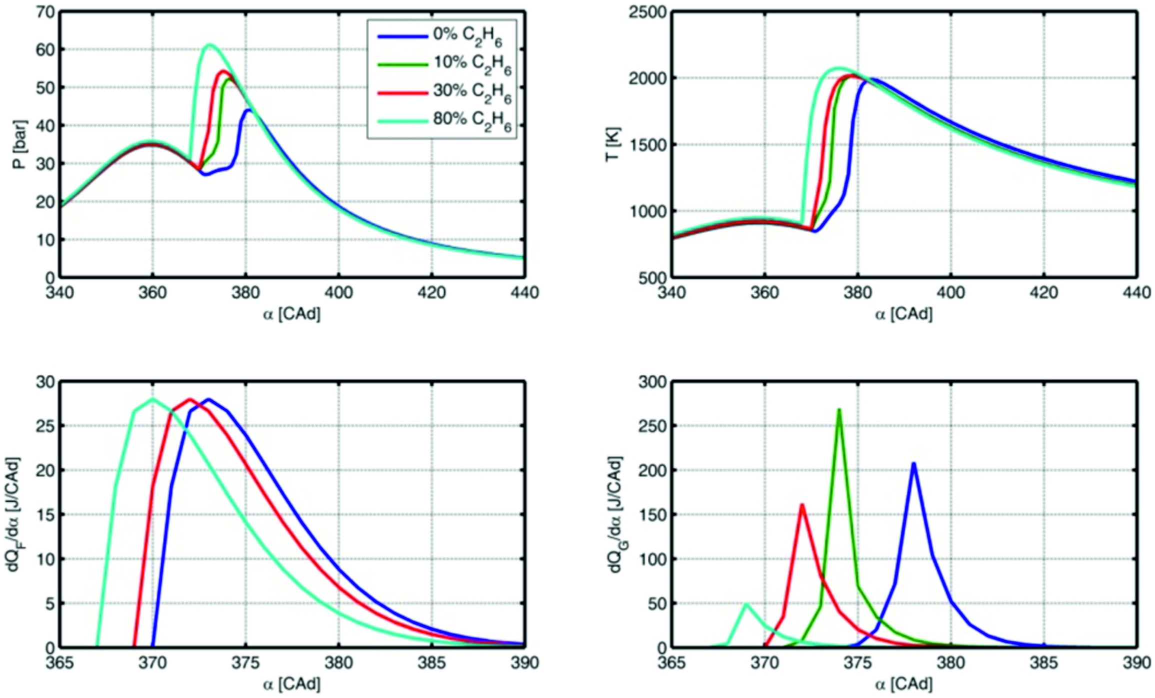

Figure 15. Pressure and temperature values in the cylinder and heat release rate from the combustion of diesel fuel and gas. Different methane content with respect to ethane (C_2H_6).

For most gaseous fuels, a delay of start of combustion is observed with the increase of their concentration levels in the charge [16]. In the case of ethane addition to gaseous fuel, it has been observed that its presence in the engine cylinder reduces the auto-ignition delay of diesel fuel (fig. 15). This reverse effect of ethane has also been observed by other researchers. Research [16] has indicated that with highly reactive gases, such as hydrogen or ethane, the presence of the gas may reduce the ignition delay period. This tendency arises, for the most part, from the high activity of the gases in pre-flame reactions and the creation of auto-ignition locations even before the injection of diesel fuel.

Fig. 15 indicates that even a small addition of ethane also significantly accelerates the ignition of gaseous fuel. In the case of using pure methane, gas would ignite ca.5 CA deg. after the

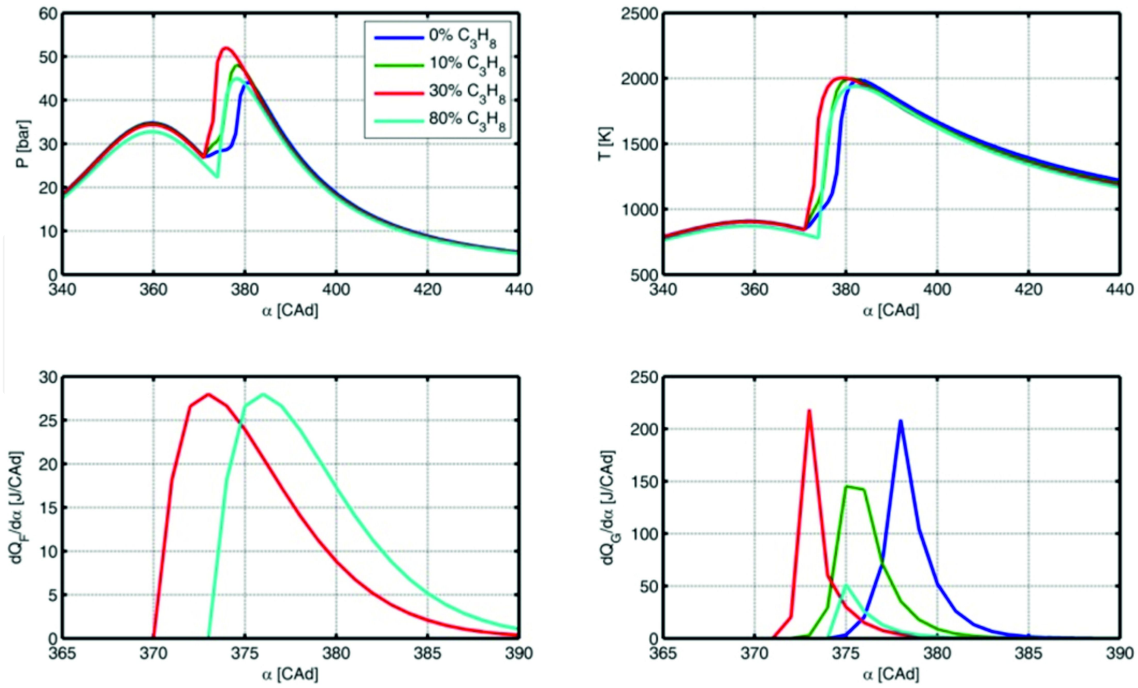

Figure 16. Pressure and temperature values in the cylinder and rate of heat release rate from the combustion of diesel fuel and gas. Different methane content with respect to propane (C_3H_8).

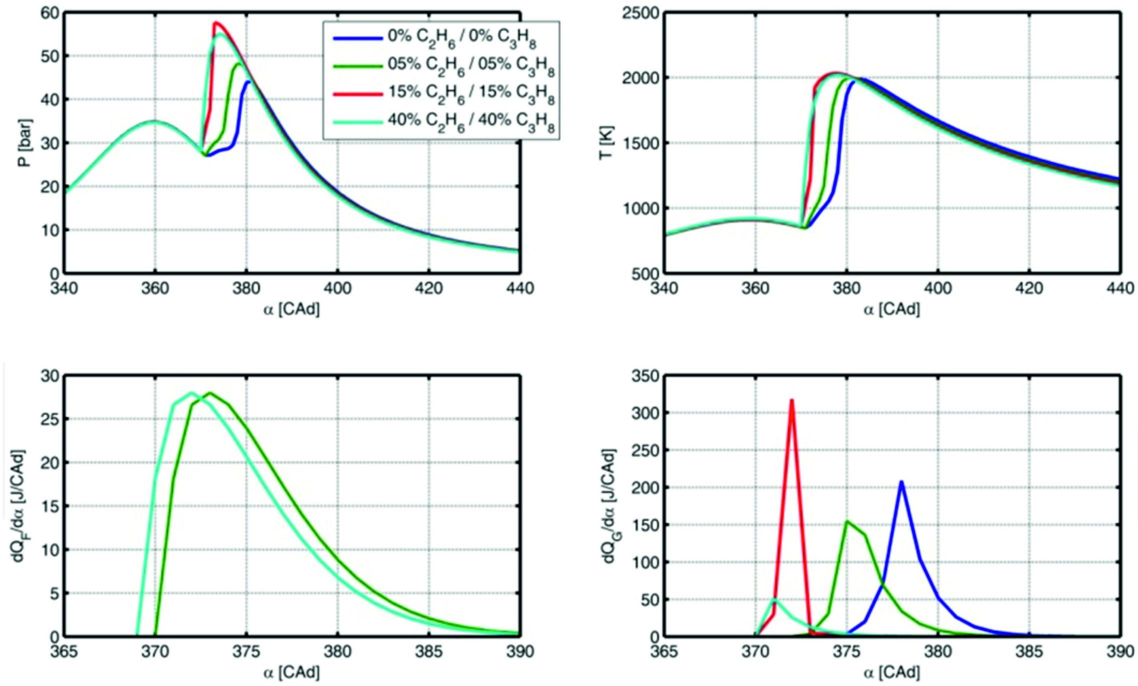

Figure 17. Pressure and temperature values in the cylinder, and rate of heat release from combustion of diesel fuel and gas. Different methane content with respect to ethane (C_2H_6) and propane (C_3H_8).

ignition of liquid fuel. On the other hand, an addition of 20% of ethane already caused the gaseous fuel to ignite together with the initial dose ignition.

The addition of propane results in a minor increase of the diesel fuel ignition delay, as shown by the data in fig 16. A slight extension of the auto-ignition delay has been observed repeatedly for increasing the percentage weight of ethane in the analysed fuel. The delta between the extreme samples (100% methane and 20% methane / 80% propane) exceeded 3 CA deg. This had significant impact on the pressure and temperature in the cylinder – fig. 16.

The length of gaseous fuel combustion was slightly higher for samples containing up to 20% additions of both C_2H_6 and C_3H_8, as compared to the base samples - marked as 100% CH_4 (fig. 15 and 16). For higher concentrations of ethane and propane, a slight increase of combustion rate has been observed, correlated to the increase of maximum heat release rate. In each case, the earlier ignition of gaseous fuel would generate a more rapid increase of pressure and temperature, combined with the increase of maximum combustion pressure and temperature. An increase of maximum pressure by 33% for even a small (10%) addition of C_2H_6 significantly improves the performance of the engine, powered by ethane-enhanced gaseous fuel, as compared to base methane fuel. The observed increase of combustion temperature (by 5% on average) with even minor enhancements with ethane may, at the same time, increase the emissions of NO_X.

Important data on the combustion process in the engine powered by the mixture of diesel fuel, methane and ethane was provided by the analysis of the heat release rate from combustion of both gaseous combustible components This analysis indicated that earlier ignition of gaseous fuel is caused by earlier ignition of ethane, which requires less energy to initiate combustion. Rapid heat emission from combustion of ethane also accelerates the ignition of methane, which, however, has always had a slower ignition than ethane. With identical percentage weights (50%), ethane burns longer than methane, at the same time generating smaller heat increments, despite its higher calorific value. An increase of the length of burn of gaseous fuel as a whole, observed for additions of up to 20% of methane, arises in equal proportions from the distribution in time of combustion of individual gaseous fractions (ethane ignites quicker than methane) and from the slower rate of the ethane oxidation reaction.

Similarly to ethane addition, the pressure and temperature of combustion of the methane-propane mixture was determined mainly by the kinetics of combustion of gaseous fuel. Identically to ethane, propane accelerated the ignition of the gaseous mixture by lowering the activation energy. The quicker ignition of propane enriched gas mixture generated a higher maximum temperature and pressure than for pure diesel/methane; however, due to the increase of ignition delay caused by increasing the concentration levels of propane, the obtained increments were smaller than for ethane. This effect also explains why, despite the higher calorific value of propane, the sample with the highest concentration level of the gas (80% propane) generated smaller increments of pressure and temperature than 10% propane.

In the case of adding equal volumes of ethane and propane to methane, the ignition point of the diesel fuel does not visibly change (fig. 17). This means that the accelerating effect of ethane and decelerating effect of propane appear to cancel each other out. The combustion process is determined by the moment of ignition of gaseous fuel, which occurs for all additions, together with liquid fuel ignition (fig. 17). Still ethane seems to have stronger inhibiting effect on diesel fuel auto-ignition, compared to the ignition delay increase caused by propane.

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4. Conclusions

It is clear that methane – based gaseous fuels, especially off renewable origin, will have an important role in global energy share in the nearest future. It was proven that modern CI engines operating in dual – fuel PCCI mode are a good solution for effective and clean gaseous fuel utilization.

The material discussed proves that dual-fuel technology is already mature enough to be used at steady state operation – for energy generation for example. We have showed that basic regulation strategies can help minimizing diesel consumption to 10% of total energy introduced with the fuel. The engine efficiency can be maintained at the same level as for standard diesel engine and in some cases slight efficiency improvement can be reached. This is combined with lower CO and CO_2 emission and can be maintained for different gaseous fuel composition including low calorific gases with high CO2 content.

The major drawbacks that remain are: higher NOx and THC emmision and poor performance at low loads – to be dealt with more sophisticated control algorithms. Also successful transient control is a challenge – due to the fact that mutual interactions of gaseous and liquid fuels haven't been sufficiently understood. Future implementation should focus on dealing with dose issues - discovering new control concepts for the engine itself, combined with methane aftertreatment technology development.

Also the posiboilitie of introducing a multi-fuel engine, which is able to operate on different gaseous fuels, have been shown. Gaseous fule composition changes have a significant effect on the combustion process.

Also it was proven that real progress in the field of multi-fuel RCCI, PCCI and HCCI concepts is possible with proper, control oriented simulation tools as support. Development of dose tools should be therefore an equally important goal as development of new engine concepts itself.

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