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Numerical investigation of the impact of gas composition on the combustion process in a dual-fuel compression-ignition engine

Maciej Mikulski [a,*], Sławomir Wierzbicki [b]

[a] TNO Automotive, Automotive Campus 30, 5708 JZ, Helmond, The Netherlands
[b] University of Warmia and Mazury in Olsztyn, Faculty of Technical Sciences, 46 A, Słoneczna St., 10-710, Olsztyn, Poland

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ABSTRACT

This study discusses the model of operation of a dual-fuel compression-ignition engine, powered by gaseous fuel with an initial dose of diesel fuel as the ignition inhibitor. The study used a zero-dimensional multiphase mathematical model of a dual-fuel engine to simulate the impact of enhancing Natural Gas (NG) with other gases on the combustion process. The model simulated the thermodynamic parameters of the gas mixture in the cylinder of a dual-fuel (NG/Diesel), turbocharged, four cylinder CRDI (Common-Rail Direct Injection) engine. The tests discussed herein were conducted for steady state engine operation, for partial load and constant consumption of gaseous fuel. In the discussed tests, carbon dioxide and higher hydrocarbons (ethane and propane) were used as additions to NG.

It has been shown that a change of gas composition has a significant impact on the combustion process and parameters of operation of a dual-fuel engine. The combustion of gas additives largely determines the combustion of both the main component of gaseous fuel and the initial dose of diesel fuel. The addition of higher hydrocarbons to methane can improve engine performance by as much as 6% with additions of higher carbon amounting to 20% of total fuel volume. Also, it has been shown that changes of gas composition significantly impact the ignition delay of the initial diesel dose.

1. Introduction

Increasing demand for gaseous fuels, arising from the depletion of fossil fuels, as well as environmental and economic aspects, defines new requirements for combustion engines. Among combustion engines, particularly important are multi-fuel compression-ignition engines, which are able to work with various gaseous fuels, with a minimum pilot dose of diesel fuel as the ignition inhibitor.

Many authors have conducted research on optimal construction parameters of such engines. Bora et al. Bora et al. (2014) investigated the effect of compression ratio on performance end emissions of a dual-fuel engine running on raw biogas. Increasing the compression ratio from base value of 16–18 lead to an increase in break thermal efficiency by 3.5% for maximum loads. At the same time a decrease in Carbon monoxide (by 26.2%) and total hydrocarbons (42%) emission was observed, with increasing NO_x and CO_2. The authors suggested to operate the dual fuel engine at highest possible compression ratios, limited by knock combustion of gaseous fuel.

The dual-fuel system requires particularly careful selection of the regulation characteristics of the engine to ensure correct combustion of the gaseous mixture. Studies by a number of authors have indicated that it is advisable to regulate parameters such as volume and temperature of aspirated air and gas. Heating up the mixture has been researched i.e. by Motyl and Lisowski (2008). Among the regulation parameters the key impact on combustion in dual-fuel engine is exerted by the setting of parameters of liquid fraction injection. The crucial aspects here are the amount of injected fuel (Barik and Murugan, 2014; Liu et al., 2013; Papagiannakis et al., 2007; Wierzbicki and Śmieja, 2014), fuel injection advance angle (Liu et al., 2013; Raj et al., 2013; Ryu, 2013; Roy et al., 2014; Mikulski et al., 2015a) and the quality of spraying the liquid fuel dose (Raj et al., 2013; Ryu, 2013). In 2009 Sahoo et al.

Abbreviations: ANN, Artificial Neural Network; CA, crank angle degrees; CI, compression ignition; CRDI, Common Rail Direct Injection; GEP, Gene Expression Programming; HCCI, Homogeneous charge compression ignition; NG, natural gas; SI, spark ignition; TDC, Top Dead Center; CH_4, methane; C_2H_6, ethane; C_3H_8, propane; CO, carbon monoxide; CO_2, carbon dioxide; NO_x, nitrous oxides.

* Corresponding author.
E-mail addresses: maciej.mikulski@tno.nl (M. Mikulski), slawekw@uwm.edu.pl (S. Wierzbicki).

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Nomenclature and units

[X]	molar concentrations of specific chemical compounds [mol/m^3]
Ea	Energy of activation [MJ/mol]
H	calorific value [MJ/mol]
N	number of moles [mol]
P	chemical Power of the introduced fuel [kW]
p	in-cylinder pressure [bar]
Q	energy from combustion [J]
T	in-cylinder pressure [K]
α	Crank shaft rotation angle [CA]
Φ	air/fuel ratio [−]

Subscripts

air	air
d	diesel fuel
g	gaseous fuel
i	individual combustible gaseous component (1 − methane, 2 − ethane, 3 − propane)
SOC	start of combustion

(2009) published a critical review of papers that discussed the effect of engine parameters on the performance of dual-fuel gas-diesel engines. The impact of those parameters is accurately described in the literature. One of the key conclusions of the review was that, for lean gas and air mixtures, gaseous fuel may combust only in the flame propagation zone of the liquid fuel combustion. With poor spray of the diesel fuel, the propagation zone and the amount of gas that does not participate in the reaction may reach 50%.

With the increasing role of gaseous fuels, the composition of the supplied gas becomes one of the key factors for the combustion process (Mikulski et al., 2015b); the composition may vary (in the case of natural gas) depending on the mining site or the production and purification process (in the case of biogas). Increasingly, engines are fuelled by waste gases from industrial processes, with high carbon dioxide content. In 2014 Kakaee et al. (2014) published an in-depth review of technical papers concerning impact of gas composition on the combustion process in gaseous engines. Most of the papers that were discussed, considered spark-ignition HCCI systems. It was stated as a main conclusion, that changes of gas composition can significantly affect both the operation of the entire engine and its emissions. Currently, the mechanisms of combustion of specific combustible components of the gaseous mixture are relatively well known (Fomin et al., 2014; Fischer and Jiang, 2014). Basic research of the combustion process of the main combustible component of natural gas, i.e. methane, prove (Pawlaczyk and Gosiewski, 2009; Miyata et al., 2015) that the process is very complex and comprises a set of a total of 132 intermediate chemical reactions, the speed of which are mutually correlated by the concentration levels of individual reagents.

In the dual-fuel engine however, the combustion process is much more complicated. Individual-fuel fractions interact strongly with each other. Available studies in this field indicate that changing the gas composition significantly alters the combustion process of liquid fuel. This impact cannot be fully explained only by a different burning rate and different calorific values of specific reagents. Thus, combustion of diesel fuel has considerable impact on the combustion process of gas, and vice versa. The temperature and composition of a mixture have been found to have a significant effect on the maximum pressure of combustion, the temperature and pressure growth rate after compression-ignition and its delay (Korakianitis et al., 2011). It has been shown that there is a limit of the initial temperature which, when exceeded, eliminates any changes in the operational parameters of the engine work. The authors conclude that the temperature and composition of the fuel mixture may be used as regulation parameters of an dual-fuel engine. This has been recently confirmed by Maizonnasse et al. (2013) for dual-fuel engines working on high gas-oil substitutions levels.

Another study (Czerwiński and Comte, 2001) examined the effect of rapid changes of the gas composition on the operation of a spark-ignition engine fuelled with natural gas; it has been shown that in such cases, the power, efficiency and emissivity of an engine may change. Moreover, considerable amounts of non-flammable components of gas made the work of an engine uneven. If a pilot dose is used in an engine, the smaller the “gaseous fuel/diesel oil” ratio, the more similar the effect of changes of the gas composition will be to those observed in a spark-ignition engine.

Lim et al. (2015) used methane enriched biogas blends (81–97% CH4) for fuelling a CNG-diesel city bus in real driving test cycles (ETC and NIER -6). The study showed considerable decrease in emissions for lower methane concentrations. Furthermore no significant difference in fuel economy was observed between the gases in either of the test cycles. Chandra et al. (2011) obtained similar conclusions for small stationary engine converted from CI to SI mode, operated on methane enriched biogas.

Mustafi et al. (2013) carried out a comparative study of gaseous engines working on different biogases. Significant differences were observed in emissions between fuels. Henham and Makkar (1998) examined the engine performance fuelled with different quality of biogas. The authors were able to reach only 60% substitution of diesel with low quality biogas, with high concentration of CO_2. The limit was due to knock combustion. Enriching the gas with methane gave a significant improvement in the gas-diesel substitution level.

Azimov et al. (2012) have examined the impact of hydrogen and carbon dioxide additions on the operation of the dual-fuel engine. A special work site was used, allowing the researchers to prepare a fuel mixture of any composition. An increased amount of hydrogen has induced an increase in the mean combustion temperature and efficiency, but also a significant increase in NO_x emissions, and has shortened the ignition delay period. The results have also shown that when CO_2 content in the gas reaches 34%, the rate of maximum pressure rise, as well as the mean combustion temperature, thermal efficiency and NO_x decrease, despite the increase in the fuel mass fraction burned. This confirms earlier experimental results by Bari for dual-fuel diesel engine (Bari, 1996), who observed that the presence of CO_2 in biogas up to 40% did not deteriorate the engine performance. Furthermore, the addition of O2 to the gas would reduce the ignition delay, whereas the CO would enhance the flame speed. The effect of oxygen enriched air on the performance of a biogas dual-fuel engine, has been further investigated by Cacua et al. (2012). The results indicated that performance drawback effects associated with large CO_2 concentrations, can be successfully limited by a small addition of oxygen.

Many works have been devoted to research the effect of using different bio-fuels as liquid pilot dose, on dual-fuel engine operation (Korakianitis et al., 2011; Luijten and Kerkhof, 2011; Banapurmath et al., 2009; Mikulski et al., 2016a). All of them reported significantly different combustion process duration for changes in liquid fuel composition. Korakianitis et al. (2011) investigated the effect of gas composition on the process of combustion, in a dual-fuel engine using rapeseed methyl ester (RME) as pilot fuel to ignite low-calorific gases. The parameters of engine operation with such fuels can be improved considerably with a small addition of hydrogen.

1.1. Motivation for the present research

Studies on gas engines clearly indicate that changes to gas composition have significant impact on the working parameters of the unit. Current control system solutions for dual-fuel engines do not provide for the gas composition in engine control algorithms. Including this parameter might improve the performance and durability of the engine and reduce emissions. To develop such algorithms, a detailed, both qualitative and quantitative, description of the mutual effects of specific components on the combustion process in a dual-fuel engine is necessary. The current state of the art in this field appears insufficient. The reason for this is that it is not possible to develop a coherent description of such interactions only through experimental research. This arises from the fact that the observable process is the combined combustion of diesel fuel and gas in the cylinder of the dual-fuel engine. Since most of the latest works concerning the impact of gaseous fuel composition are experimental research, for different kinds of engines, they don't explain the nature of the problem. Introduction of an appropriate mathematical model of a dual-fuel engine, verified for a broad range of parameters of engine operation, may be a method of analysis of the phenomenon discussed herein by enabling observation and analysis of specific sub-processes in the combustion chamber.

Simulations of dual fuel NG/Diesel combustion have been reported with different approaches and for different purposes. The research works include detailed 3D CFD simulations (Nieman et al., 2012; Puduppakkam et al., 2011), or faster reaction kinetic based multi–zone simulations (Egüz et al., 2013a, 2013b; Eichmeier et al., 2014). Main works included simulations for PCCI or RCCI combustion modes. Despite much longer history of the concept classic dual-fuel compression-ignition with late injection timing and unpremixed diesel combustion, is still relatively uncharted territory as for model based approaches. Also, commercially available engine simulation packages (Yang et al., 2015) do not reflect all sub-processes in the cylinder of such an engine which makes them unsuitable for the discussed purpose. A very long computation time is another common drawback of the above models, which limits their direct use for control development.

Among the models dedicated to dual-fuel engines in literature on the subject, Black Box type models dominate – ANN (Artificial Neural Network) (Roy et al., 2014; Yang et al., 2015; Yusaf et al., 2010) or GEP (Gene Expression Programming) (Roy et al., 2015). These models base their predictions on a vast base of results of experimental research, building on correlations between specific operational parameters of the engine. The models may be a good starting point for developing the control algorithm (d'Ambrosio et al., 2014), but only for the engine on which the model has been verified. Such models are not based on the physical nature of phenomena occurring in the engine, and thus are unable to assist in comprehending them. The few available thermodynamic models (Alla et al., 2001; Mansour et al., 2001; Koszalka, 2014) focus on specific sub-processes and may be used as sub-models in complete models. However, there are no comprehensive models of a dual-fuel engine cycle that would include all processes in the cylinder and provide sufficiently accurate prediction on the changes of the thermodynamic parameters of the medium, using only basic data of the object, available from its technical documentation. A comprehensive model of a dual-fuel engine has been proposed i.e. by Papagiannakis et al. (2007). Among the available models, only a handful has undergone complete experimental verification, which further diminishes their reliability.

To compensate on those drawbacks, in the attempt to create a fast and reliable dual fuel combustion model which can be used for control development as well as to give insight on the in-cylinder processes, the authors proposed their own approach. A zero-dimensional single-zone model with Wiebe fit as diesel burn rate representative and single-step macro-reaction kinetics describing gaseous fuel combustion had been introduced (Mikulski et al., 2015c). The model underwent a detailed, multi-step verification process, demonstrating high conformity of calculation results with experimental data for estimations of engine pressure and performance values (Mikulski, 2014).

Despite its relatively simple and partially phenomenological nature the model proved to be able to capture the main trends of the mutual impact of the combustion of liquid fuel on gaseous fuel burn rate, this way providing valuable insight on the phenomena of dual-fuel combustion (Mikulski et al., 2015a). In this study, the same model was used to preform trend analysis of the impact of methane enhancement with carbon dioxide and higher gaseous hydrocarbons on the combustion process in the combustible engine. The goal of the study was to contribute towards estimating the impact of gaseous fuel composition on the combustion characteristics and performance of dual-fuel CI engines.

2. Methods

2.1. Dual-fuel engine mathematical model used in the study

In simulation tests, we used a mathematical model of the working cycle of a dual-fuel engine powered by gaseous fuel with an initial dose of diesel fuel, including the phases of: compression, combustion and decompression. A detailed description of the developed mathematical model with in-depth discussion on the assumptions, as well as the methodology of numeric calculations, can be found in another study by the authors (Mikulski et al., 2015c). Provided below is a concise summary of the key elements of the model. The design structure of the model is illustrated in Fig. 1.

The developed model assumes 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 components. The state of charge parameters in the cylinder was described with the use of the equation of 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, the head and the bottom of the piston. The heat exchange model was based on the gas – cylinder wall convection, conduction through the wall and cylinder wall – cooling fluid convection.

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 author's proprietary correlation, based on normal distribution (Mikulski, 2014). The starting point of diesel fuel combustion is determined by the ignition delay period. The discussed model addressed this using the equation proposed by Assanis (Assanis et al., 2003), due to the fact that this equation provides a more accurate description of the impact of the presence of gaseous fuel in the cylinder on the auto-ignition delay of diesel fuel (Mikulski, 2014):

$$\tau_{id} = 2,4\phi^{-0,2}p^{-1,02}\exp\left(\frac{E_a}{\bar{R}T}\right) \tag{1}$$

Diesel fuel serves as the ignition inhibitor for the gaseous fuel. Therefore, it is assumed that the angle of start of combustion α_{SOC}

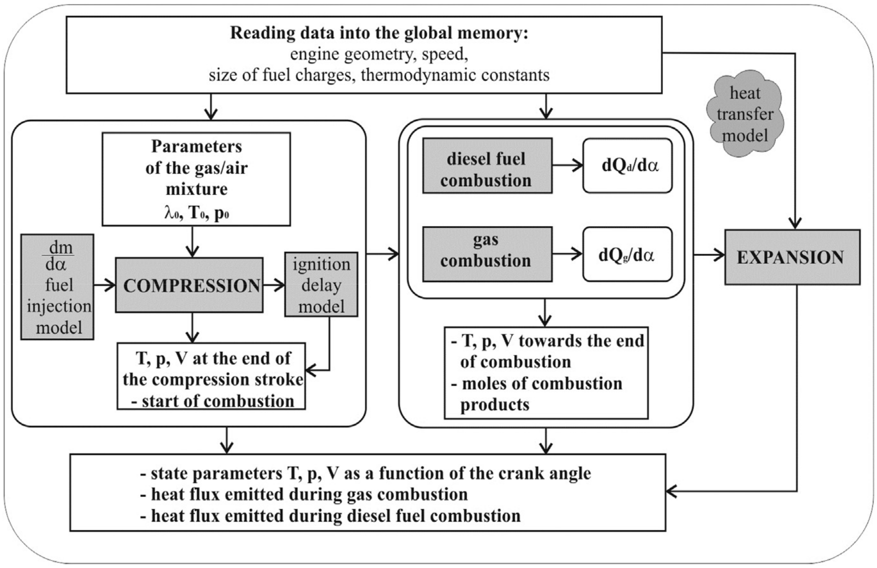

Fig. 1. Design structure of the simulation model.

for both fuel types is identical. A 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:

$$\frac{dQ_d(\alpha)}{d\alpha} = P_d\left\{1 - \exp\left[-6,908\cdot\left(\frac{\alpha-\alpha_{SOC}}{\Delta\alpha_d}\right)^{m_d+1}\right]\right\} \quad (2)$$

The model of natural gas 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 emission from combustion of individual components is determined by the kinetics of chemical reaction, with the following formula for the global reaction rate:

$$\frac{d[C_iH_{2i+2}]}{d\alpha} = A_i\exp\left(-\frac{Ea_i}{\overline{R}T}\right)\cdot[C_iH_{2i+2}]^{a_i}[O_2]^{b_i} \quad (3)$$

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

$$\frac{dQ_g(\alpha)}{d\alpha} = \sum\nolimits_{i=1}^{3}\frac{dQ_i(\alpha)}{d\alpha} = \sum\nolimits_{i=1}^{3}H_i\frac{dN_i(\alpha)}{d\alpha} \quad (4)$$

The constant values found in Eq. (3) are summarised for specific gases in Table 1.

The proposed model underwent a detailed, multi-step verification process. At the first stage equation (1) was validated to check its capability to capture ignition delay in a dual fuel engine (Piętak and Mikulski, 2011). It was concluded that the Assanis correlation can predict ignition delay with the accuracy of + - 1 CA up to gas/diesel Blend Ratios of 80% for all loads conditions. At the second phase the Wibe function was fitted for different, single fuel (diesel) engine operating points (Mikulski et al., 2016b). The result of this work was a globally (all operating points) fitted exponent coefficient of 0.4 and a simple and explicit correlation for diesel combustion duration ($\Delta\alpha$d) with respect to fuel consumption (G_d):

$$\Delta a_d = 5.52\cdot G_d + 18.57 \quad (5)$$

At the final phase, the validation of different global mechanisms of gaseous fuel combustion was performed, resulting in selection presented in Table 1. The whole model was validated for broad range of dual-fuel operating points and a methodology of tuning diesel combustion duration for the account of gaseous fuel blend ratio was established (Mikulski and Wierzbicki, 2016).

The macro-kinetics of the three combustion reactions employed in the model has enabled us to examine the impact of the gas composition on thermodynamic parameters and the combustion process. This is a considerable advantage over models operating on the set combustion function for studies on enabling operation of the engine on a possibly broad range of gaseous fuels.

2.2. Methodology of the research

As the starting point for the present study, the operating parameters of an ADCR engine were used, at a rotational speed of 3400 rpm and break torque of 100 Nm. The key technical parameters of the simulated unit are summarised in Table 2. For the present research 70% CH4/diesel blend ratio was taken as a reference experimental case. For this case diesel fuel (2.2 kg/h) was injected 6 CA before TDC (Top Dead Centre). Other relevant operating parameters case can be found in Table 3 (Reference case).

At the first step the model has been fitted for the reference case according to the procedure described in the previous chapter. From that the Wiebe function parameters were obtained: exponent coefficient $m_d = 0.4$ and diesel combustion duration $\Delta\alpha_d = 28$ CA. The comparison of the experimental and fitted model pressure trace was shown on Fig. 2. The visible difference in the expansion part are due to different combustion efficiencies. Due to model limitations (single zone nature), it was assumed in the study that the entire volume of gas supplied to the cylinder would participate in the reaction, and the combustion of the gaseous fuel was complete, while measured fuel conversion efficiency was 96% in this case.

In the presented research, the simulation parameter was the composition of gaseous fuel determined by the share of individual components. The start of ignition was calculated by Eq. (1). Gas composition influenced the ignition delay threw different specific

Table 1
Values of constants in Eq. (3).

i		A	Ea	a	b	H	H
		[−]	[MJ/mol]	[−]	[−]	[MJ/mol]	[MJ/kg]
1	CH_4	$8.3 \cdot 10^6$	0.125	0.3	1.3	802.5	50.03
2	C_2H_6	$1.1 \cdot 10^12$	0.125	0.1	1.65	1423.7	47.3
3	C_3H_8	$8.6 \cdot 10^{11}$	0.125	0.1	1.65	2045.3	46.38

Table 2
Technical details of an ADCR engine.

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

Table 3
Scope of presented results of simulation tests. G_{air}/G_g stand for air and gaseous fuel consumption, P − chemical Power of the introduced fuel mixture (gaseous + diesel).

Sample number	Gas composition			G_{air}	G_g	P_{d+g}
	% CH_4	%	Gas	[kg/h]		[kW]
1[a]	100	0	CO_2	264	4.3	84.66
2	50	50		260	8.6	
3	20	80		254	21.5	
4	10	90		250	42.9	
5[a]	100	0	C_2H_6	264	4.3	84.66
6	80	20				84.01
7	50	50				83.03
8	20	80				82.05
9[a]	100	0	C_3H_8	264	4.3	84.66
10	80	20				83.79
11	50	50				82.48
12	20	80				81.17
13[a]	100	0	C_2H_6/C_3H_8	264	4.3	84.66
14	80	10/10				83.9
15	50	25/25				82.75
16	20	40/40				81.63

[a] Reference case for witch the model was validated.

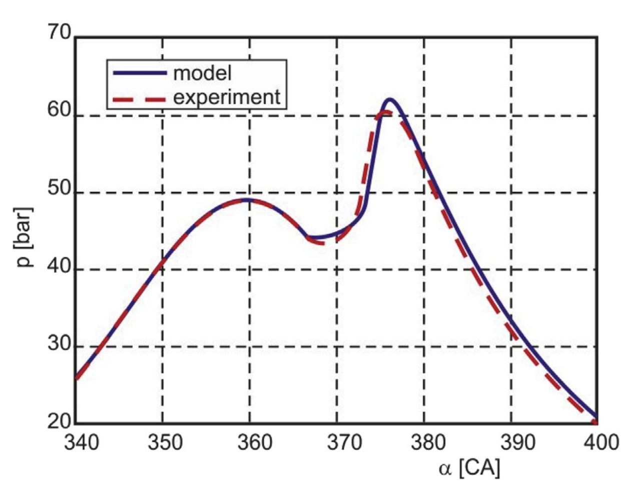

Fig. 2. Experimental and simulated pressure vs crank angle for the reference case.

heats (leading to different in-cylinder temperatures and pressures at start of ignition) and different diesel air/fuel ratios. It was assumed that diesel combustion duration is not inflicted by changing the gas composition. Dow this is not entirely true the simplification is justified since the possible differences within the value of $\Delta\alpha_d$ would not influence the combustion phasing of gaseous fuel significantly.

Firstly, gas composed of methane and carbon dioxide was tested to analyse how the addition of non-combustible gases impacts the combustion parameters. Due to the fact that high carbon dioxide content significantly impacts the volume of gas to be supplied to the cylinder to ensure the set energy share, the gas consumption was changed with the changes of composition thereof. Increasing the volume of supplied gas resulted in reduction of the volume of air aspirated by the engine, which was included in the calculation.

In other test runs, the impact of the addition of higher hydrocarbons (ethane and propane) to methane on the combustion process of individual phases was analysed. The tests assumed a constant level of gas and air consumption. In those test cases, change of the per cent weight of specific combustible components had minor impact on the calorific value of the supplied gaseous fuel and, at the same time, on the power supplied to the engine in the fuel (Table 1). Both ethane and propane have a higher molecular weight than methane, and their values for the heat of combustion per 1 kg of gaseous fuel are lower (Table 1) despite higher heat of combustion per 1 mol of gas.

The scope of conducted simulation tests and the operating parameters of the engine are provided in Table 3.

For all analysed points (Table 3), the model calculations were done in accordance with the diagram provided in Fig. 1, with the calculation step equivalent to 0.5 CA. The heat exchange model requires average temperatures of cylinder walls and average heat exchange coefficients of the charge in the cylinder throughout the cycle. The set initial values of these parameters had been included, and the programme in the scheme provided in Fig. 1 was launched in a loop leading to auto-correlation of the results. Following the calculations done by the basic programme, new average values were calculated, and in the next loop, the results were recalculated. The procedure was repeated until a self-consistency of results is obtained, i.e. to the moment when the delta of average values from the last two loops did not exceed the set accuracy limit.

3. Results and discussion

3.1. Impact of the addition of carbon dioxide to gaseous fuel on the combustion process

Analysing the impact of addition of CO_2 to methane on the combustion process, we have noticed a visible difference in the results of calculations of diesel fuel ignition delay (Figs. 3 and 10, Fig. 10). The addition of carbon dioxide increases the ignition delay for diesel fuel. The increase is correlated to the increase of gas concentration levels compared to aspirated air. This confirms the results obtained by Karim et al. (Motyl and Lisowski, 2008; Assanis et al., 2003), who have explained such a tendency by the different physico-chemical properties of the added gas, i.e. lower specific heat and thermal conductivity. This results in lower pressure and temperature values in the compression phase, which also have a considerable effect on the ignition delay process. This tendency can be observed in the plots of pressure and temperature (Fig. 3). The delta between the pressure obtained by the compression process in TDC for samples 1 and 4 is ca. 4 bar, was resulting in a delta of ca. 63 K in temperature.

Fig. 3 indicates that gaseous fuel or, more precisely, methane as its component, burns much faster than diesel fuel. With double the methane amount to diesel fuel amount, the combustion rate was more than doubled. In the discussed case, the combustion rate of gas does not significantly depend on its carbon dioxide content. In the range of up to 50% of methane content in the analysed mixture, the impact of impurities in gaseous fuel on engine operation were virtually negligible, according to the simulation results. This arises from the fact that the amount of the supplied gas was still very low compared to air volume and has no significant impact on changes of oxygen concentration levels in the mixture. On the other hand, the volume of air supplied to the cylinder remained unchanged for all test runs. For these reasons, the gas combustion process is identical, with up to 50% addition of impurities. With a carbon dioxide per cent weight of 20% of the total gas volume, a slight gaseous fuel ignition delay was observed (by 2 CA) compared to sample 1, with almost negligible ignition delay of diesel fuel (0.4 CA). This impacts the combustion process, generating pressure and temperature lower by ca. 5% in this phase of the cycle. According to model calculations, problems with engine operation may occur with per cent weight of combustible components lower than 20%. Reducing the methane content below 10% results in lack of ignition due to a considerably delayed combustion of diesel fuel, caused by an increase of the gas concentration level. The presence of impurities in gaseous fuel becomes crucial for engine operation quality under high loads when the content of combustible gaseous components considerably increases (compared to the simulated operating point). The fivefold increase of the amount of supplied gas, necessary to maintain the operating parameters at the set level, must in

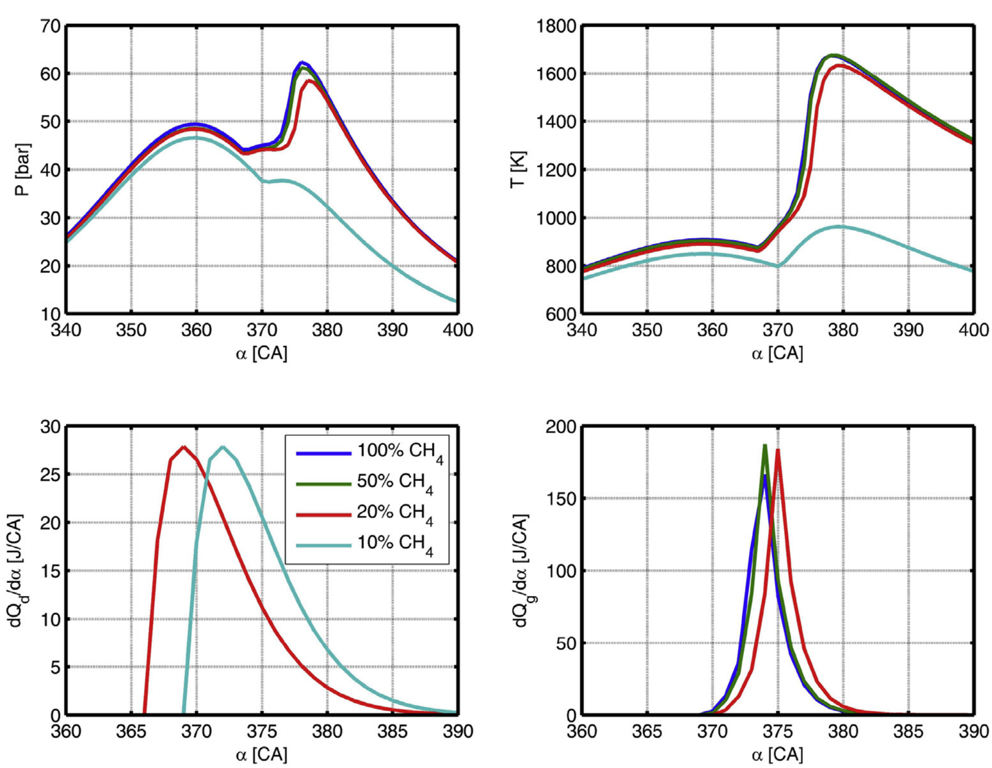

Fig. 3. Pressure and temperature values in the cylinder and heat release rate from the combustion of diesel fuel and gas. Results of simulations for samples 1–4. Different methane content with respect to carbon dioxide.

this case result in a significant reduction of the oxygen concentration level in the analysed mixture and, consequently, slow down the processes of ignition and combustion of specific fuels. The engine is still able to operate in dual-fuel mode, but it is necessary to provide the appropriate characteristics of diesel fuel combustion.

3.2. Impact of the addition of ethane to gaseous fuel on the combustion process

Next, the impact of the addition of ethane to methane on the combustion process in a dual-fuel engine was analysed. Results for simulation runs 5–8 are provided in Fig. 4. Additionally, Fig. 5 presents the combustion process of both combustible components of the gas (methane and ethane).

For most gaseous fuels, a delay of start of combustion is observed with the increase of their concentration levels in the charge (Karim, 2003). 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. 10). This reverse effect of ethane has also been observed by other researchers. Research (Karim, 2003) 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. 4 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. 3 CA after the 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 length of gaseous fuel combustion was slightly higher for sample 6, as compared to the base sample (5). For samples 7 and 8, 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 even by 33% (for a 50% addition of methane) significantly improves the performance of 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 methane may, at the same time, increase the emissions of nitrogen oxide.

Important data on the combustion process in the engine powered by the mixture of diesel fuel, methane and ethane are provided by the analysis of the heat release rate from combustion of both gaseous combustible components (Fig. 5).

Fig. 4 indicates 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.

3.3. Impact of the addition of propane to gaseous fuel on the combustion process

The addition of propane results in a minor increase of the diesel

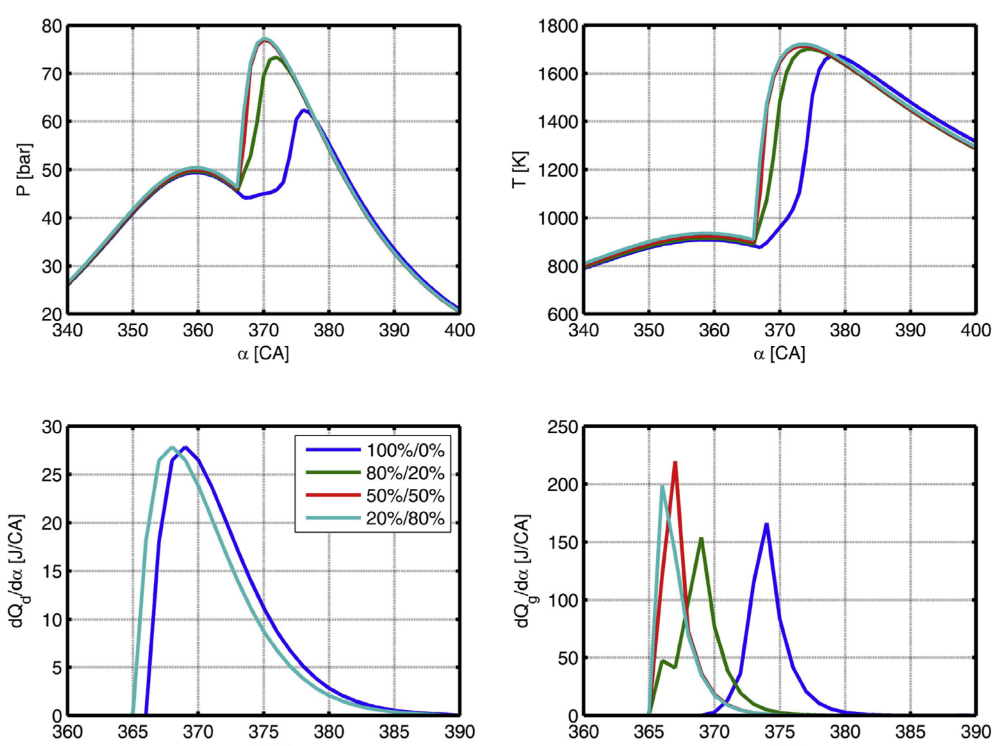

Fig. 4. Pressure and temperature values in the cylinder and heat release rate from the combustion of diesel fuel and gas. Results of simulations for samples 5–8. Different methane content with respect to ethane (C_2H_6).

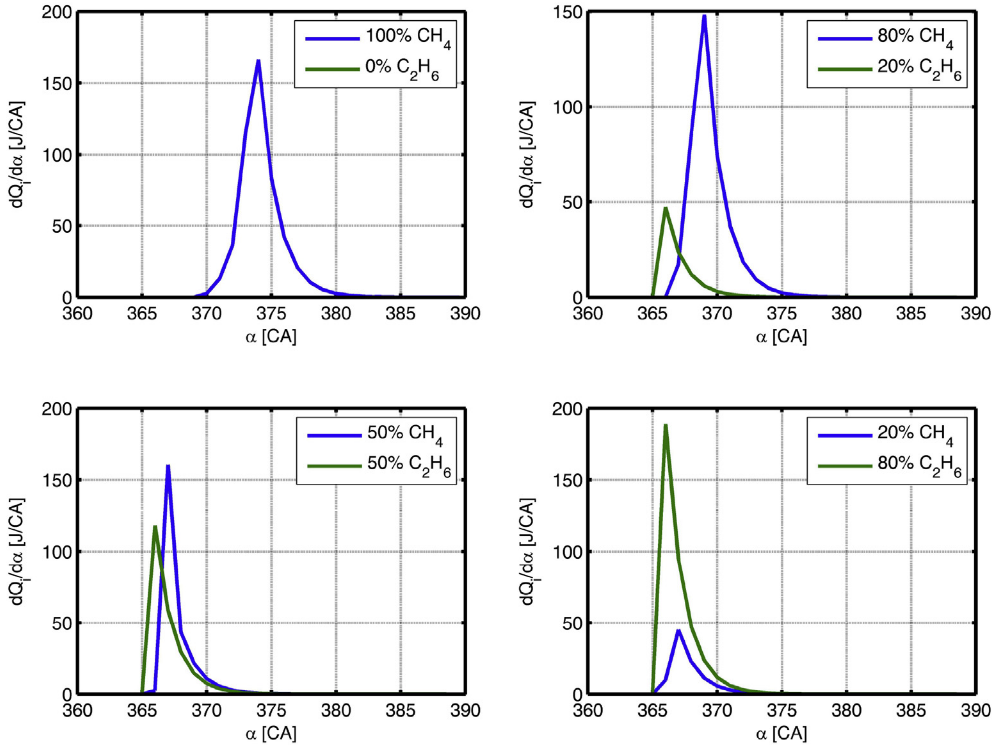

Fig. 5. Heat release rates from combustion of individual combustible components of the gas (methane and ethane). Results of simulations for samples 5–8. Different methane content with respect to ethane (C_2H_6).

fuel ignition delay, as shown by the data in Fig. 10. 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) did not exceed 1 CA. This had some impact on the pressure and temperature in the cylinder – Fig. 6.

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 previously observed ignition generated a higher maximum temperature and pressure than for pure 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 (sample 12–80% propane) generated smaller increments of pressure and temperature than sample 11 (50% propane).

As has been observed (Fig. 7), the combustion rate of propane in a dual-fuel engine is similar to methane, with the former generating higher maximum heat release rates due to its higher calorific value. The start of ignition of propane is identical to that of diesel fuel, and even a small addition of this gas may accelerate the combustion of methane.

3.4. *Impact of addition of C_2H_6 and C_3H_8 to gaseous fuel on the combustion process*

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. 10). 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. 8).

Similarly to samples analysed previously, quick ignition of the fuel generates higher increments of pressure and temperature in the cylinder in combustion phase. Fig. 9 shows that with equal volumes of methane and ethane, the burn rates of both fuels are virtually identical. This can be explained by similar form of equations describing the kinetics of reaction of both fuels (2). The burn rate differs only by coefficient A_n. At the same time, low value of this coefficient for propane is compensated by higher calorific value. As a result, the delta of amplitude of heat emission rates of individual gases is small.

3.5. *Comparative analysis of combustion parameters in a dual-fuel compression-ignition engine*

Further on, the synthetic parameters describing the combustion process for all simulation points were compared. Below, the results are provided for calculations of diesel fuel ignition delay, maximum combustion pressure and temperature, maximum heat release rate from gaseous fuel combustion and indicated efficiency. The effect of enhancement of methane with various gases on ignition delay has been discussed in detail in previous sections. A summary of results of calculations of ignition delay for all test samples (Fig. 10) would, however, contribute to the coherence of the study.

According to the model calculations, the addition of higher hydrocarbons to methane has increased the maximum combustion pressure (Fig. 11). It should also be borne in mind that an addition of ethane or propane has slightly reduced the calorific value of the gaseous fuel, and its amount has been set constant for simulations run with these gases. At the same time, the addition of ethane generated the highest increase of combustion pressure. The

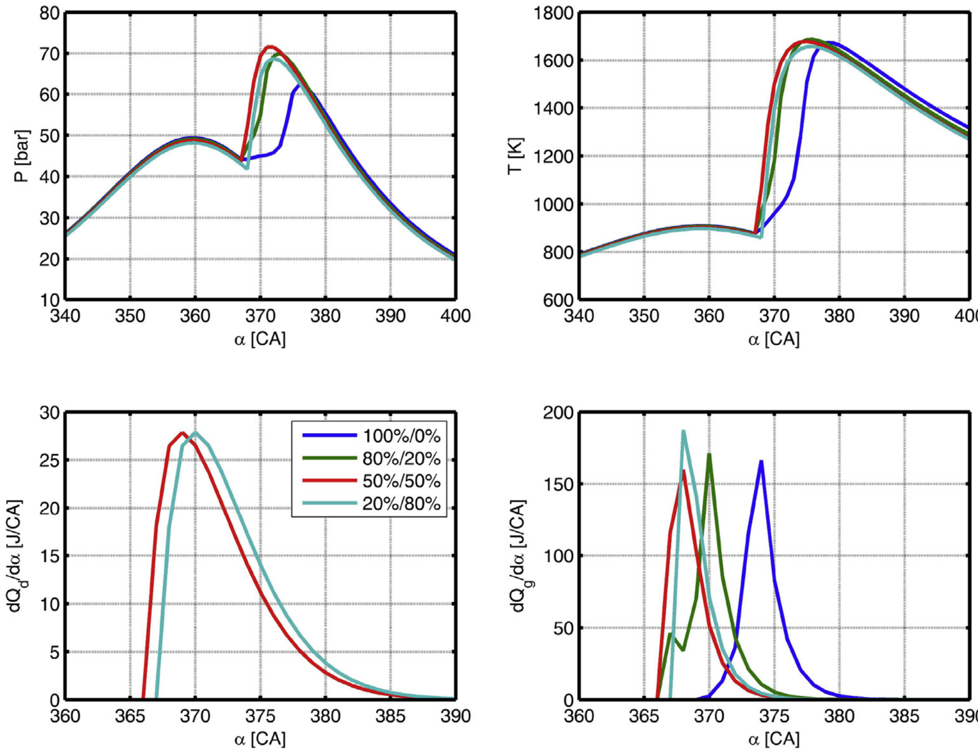

Fig. 6. Pressure and temperature values in the cylinder and rate of heat release rate from the combustion of diesel fuel and gas. Results of simulations for samples 9–12. Different methane content with respect to propane (C_3H_8).

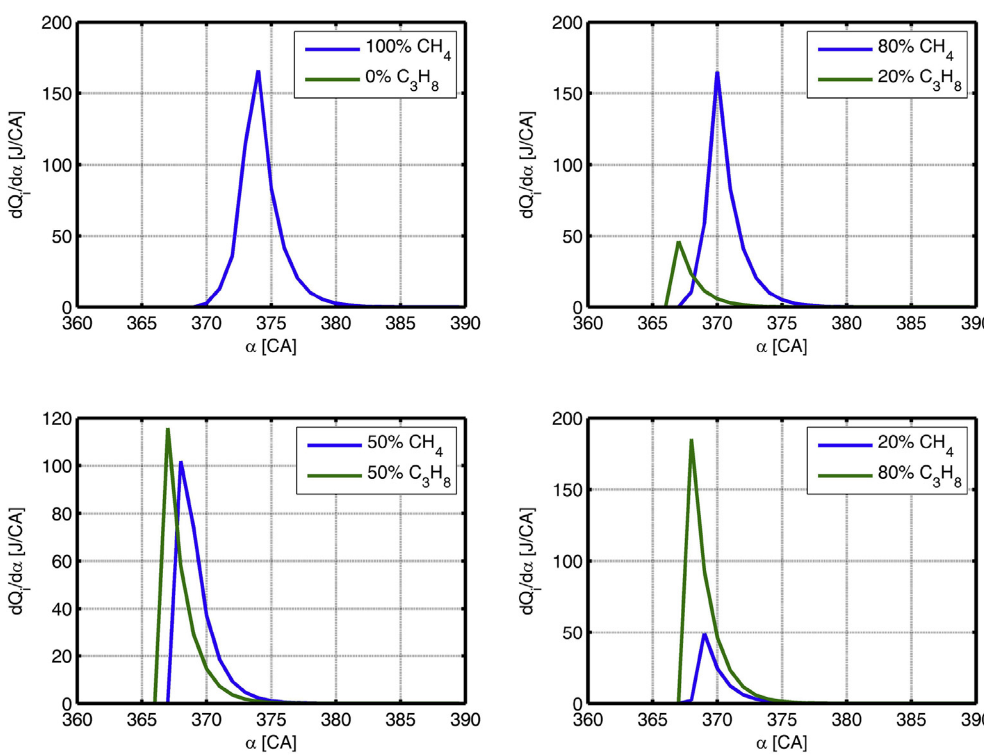

Fig. 7. Heat release rates from the combustion of individual combustible components of the gas (methane and ethane). Results of simulation for samples 9–12. Different methane content with respect to propane (C_3H_8).

addition of carbon dioxide has no significant impact on combustion pressure, provided that the amount of methane in the mixture is constant. Obviously, this effect is observable in mixtures with a carbon dioxide content up to 80%. Higher levels would cause major

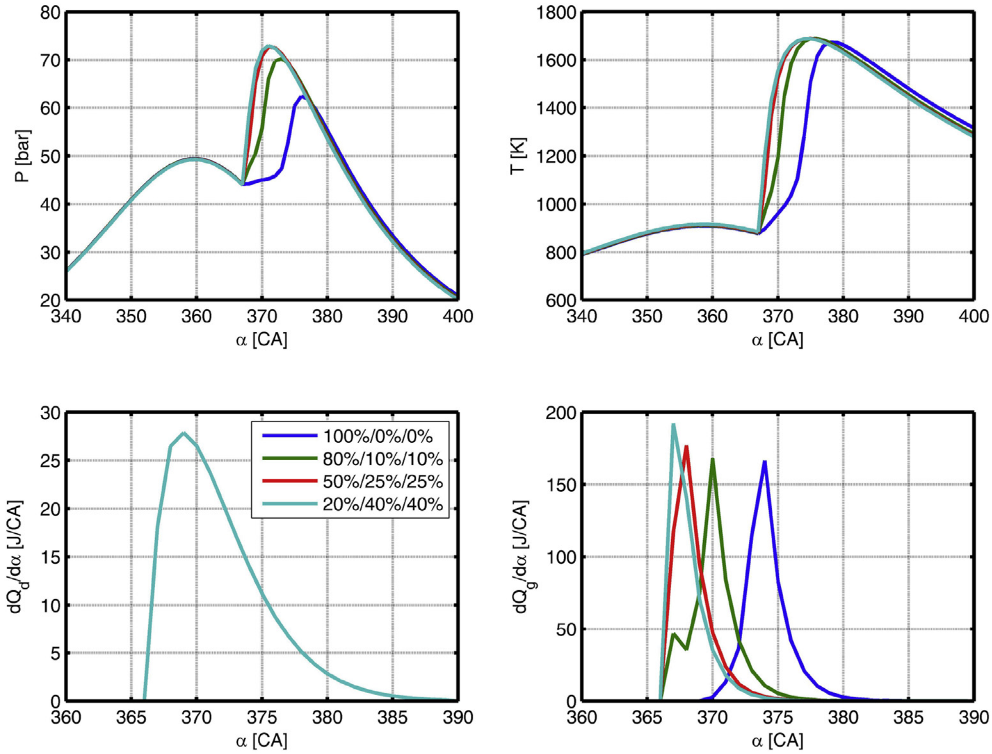

Fig. 8. Pressure and temperature values in the cylinder, and rate of heat release from combustion of diesel fuel and gas. Results of simulations for samples 13–16. Different methane content with respect to ethane (C_2H_6) and propane (C_3H_8).

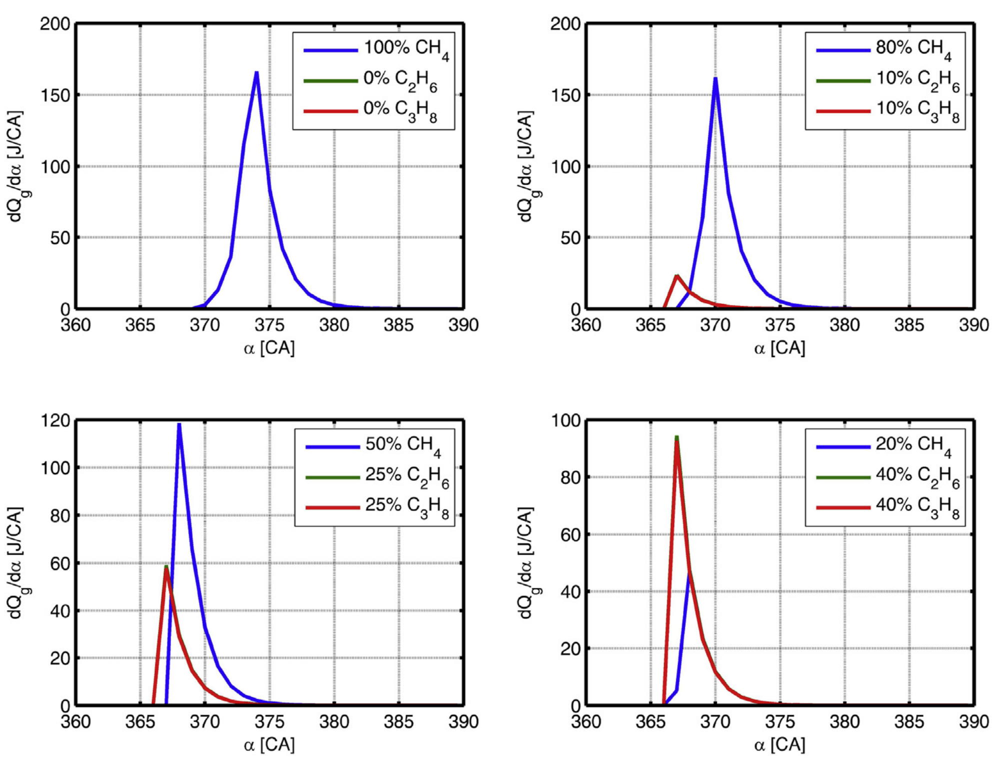

Fig. 9. Heat release rates from combustion of individual combustible components of the gas (methane and ethane). Results of simulation for samples 13–16. Different methane content with respect to ethane (C_2H_6) and propane (C_3H_8).

changes to this parameter due to increased diesel fuel ignition delay. Interestingly, the increase of maximum combustion pressure in the tests has not been strongly correlated to the increase of maximum combustion temperature, and particularly with the

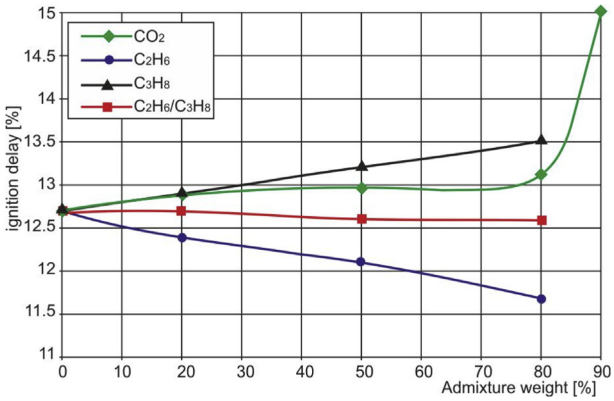

Fig. 10. Angle of ignition delay of diesel fuel, depending on the type and per cent weight of an additive in gaseous fuel.

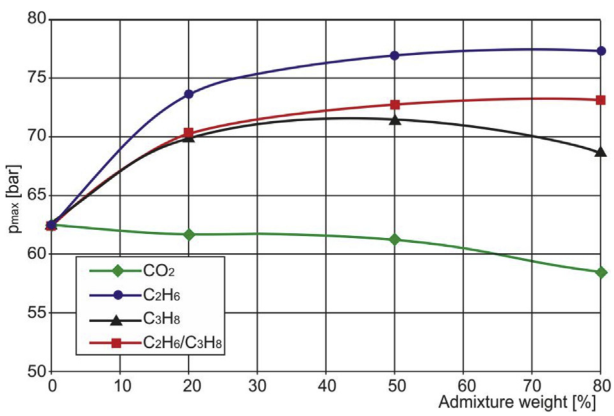

Fig. 11. Maximum combustion pressure in the engine cylinder, depending on the type and per cent weight of an additive in gaseous fuel.

obtained values of maximum heat from the combustion of gaseous fuel.

In the case of maximum combustion temperature (Fig. 12), increasing the per cent weight of ethane has resulted in a consistent increase of this parameter's value. Combustion of a mixture with 80% content of this gas has generated a temperature that has been higher by 46° than in the combustion of pure methane. In the case of larger additions of propane, the maximum temperature of the combustion process has been reduced, despite the initial upward tendency observed for lower concentration levels of this gas. The downward tendency is a result of the delayed diesel fuel ignition (and thus the gaseous fuel ignition), arising from the presence of propane in the cylinder. As evident from Fig. 12, the combustion of propane can generate lower combustion temperatures than the combustion of propane, which allows for a reduction of NO_x, at the same time maintaining better performance arising from higher combustion pressure (Fig. 13).

In the case of the maximum combustion rate of gaseous fuel (Fig. 12), a clear upward tendency for individual test samples is not observed. This arises from the previously discussed overlapping of specific phases of combustion of methane and the additive gas. The combustion rates of individual fractions were discussed in previous sections. However, analysis of Fig. 13 indicates that an addition of methane causes a reverse effect in terms of impact on the maximum heat release rate than an addition of propane. The correlation here is clearly visible from the analysis of combustion of even admixtures of these gases. The maximum heat release rate rises in a stable manner with the increase of the per cent weight of the ethane/propane mixture.

The addition of higher hydrocarbons improves the indicated performance of the engine (Fig. 14); a 20% addition of ethane resulted in performance improvement by 6%. Further increase of the per cent weight of this gas resulted in considerably smaller improvements of performance. The delta between the performance of an engine powered by a 20% mixture of ethane in methane and the 80% methane mixture was 3%. The addition of propane yielded smaller performance increments. For extreme admixtures of propane, a drop of performance can be expected due to the significant ignition delay of the mixture. However, the effect can be neutralised (to a small extent) by appropriate setting of injection of the pilot dose of diesel fuel. Increased concentration levels of carbon dioxide do not significantly impact the engine performance for a per cent weight up to 50%. Higher concentrations can deteriorate performance due to obstructed ignition of gaseous and diesel fuel. The performance deterioration should be expected to be more visible for higher loads and per cent weight of gaseous fuels. In such cases, the volume of supplied gas has significant impact on the reduction of the volume of supplied air.

4. Conclusions

This study has shown that an increase of carbon dioxide levels in gaseous fuel affects the combustion process only if the content of carbon dioxide is high (above 80%). Problems with the ignition of gaseous fuel occur due to the considerable impact of carbon dioxide

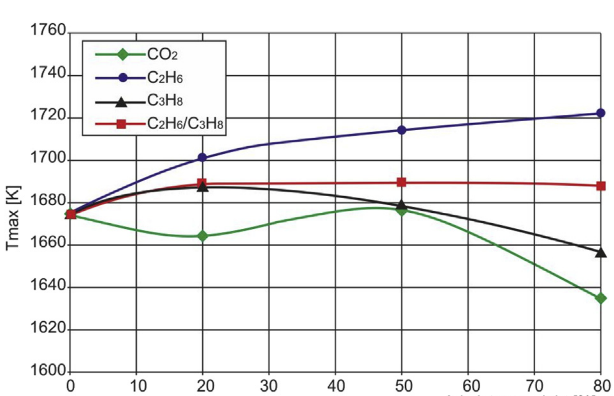

Fig. 12. Maximum combustion temperature in the engine cylinder, depending on the type and per cent weight of an additive in gaseous fuel.

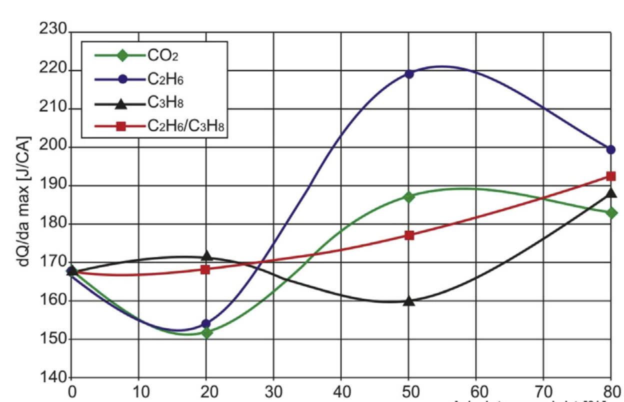

Fig. 13. Maximum heat release rate from combustion, depending on the type and per cent weight of an additive in gaseous fuel.

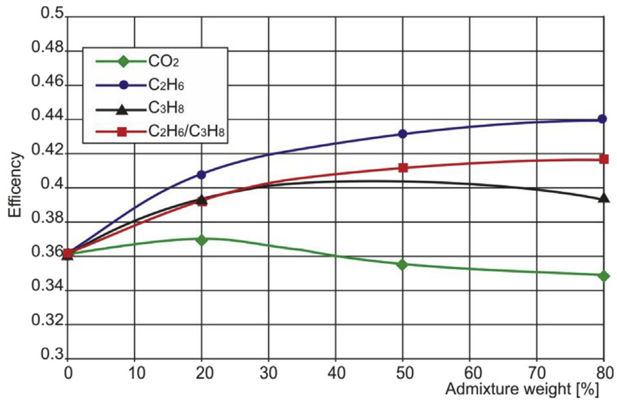

Fig. 14. Indicated efficiency of the engine, depending on the type and per cent weight of an additive in gaseous fuel.

on the increase of ignition delay of the initial dose of diesel fuel. The combustion of low calorific value gases with low methane content is possible, but requires appropriate control of the diesel fuel combustion process.

The presence of higher hydrocarbons in gaseous fuels shifts the maximum heat release rate towards TDC. This is caused by the fact that both ethane and propane burn faster than methane. Similarly, the ignition of both fuels is more reliable at lower temperatures. In the case of additions, the basic dose of methane also ignites faster due to high momentary heat increment, caused by prior combustion of higher hydrocarbons. This opens up the possibility of using small additions of higher hydrocarbons to improve the properties of low calorific value gases, particularly for high engine loads. The calculations in the model indicate that the combustion process of mixtures containing higher hydrocarbons is also more controlled. Due to different combustion points of methane, ethane and propane, the total combustion process is slightly extended. The same imparity of ignition points allows us to conclude that the volume of un-combusted CH_4 may be significantly reduced in this case, especially in low concentration levels, which would have a positive effect on emissions and the economical aspects of the engine operation.

Adding higher hydrocarbons to the gas improves the performance of the engine. Considerable benefits in this respect can be obtained even with small (below 20%) additions of ethane or propane. Furthermore, applying symmetrical additions of both fuels enables maintaining a fixed ignition point, regardless of the volume of added gas.

Thus, the study has proven that the composition of the gas has a significant impact on the parameters of operation of a dual-fuel engine, in particular for attempts to minimize the initial dose of diesel fuel. To maximize engine performance, gas composition should be included in control algorithms. For this purpose, it will be necessary to analyse in detail the impact of the mutual effects of the combustion processes of specific components in a dual-fuel engine. Simulation tests offer a wide range of possibilities in this respect.

Moreover, as has been shown, it may be interesting to study the impact of gas additions on the combustion process for various engine loads, especially for loads approaching maximum. For these points, the impact of additions should be most visible due to the high concentration of fuel in the engine cylinder as compared to air.

The model described herein has certain limitations. The crucial limitation is its inability to predict the portion of the gas and air mixture that will be covered by the propagation zone of the flame of diesel fuel combustion. This is important, because it determines the amount of gas participating in the reaction. The accuracy of results can be additionally improved by using the dimensional model and introducing more complex kinetic mechanisms of gas combustion. Therefore, further works on development of simulation tools is justified.

References

Alla, G.H.A., Soliman, H.A., Badr, O.A., Rabbo, F.M., 2001. Combustion Quasi-zone predictive model for dual fuel engines. Energy Convers. Manag. 42, 1477–1498. http://dx.doi.org/10.1016/S0196-8904(00)00143-6.

Assanis, D.N., Filipi, Z.S., Fiveland, S.B., Syrimis, M., 2003. A predictive ignition delay correlation under steady-state and transient operation of a direct injection diesel engine. J. Eng. Gas. Turbines Power 125 (2), 450–457. http://dx.doi.org/10.1115/1.1563238.

Azimov, U., Tomita, E., Kawahara, N., Dol, S.S., 2012. Combustion characteristics of syngas and natural gas in micro-pilot ignited dual-fuel engine. World Acad. Sci. Eng. Technol. 72, 12–22.

Banapurmath, N.R., Tewari, P.G., Yaliwal, V.S., Kambalimath, S., Basavarajappa, Y.H., 2009. Combustion characteristics of a 4-stroke CI engine operated on Honge oil, Neem and Rice Bran oils when directly injected and dual fuelled with producer gas induction. Renew. Energy 34, 1877–1884. http://dx.doi.org/10.1016/j.renene.2008.12.031.

Bari, S., 1996. Effect of carbon dioxide on the performance of biogas/diesel dual-fuel engine. Renew. Energy 9 (1–4), 1007–1010. http://dx.doi.org/10.1016/0960-1481(96)88450-3.

Barik, D., Murugan, S., 2014. Investigation on combustion performance and emission characteristics of a DI (direct injection) diesel engine fueled with biogasdiesel in dual fuel mode. Energy 72, 760–771. http://dx.doi.org/10.1016/j.energy.2014.05.106.

Bora, B.J., Saha, U.K., Chatterjee, S., Verr, V., 2014. Effect of compression ratio on performance, combustion and emission characteristics of a dual fuel diesel engine run on raw biogas. Energy Convers. Manag. 87, 1000–1009. http://dx.doi.org/10.1016/j.enconman.2014.07.080.

Cacua, K., Amell, A., Cadavid, F., 2012. Effects of oxygen enriched air on the operation and performance of a diesel-biogas dual fuel engine. Biomass Bioenergy 45, 159–167. http://dx.doi.org/10.1016/j.biombioe.2012.06.003.

Chandra, R., Vijay, V.K., Subbarao, P.M.V., Khura, T.K., 2011. Performance evaluation of a constant speed IC engine on CNG, methane enriched biogas and biogas. Appl. Energy 88, 3969–3977. http://dx.doi.org/10.1016/j.apenergy.2011.04.032.

Czerwiński, J., Comte, P., 2001. Influences of gas quality on a natural gas engine. In: SAE Technical Paper. http://dx.doi.org/10.4271/2001-01-1194, 2001-01-1194.

d'Ambrosio, S., Finesso, R., Fu, L., Mittica, A., Spessa, E., 2014. A control-oriented real-time semi-empirical model for the prediction of NOx emissions in diesel engines. Appl. Energy 130, 265–279. http://dx.doi.org/10.1016/j.apenergy.2014.05.046.

Egüz, U., Maes, N.C.J., Leermakers, C.A.J., Somers, L.M.T., De Goey, L.P.H., 2013. Predicting auto-ignition characteristics of RCCI combustion using a multi-zone model. Int. J. Automot. Technol. 14 (5), 693–699.

Egüz, U., Leermakers, C.A., Somers, L.M., de Goey, L.P., 2013. Premixed charge compression ignition combustion modelling with a multi-zone approach including inter-zonal mixing. Proc. Inst. Mech. Eng. Part D J. Automob. Eng. 227 (9), 1313–1324.

Eichmeier, J., Reitz, R., Rutland, C.A., 2014. Zero-dimensional phenomenological model for RCCI combustion using reaction kinetics. SAE Int. J. Engines 7 (1), 106–119.

Fischer, M., Jiang, X., 2014. An assessment of chemical kinetics for bio-syngas combustion. Fuel 137, 293–305. http://dx.doi.org/10.1016/j.fuel.2014.07.081.

Fomin, P.A., Trotsyuk, A.V., Vasil'ev, A.A., 2014. Approximate model of chemical reaction kinetics for detonation processes in mixture of CH4 with air combustion. Sci. Technol. 186 (10–11), 1716–1735. http://dx.doi.org/10.1080/00102202.2014.935643.

Henham, A., Makkar, M., 1998. Combustion of simulated biogas in a dual-fuel diesel engine. Energy Convers. Manag. 39 (16–18), 2001–2009. http://dx.doi.org/10.1016/S0196-8904(98)00071-5.

Kakaee, A.H., Paykani, A., Ghajar, M., 2014. The influence of fuel composition on the combustion and emission characteristics of natural gas fueled engines. Renew. Sustain. Energy Rev. 38, 64–78. http://dx.doi.org/10.1016/j.rser.2014.05.080.

Karim, G.A., 2003. Combustion in gas fuelled compression ignition engines of the dual fuel type. J. Eng. Gas Turbines Power 125 (3), 827–836. http://dx.doi.org/10.1115/1.1581894.

Korakianitis, T., Namasivayam, A.M., Crookes, R.J., 2011. Diesel and rapeseed methyl ester (RME) pilot fuels for hydrogen and natural gas dual-fuel combustion in compression–ignition engines. Fuel 90, 2384–2395. http://dx.doi.org/10.1016/j.fuel.2011.03.005.

Koszalka, G., 2014. Model of operational changes in the combustion chamber tightness of a diesel engine. Eksploat. Niezawodn. Maint. Reliab. 16 (1), 133–139.

Lim, C., Kim, D., Song, C., Kim, J., Han, J., Cha, J.S., 2015. Performance and emission characteristics of a vehicle fueled with enriched biogas and natural gases. Appl. Energy 139, 17–29. http://dx.doi.org/10.1016/j.apenergy.2014.10.084.

Liu, J., Yang, F., Wang, H., Ouyang, M., Hao, S., 2013. Effects of pilot fuel quantity on the emissions characteristics of a CNG/diesel dual fuel engine with optimized pilot injection timing. Appl. Energy 110, 201–206. http://dx.doi.org/10.1016/

j.apenergy.2013.03.024.
Luijten, C.C.M., Kerkhof, E., 2011. Jatropha oil and biogas in a dual fuel CI engine for rural electrification. Energy Convers. Manag. 52, 1426–1438. http://dx.doi.org/10.1016/j.enconman.2010.10.005.
Maizonnasse, M., Plante, J.S., Oha, D., Laflamm, C.B., 2013. Investigation of the degradation of a low-cost untreated biogas engine using preheated biogas with phase separation for electric power generation. Renew. Energy 55, 501–513. http://dx.doi.org/10.1016/j.renene.2013.01.006.
Mansour, C., Bounif, A., Aris, A., Gaillard, F., 2001. Gas–diesel (dual-fuel) modeling in diesel engine environment. Int. J. Therm. Sci. 40 (4), 409–424. http://dx.doi.org/10.1016/S1290-0729(01)01223-6.
Mikulski, M., 2014. Construction of a Two-phase Combustion Model in a Multifuel, Compression-ignition Engine. Dissertation. UWM, Olsztyn.
Mikulski, M., Wierzbicki, S., 2016. Validation of a zero-dimensional, 2-phase combustion model for dual-fuel compression ignition engine simulation. Therm. Sci., OnLine-First (00) (in print).
Mikulski, M., Wierzbicki, S., Piętak, A., 2015. Numerical studies on controlling gaseous fuel combustion by managing the combustion process of diesel pilot dose in a dual-fuel engine. Chem. Process Eng. Inzynieria Chem. Proces. 36 (2), 225–238. http://dx.doi.org/10.1515/cpe-2015-0015.
Mikulski, M., Wierzbicki, S., Śmieja, M., Matijošius, J., 2015. Effect of CNG in a fuel dose on compression-ignition engine's combustion process. Transport 30 (2), 162–171. http://dx.doi.org/10.3846/16484142.2015.1045938.
Mikulski, M., Wierzbicki, S., Piętak, A., 2015. The multi-phase, zero-dimensional, computational model of a multi-fuel CI engine fueled with gaseous fuel with divided injection of liquid fuel. Eksploat. Niezawodn. Maint. Reliab. 17 (1), 42–48.
Mikulski, M., Wierzbicki, S., Duda, K., 2016. Performance and emissions of a CRDI diesel engine fuelled with swine lard methyl esters-diesel mixture. Fuel 164, 206–219. http://dx.doi.org/10.1016/j.fuel.2015.09.083.
Mikulski, M., Wierzbicki, S., Piętak, A., 2016. Verification of a 2–phase, zero-dimensional model of a multifuel compression-ignition engine in single fuel operation. Appl. Mech. Mater. 817, 47–56. http://dx.doi.org/10.4028/. www.scientific.net/AMM.817.47.
Miyata, R., Fukushima, N., Naka, Y., Shimura, M., Tanahashi, M., Miyauchi, T., 2015. Direct numerical simulation of micro combustion in a narrow circular channel with a detailed kinetic mechanism. Proc. Combust. Inst. 35 (3), 3421–3427. http://dx.doi.org/10.1016/j.proci.2014.07.057.
Motyl, K., Lisowski, M., 2008. Wpływ temperatury początkowej i składu mieszaniny palnej na pracę silnika HCCI zasilanego biogazem. Inżynieria Rol. 1 (99), 311–317.
Mustafi, N.N., Raine, R.R., Verhelst, S., 2013. Combustion and emissions characteristics of a dual fuel engine operated on alternative gaseous fuels. Fuel 109, 669–678. http://dx.doi.org/10.1016/j.fuel.2013.03.007.
Nieman, D., Dempsey, A., Reitz, R., 2012. Heavy-duty RCCI operation using natural gas and diesel. SAE Int. J. Engines 5 (2), 270–285.
Papagiannakis, R.G., Hountalas, D.T., Rakopoulos, C.D., 2007. Theoretical study of the effects of pilot fuel quantity and its injection timing on the performance and emissions of a dual fuel diesel engine. Energy Convers. Manag. 48 (11), 2951–2961. http://dx.doi.org/10.1016/j.enconman.2007.07.003.
Pawlaczyk, A., Gosiewski, K., 2009. Estimation of kinetic parameters for the homogeneous oxidation of lean methane – air mixtures based on experimental temperature profiles. Chem. Process Eng. 30, 139–147.
Piętak, A., Mikulski, M., 2011. On the modeling of pilot dose ignition delay in a dual-fuel, self ignition engine. Siln. Spalinowe 50, 1–16.
Puduppakkam, K., Liang, L., Naik, C., Meeks, E., Kokjohn, S.L., Reitz, R.D., 2011. Use of detailed kinetics and advanced chemistry-solution techniques in cfd to investigate dual-fuel engine concepts. SAE Int. J. Engines 4 (1), 1127–1149.
Raj, R.E., Sekhar, S.J., Daniel, B.S.S., 2013. Influence of dual fuel twin injection on diesel engine combustion and emission characteristics. Adv. Mater. Res. 768, 206–212. http://dx.doi.org/10.4028/. www.scientific.net/AMR.768.206.
Roy, S., Banerjee, R., Das, A.K., Bose, P.K., 2014. Development of an ANN based system identification tool to estimate the performance-emission characteristics of a CRDI assisted CNG dual fuel diesel engine. J. Nat. Gas Sci. Eng. 21, 147–158. http://dx.doi.org/10.1016/j.jngse.2014.08.002.
Roy, S., Ghosh, A., Das, A.K., Banerjee, R., 2015. Development and validation of a GEP model to predict the performance and exhaust emission parameters of a CRDI assisted single cylinder diesel engine coupled with EGR. Appl. Energy 140, 52–64. http://dx.doi.org/10.1016/j.apenergy.2014.11.065.
Ryu, K., 2013. Effects of pilot injection timing on the combustion and emissions characteristics in a diesel engine using biodiesel–CNG dual fuel. Appl. Energy 111, 721–730. http://dx.doi.org/10.1016/j.apenergy.2013.05.046.
Sahoo, B.B., Sahoo, N., Saha, U.K., 2009. Effect of engine parameters and type of gaseous fuel on the performance of dual-fuel gas diesel engines – a critical review. Renew. Sustain. Energy Rev. 13 (6–7), 1151–1184. http://dx.doi.org/10.1016/j.rser.2008.08.003.
Wierzbicki, S., Śmieja, M., 2014. Visualization of the parameters and changes of signals controlling the operation of common rail injectors. Solid State Phenom. Mechatron. Syst. Mech. Mater. II 210, 136–141. http://dx.doi.org/10.4028/. www.scientific.net/SSP.210.136.
Yang, Z., Chu, C., Wang, L., Huang, Y., 2015. Effects of H2 addition on combustion and exhaust emissions in a diesel engine. Fuel 139 (1), 190–197. http://dx.doi.org/10.1016/j.fuel.2014.08.057.
Yusaf, T.F., Buttsworth, D.R., Saleh, K.H., Yousif, B.F., 2010. CNG–diesel engine performance and exhaust emission analysis with the aid of artificial neural network. Appl. Energy 87, 1661–1669. http://dx.doi.org/10.1016/j.apenergy.2009.10.009.