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Chemical and Process Engineering 2015, 36 (2), 225-238
DOI: 10.1515/cpe-2015-0015

NUMERICAL STUDIES ON CONTROLLING GASEOUS FUEL COMBUSTION BY MANAGING THE COMBUSTION PROCESS OF DIESEL PILOT DOSE IN A DUAL-FUEL ENGINE

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Protection of the environment and counteracting global warming require finding alternative sources of energy. One of the methods of generating energy from environmentally friendly sources is increasing the share of gaseous fuels in the total energy balance. The use of these fuels in compression-ignition (CI) engines is difficult due to their relatively high auto-ignition temperature. One solution for using these fuels in CI engines is operating in a dual-fuel mode, where the air and gas mixture is ignited with a liquid fuel dose. In this method, a series of relatively complex chemical processes occur in the engine's combustion chamber, related to the combustion of individual fuel fractions that interact with one another. Analysis of combustion of specific fuels in this type of fuel injection to the engine is difficult due to the fact that combustion of both fuel fractions takes place simultaneously. Simulation experiments can be used to analyse the impact of diesel fuel combustion on gaseous fuel combustion. In this paper, we discuss the results of simulation tests of combustion, based on the proprietary multiphase model of a dual-fuel engine. The results obtained from the simulation allow for analysis of the combustion process of individual fuels separately, which expands the knowledge obtained from experimental tests on the engine.

Keywords: dual-fuel engine, gaseous fuel, combustion process, mathematical model

1. INTRODUCTION

Counteracting global warming and environmental pollution are the two main challenges for mankind today. To face these challenges, international organisations pass various resolutions to limit civilization's impact on the natural environment. At the same time, numerous initiatives have been introduced, related to development of the power industry based on the renewable energy sources. The applicable regulations on the use of renewable fuels, e.g. Directive 2009/28/E of the European Parliament and that of the Council of 23 April 2009 on the promotion of electricity produced from the renewable energy sources in the internal electricity market, stipulate that by the year 2020, at least 20% of energy must be generated from renewable sources.

One method of reducing emissions of greenhouse gases is the use of gaseous fuels, both fossil fuels, such as natural gas, and alternative fuels, such as biogas. Biogas is of particular importance here; large volumes of biogas are generated by natural reactions occurring in waste landfills, waste treatment plants or animal farms. Biogas contains a large amount of methane, depending on the raw material and

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the method of its generation (45-75%); However, it should be pointed out that methane released into the atmosphere from biogas is 20 times more harmful for the ozone layer than carbon dioxide (Bade Shrestha and Narayanan 2008). It should also be noted that technologies are in development for cleaning biogas to so-called biomethane, containing over 90% of methane. Therefore, it is advisable to search for new ways of converting energy contained in such fuels into electric power or fuel for combustion engines (Börjesson and Berglund, 2006; Budzianowski, 2012).

Due to its properties, methane can be used to directly power engines with spark ignition. Using this type of fuel for a compression-ignition engine requires an external source of ignition. The reason for this is the relatively high methane auto-ignition point (above 900 K). At the moment, the use of this type of fuel in dual-fuel compression-ignition engines is being intensively researched (Korakianitis et al., 2011; Szczurowski et al., 2014).

In this solution, during the compression stroke, a mixture of air and gaseous fuel is drawn in (similarly to spark-ignition engines), and at the end of the compression stroke, a dose of liquid fuel is injected into the cylinder to cause auto-ignition, initiating the combustion of gaseous fuel. This method of fuelling, requires a different approach to the control of the engine power system (Korakianitis et al., 2011; Yoon and Lee, 2011).

1.1. The combustion process in dual-fuel compression-ignition engines

Currently, fuel injection systems for compression-ignition engines are commonly controlled by electronic Common Rail (CR) systems. CR systems enable simple regulation of the fuel pressure and the moment of opening the injector. Furthermore, CR systems allow the division of the fuel dose into several parts, injected in a sequence, depending on the current operating conditions of the engine. With variable adjustment of fuel pressure and dividing the fuel dose into several smaller doses, injected in a sequence, the combustion process can be controlled by reducing the maximum heat release rate and extending the combustion time of diesel fuel (Carlucci et al., 2003; Kuti et al., 2013; Wierzbicki, 2014; Yan and Wang, 2011).

In dual-fuel engines, due to the fact that the entire dose of gaseous fuel is located in the combustion chamber, the auto-ignition of the first fuel dose also ignites the gaseous fuel, which makes combustion control in such engines difficult. With this type of fuel injection strategy, the selection of the pilot dose in these engines is essential, as the injection of the dose determines the starting point and the process of combustion of gaseous fuel (Ryu, 2013; Roy, 2014).

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 e.g. by (Motyl and Lisowski 2008). Among the regulation parameters of the engine, the key impact on combustion in a dual-fuel engine is exerted by the setting of parameters of liquid fraction injection. The key factor here is the volume of injected fuel (Doijode et al., 2013; Papagiannakis at al., 2007; Stelmasiak, 2014), fuel injection angle (Selim, 2004; Semin and Rosli, 2009) and the method of spraying the liquid fuel dose. Korakianitis (Korakianitis, 2011) has clearly shown that for lean mixtures of gas and air, gaseous fuel may burn only in the propagation zone of the liquid fuel burn flame. With poor spray of the diesel fuel, the propagation zone and the volume of gas that does not participate in the reaction may reach 50%.

1.2. Motivation for the present research

The cited studies on the impact of the parameters of liquid fuel injection on the combustion process in a dual-fuel engine only consider transition processes. It seems that changing the liquid fuel dose indirectly impacts the combustion process of gaseous fuel by managing the properties of diesel fuel combustion. This statement cannot be confirmed only by experimental tests. This arises from the fact that the observable process is the combined combustion of diesel fuel and gas. Hence the lack of studies that would provide information on the impact of the liquid fuel combustion process on the properties of gas combustion in a dual-fuel engine. The Introduction of an appropriate mathematical model of a dual-fuel engine, verified for a broad range of engine operation parameters, may be a method of analysing the phenomenon discussed herein.

Simulating dual-fuel engines is still relatively uncharted territory. In this study, we used a zero-dimensional multiphase mathematical model, proposed by (Mikulski, 2014), to examine the mutual effect of combustion processes of liquid and gaseous phases in a dual-fuel engine.

2. MATERIALS AND METHODS

The model used for the simulation tests described herein includes the phases of compression, combustion and decompression in the chamber of a dual-fuel compression-ignition engine. A detailed description of the developed mathematical model, as well as the methodology of numeric calculations, can be found elsewhere (Mikulski et al., 2015). The key assumptions of the model are as follows:

- 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 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:

$$\begin{cases} \dfrac{dQ_{in}}{d\alpha} = \dfrac{dU}{d\alpha} + p\dfrac{dV}{d\alpha} + \dfrac{dQ_{out}}{d\alpha} - h_d \cdot \dfrac{dM_d}{d\alpha} \\ p \cdot V = N(\alpha)\bar{R}T \end{cases} \tag{1}$$

  with $Q_{in,}$ as an energy introduced to the system, being a sum of Q_d and Q_g, representing heat released from combustion of diesel and gaseous fuel respectively. The first equation expresses the first law of thermodynamics in a differential form for an open-loop system. The second equation (ideal gas law) enables us to introduce the dependence of pressure and temperature and thus bring the system (1) to one, strongly nonlinear differential equation for the temperature of the system (T).
- The specific heat values of individual components of the mixture, necessary to calculate the changes of internal energy $dU/d\alpha$, were determined as continuous functions of temperature, by interpolating data from JANAF Thermochemical Tables (Chase, 1998), using 4th degree polynomial.
- The model included heat exchange with the walls of the combustion chamber ($dQ_{out}/d\alpha$) 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 ($dM_d/d\alpha$) was simulated with the use of the authors' proprietary correlation, based on normal distribution (Mikulski et al., 2015).
- The starting point of diesel fuel combustion is determined by the ignition delay period. The model also used the equation proposed by Assanis et al. (2003), 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 (Pietak and Mikulski, 2011). 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 (Heywood, 1988), with the coefficient $m_d = 0.4$, obtained from verification tests:

$$\frac{dQ_d(\alpha)}{d\alpha} = P_d\left\{1 - \exp\left[-6.908\left(\frac{\alpha - \alpha_{SOC}}{\Delta\alpha_d}\right)^{m_d+1}\right]\right\} \tag{2}$$

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 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 release from the combustion of individual components is determined by the kinetics of the chemical reaction, with the following formula for global reaction rate:

$$\frac{d[C_iH_{2i+2}]}{dt} = A_n\exp\left(-\frac{Ea_i}{RT}\right)[C_iH_{2i+2}]^{a_i}[O_2]^{b_i} \tag{3}$$

- The formulae 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 ($N_i(\alpha)$) and, at the same time, the heat release rate from the combustion of gaseous fuel:

$$\frac{dQ_g(\alpha)}{d\alpha} = \sum_{i=1}^{3}\frac{dQ_i(\alpha)}{d\alpha} = \sum_{i=1}^{3}H_i\frac{dN_i(\alpha)}{d\alpha} \tag{4}$$

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

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

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

The described model has been verified for an ACDR compression-ignition engine (Table 2), operating in a dual-fuel system. The verification has shown very good conformity of the results of calculations (Mikulski, 2014). The verification has been based on the comparison of the calculated values of pressure for each rotation angle of the crankshaft with the values recorded from experimental tests on the ADCR engine. In the verification procedure, the appropriate coefficients of Wiebe's function have been determined, characterising the combustion process of diesel fuel. The scope and summary of results of verification are provided in Table 3.

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
Fuel consumption at torque peak	210 g/kWh

Table 3. Scope of conditions for which the model has been verified and results of verification

Verification stage	Scope of verification				Max. error		
	n	Tr	p_{max}	S_d	momentary		in cycle
	rpm	[Nm]	[bar]	[%]	[bar]	[%]	[%]
Compression and decompression	750 3400	0	27 42	0	1.1	5	6.6
Diesel fuel operation (undivided fuel dose)	2300 3400	50 200	49 92	100	6.5	8	8.4
Diesel fuel operation (divided fuel dose)	1500	20-200	36-87	100	3.7	6	6.8
Diesel fuel + CNG operation (undivided diesel dose)	3400	50-200	35-48	80	6.5	11	6.1
Diesel fuel + CNG operation (divided diesel dose)	1500	50 150	40 72	16 80	8	15	6.2

In the discussed studies, an engine was used as the test subject, for which the model was verified for dual-fuel operation with diesel fuel and natural gas. Calculations would begin by introducing the basic set of input data into the model, i.e. engine geometry, material data of the engine (necessary to calculate heat exchange), operating parameters, etc.

The model calculations were done in accordance with the diagram provided in Fig. 1, with the calculation step equivalent to 1 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 was launched in a loop, leading to auto-correlation of 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 was obtained, i.e. to the moment when the delta of average values from the last two loops did not exceed the set accuracy limit.

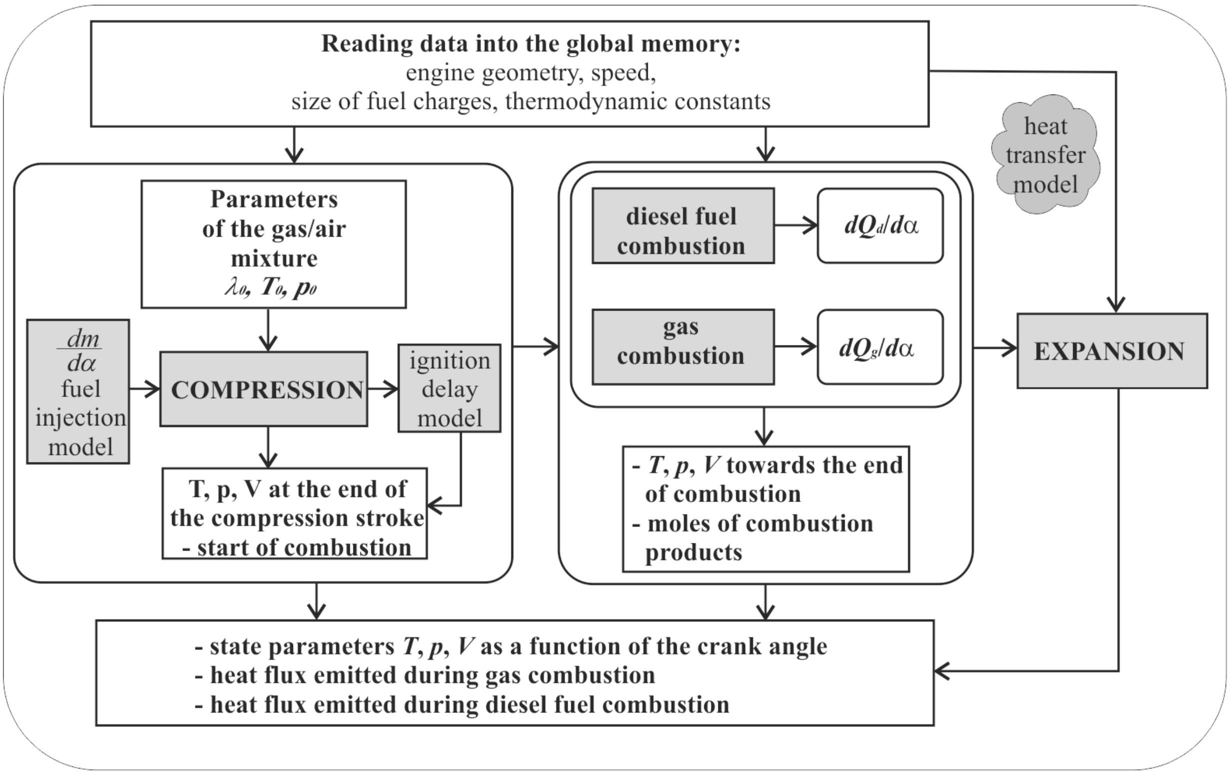

Fig. 1. Design structure of the basic programme (sub-models not included)

As the starting point for the present study, the operating parameters of an ACDR engine were used, at a rotational speed of 3400 rpm and 50 Nm load. The correctness of the model for these parameters was verified for a broad range of gaseous fuel proportions. The scope of conducted simulation tests and the operating parameters of the engine are provided in Table 4.

In the calculations, it was assumed that the engine was fuelled by natural gas with a calorific value of 31.0 MJ/m^3. The chemical composition of the simulated gaseous fuel was as follows: - Methane 98.8% - Nitrogen - 1% - Carbon dioxide - 0.2%. It was assumed that the entire volume of gas supplied to the cylinder would participate in the reaction, and the combustion was perfect. The per cent weight of the gas fed to the engine S_g=(Q_g/Q_{1f})100, was calculated using the energy criterion with the assumption that the total amount of energy in fuel in single-fuel operation and the total amount of energy fed in liquid and gaseous fuel fulfill the equation $Q_g + Q_d = Q_{1f}$.

Table 4. *Scope of presented results of simulation test with values of the input parameters introduced to the model.*

<table>
<tr><td rowspan="2">Sample</td><td>S_d</td><td>G_{air}</td><td>G_d</td><td>G_g</td><td>α_{inj}</td><td>$\Delta\alpha_d$</td><td rowspan="2">Comments</td></tr>
<tr><td>[%]</td><td colspan="3">[kg/h]</td><td colspan="2">[CA deg]</td></tr>
<tr><td>1</td><td>100</td><td>264</td><td>7.3</td><td>0.0</td><td rowspan="4">354</td><td rowspan="4">53</td><td rowspan="4">variable dose of diesel/CNG</td></tr>
<tr><td>2</td><td>80</td><td>264</td><td>5.8</td><td>1.2</td></tr>
<tr><td>3</td><td>50</td><td>264</td><td>3.7</td><td>3.1</td></tr>
<tr><td>4</td><td>30</td><td>264</td><td>2.2</td><td>4.3</td></tr>
<tr><td>5</td><td rowspan="4">30</td><td rowspan="4">264</td><td rowspan="4">2.2</td><td rowspan="4">4.3</td><td rowspan="4">354</td><td>43</td><td rowspan="4">variable diesel combustion duration $\Delta\alpha_d$</td></tr>
<tr><td>6</td><td>33</td></tr>
<tr><td>7</td><td>23</td></tr>
<tr><td>8</td><td>13</td></tr>
<tr><td>9</td><td rowspan="4">30</td><td rowspan="4">264</td><td rowspan="4">2.2</td><td rowspan="4">4.3</td><td>350</td><td rowspan="4">23</td><td rowspan="4">variable injection angles α_{inj}</td></tr>
<tr><td>10</td><td>354</td></tr>
<tr><td>11</td><td>358</td></tr>
<tr><td>12</td><td>362</td></tr>
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3. RESULTS AND DISCUSSION

3.1. Impact of diesel fuel dose

Fig. 2 shows some of the results of simulations for samples 1-4, illustrating the impact of diesel fuel on the combustion process in a dual-fuel engine. The presented diagrams show the charge pressure and temperature values in the cylinder and the heat release rate from combustion of diesel fuel and natural gas, respectively.

Reducing the dose of liquid fuel, assuming that the angle of combustion is identical as for single-fuel operation, results in reducing the maximum heat release rate. This, in turn, generates lower temperatures on the initial stages of combustion and is the cause of the ignition delay of gaseous fuel. This is clearly visible in the diagram of heat release rate from CNG combustion. With a small dose of gaseous fuel, methane ignites virtually at the same time as the liquid fraction. Reduced heat release intensity with a dose with 50% of gas causes a visible extension of methane ignition delay by ca. 5° of crank angle (CA). This fact has a profound impact on the pressure values, causing a shift of the peak pressure further in the decompression phase. With the increase of gas concentration levels, the delay of diesel fuel auto-ignition also increases. This confirms the results of cited experimental studies on this process and the results of initial research by the author. This is a secondary effect of reduced temperature and pressure in the compression phase, arising from the differences of thermal conductivity of the gas-air mixture compared to clean air.

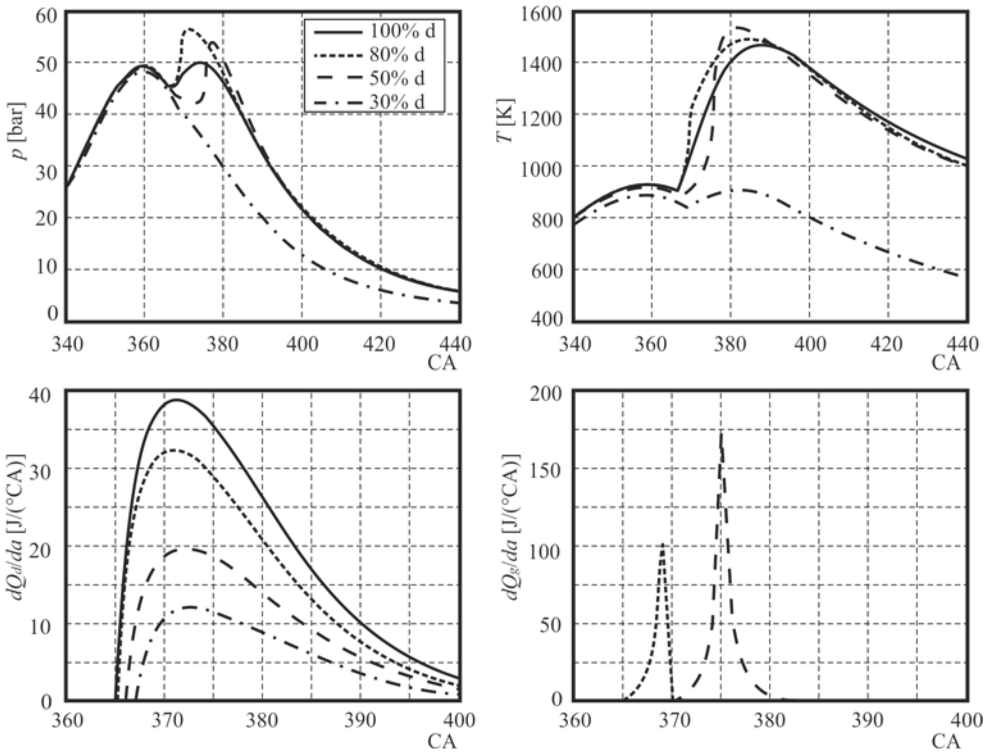

Fig. 2. Simulation results for samples 1-4, different proportions of diesel fuel and CNG, fixed length of combustion of diesel fuel

For a 30% pilot dose of diesel fuel, the temperature generated in the combustion thereof is too low to ignite methane. The assumed angle of combustion is probably much longer than the actual angle in the engine. Any change of the combustion duration of the liquid fraction has a significant impact on the combustion process of gaseous fraction, as shown in the next diagram. The results of simulations conducted for the 20% dose of diesel fuel have indicated that for the assumed conditions, the change of angle of combustion failed (within the feasible values) to ignite methane in each case.

3.2. Impact of the combustion duration of diesel fuel

The combustion duration of diesel fuel, or, more accurately, the heat release rate from its combustion, has a profound impact on the pressure values. The angle of combustion depends on the quality of fuel spray, the advance angle of fuel injection, auto-ignition delay and the geometry of the combustion chamber. With so many factors, it is difficult to choose the appropriate values for each situation. Tests conducted as part of verification have allowed us to estimate the approximate criteria of selection of these values for the analysed engine. According to the developed criterion, the angle of combustion in the conditions set for samples 5-8 should fall within the range of 33° CA. Furthermore, the combustion process may be controlled to some extent by appropriate distribution of the time of injection or dividing the injection into several doses. Therefore, further studies are necessary on the impact of the length of the diesel fuel combustion process on the processes occurring in a multi-fuel engine.

Reducing or extending the combustion duration of diesel fuel may be the key to stable combustion of gas in the given conditions. Reduced combustion may cause the gas to ignite even with very small initial doses of diesel fuel. Fig. 3 indicates that appropriate setting of the angle may have a positive impact on the average indicated pressure and performance of the engine by shifting the peak pressure towards the decompression process. At the same time, the maximum pressure value is reduced, and gas combustion duration is extended. This may limit the range of knocking combustion of the gaseous fraction. Too rapid combustion of diesel fuel causes quick ignition and combustion of the gaseous fraction. This results in maximisation of the combustion temperature, which may have a negative effect on nitrogen oxides emissions.

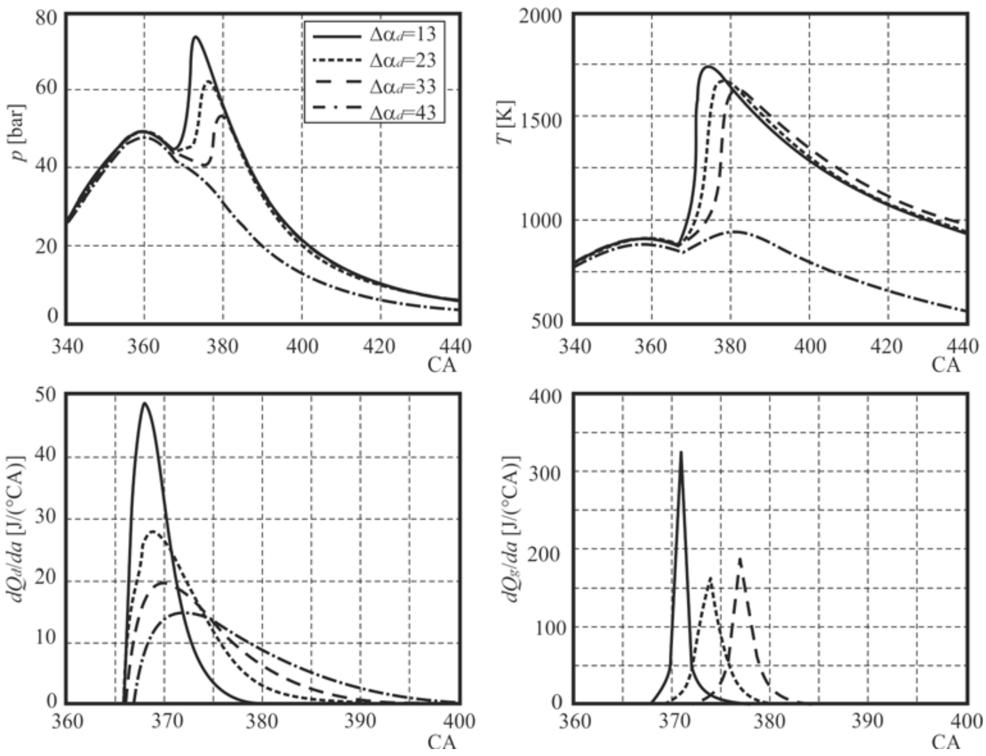

Fig. 3. Results of simulation for samples 5-8, U_d = 30%, variable combustion durations of diesel fuel

Too rapid combustion of diesel fuel may reduce the volume of gas participating in the reaction and result in discharging the unburned hydrocarbons with exhaust fumes. In the case of lean gas-air mixtures, only a portion of gaseous fuel in the propagation zone of combustion of sprayed liquid fuel is burned. In the event of a short combustion time, the exhaust fume zone is insufficiently mixed with the fresh charge within the flame zone. In extreme cases, this may result in burning only 65% of the gas in the cylinder. This causes a dramatic drop in general performance and an increase of hydrocarbon emissions.

3.3. Impact of the diesel fuel injection advance angle

In the dual-fuel engine, the ignition point of the initial dose can be controlled in a much broader scope. Control is performed mainly by setting the starting point of liquid fuel injection (Fig. 4). The model enables direct analysis of the impact of the control value on the combustion process by allowing dynamic calculation of the angle of ignition delay.

If the fuel is injected before the TDC, changing the angle of injection does not generate any major changes of the ignition point. This arises from the fact that fuel injected early before TDC coincides with much lower pressure and temperature values in the cylinder, which significantly prolongs the auto-ignition delay. The later injection is compensated by a significantly reduced angle of ignition delay of the initial dose. However, an earlier injection always ensures more reliable ignition of the gaseous fuel and faster combustion thereof, and consequently, high pressure increments. The calculations clearly show that injecting the fuel too late may result in failure to ignite the gaseous fuel, despite burning the diesel fuel.

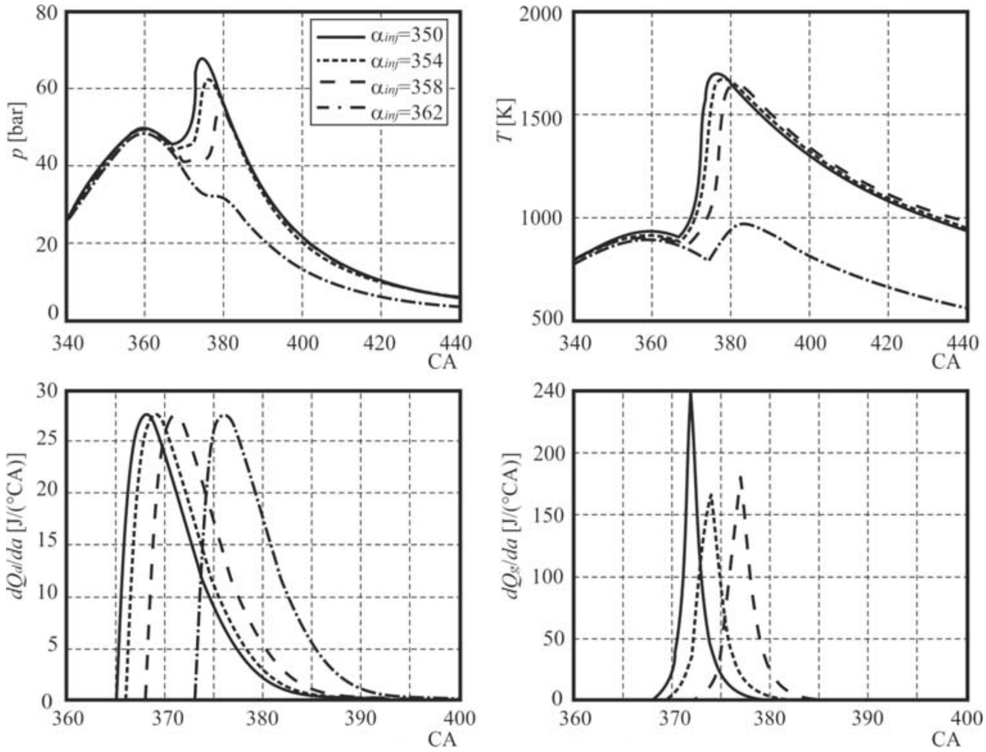

Fig. 4. Results of simulation for samples 9-12, U_d = 30%, variable angles of injection start

3.4. Comparative analysis of combustion performance in a dual-fuel compression-ignition engine

Further on, the parameters describing the combustion process for all simulation points were compared. Below, the results are provided for calculations of: maximum combustion pressure and indicated efficiency (Fig. 5), maximum combustion temperature and maximum heat release rate from gaseous fuel combustion (Fig. 6), and ignition delay of diesel and CNG associated with the duration of CNG combustion (Fig.7).

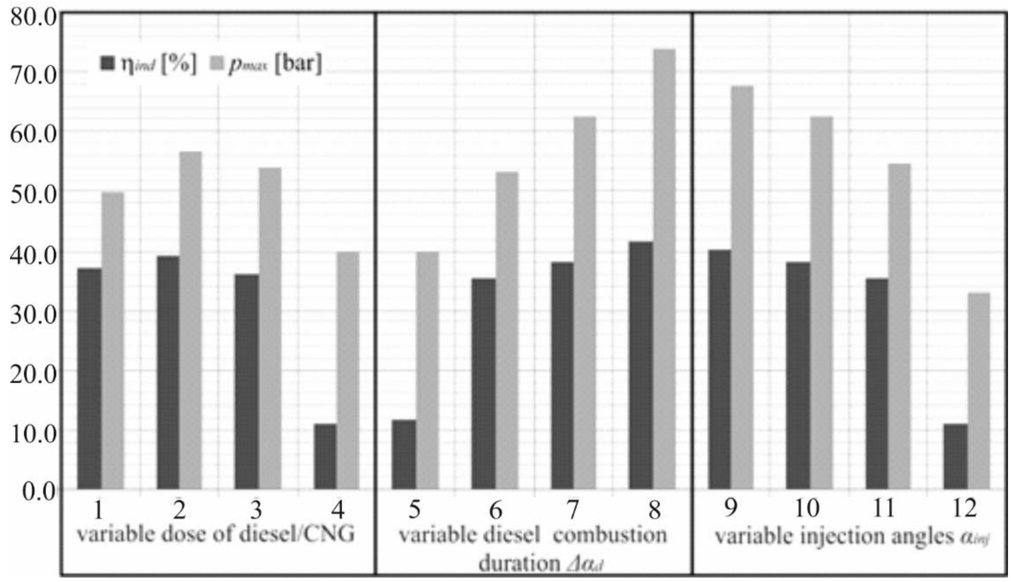

Fig. 5. Maximum combustion pressure in the engine cylinder (p_{max}) and indicated efficiency (η_{ind}) for all simulation points

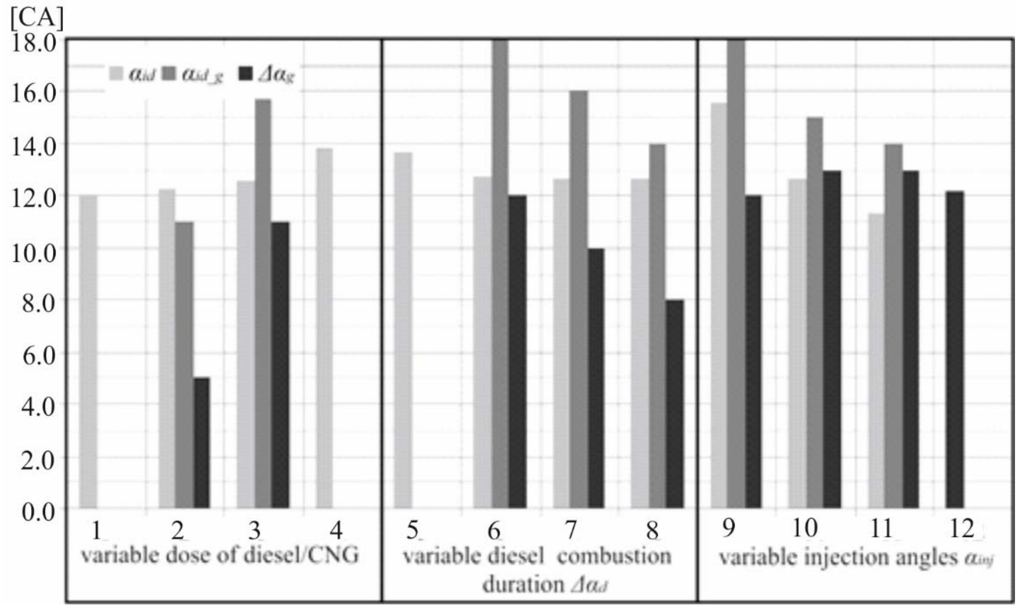

Fig. 6. Ignition delay angle of diesel (α_{id}) and CNG (α_{id_g}) associated with the duration of CNG combustion ($\Delta\alpha_g$) for all simulation points

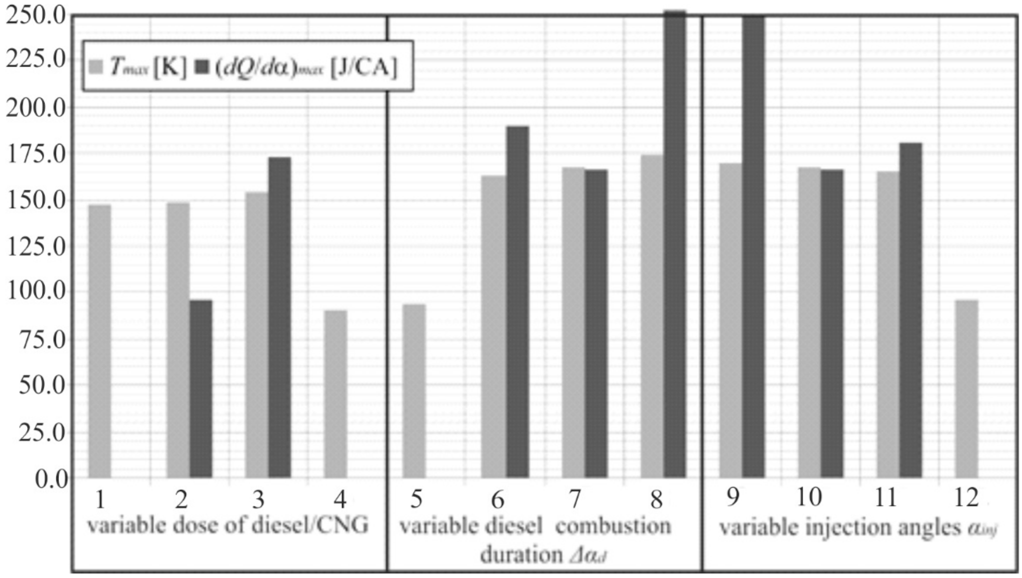

Fig. 7. Maximum combustion temperature (T_{max}) in the engine cylinder and maximum heat release rate from gaseous fuel combustion ($(dQ/d\alpha)_{max}$) for all simulation points

According to the model calculations, increasing the energy share of CNG can at some points lead to increased engine indicated efficiency by 2% (Fig. 5), due to higher maximum pressure, generated by higher heat release rates of CNG, compared to diesel fuel. For energy shares of gaseous fuel of 50%, a slight (1% less compared to traditional diesel operation) decrease in efficiency was observed. This tendency was more significant for higher gaseous fuel energy shares and was caused by significantly increasing ignition delay of gaseous fuel (Fig. 7), generating lover maximum pressures (Fig. 6). The increased amount of gaseous fuel fed to the engine also increased the ignition delay period of the diesel fuel (Fig. 7). This confirms the results obtained by other authors in terms of effect of the addition of methane into the air on the process of ignition delay in a dual fuel engine (Aesoy and Valland, 1996).

Shortening the duration of diesel fuel combustion always caused an increase in indicated efficiency, maximum pressure and temperature (Fig. 5, Fig. 6). This is caused by accelerating the induction time of gaseous fuel ignition, keeping constant diesel fuel ignition delay. Higher heat release rates of diesel generate higher heat release rates of CNG combustion, shortening the combustion time. Figure 5 indicates that a proper regulation of diesel combustion duration alone can have the most significant effect on engine performance, leading to an increase of 3.5 % (sample 8) in indicated efficiency, compared to mono-fuel, diesel operation (sample 1). Changing the injection angle has only limited impact on improving combustion. For the 30% energy share of gaseous fuel and current speed and load parameters, the angle of 10 CA before TDC was optimal in terms of engine performance (Fig. 5). Shifting the injection angle forwards TDC decreased indicated efficiency dew to decreasing maximum heat release rate of gaseous fuel and later ignition resulting in lower maximum combustion pressures. Shifting the injection angle further before TDC, from optimal -10 CA, also resulted in a slight decrease of efficiency, by generating very long ignition delay periods.

Figure 6 indicates that adding gaseous fuel always leads to the increase in combustion temperature compared to pure diesel operation. This might have a negative effect on NO_x emission. Proper control strategies can limit the increase of combustion temperature.

4. CONCLUSIONS

Model calculations have allowed us to isolate the impact of specific elementary phenomena in the engine's cylinder and to analyse them separately. This significantly expands the range of options available in experimental studies. The results of the tests have considerably broadened our knowledge on the mutual effect of the combustion processes of liquid and gaseous phases in a dual-fuel engine. The following detailed conclusions have been formulated:

- Reducing the dose of diesel fuel, while maintaining a fixed combustion duration, causes an increase of the ignition delay and heat release rate from the combustion of gaseous fuel.
- Too small dose of diesel fuel, burning over a long period of time, is incapable of igniting the gaseous fuel.
- The ignition conditions of gaseous fuel may be significantly improved by increasing the combustion rate of diesel fuel. Accelerating the combustion rate of diesel fuel allows to use extremely small initial doses in dual-fuel engines.
- Accelerating the combustion rate of diesel fuel accelerates the combustion rate of gaseous fuel. Applied in the appropriate scope, it can improve engine performance. Too rapid combustion of diesel fuel may affect the combustion process of gas, causing imperfect combustion and thus a reduction of engine performance and an increase of emissions.
- Increasing the angle of injection start results in an increase of the angle of ignition in a dual-fuel engine. However, the impact is much less prominent than in a traditional compression-ignition engine and drops with the increase of the gaseous fuel dose.

One should bear in mind that the above conclusions are in force, when complete gaseous fuel combustion is achieved. The used mathematical model is unable to predict which part of the CNG amount in the cylinder is participating in the reactions due to its zero-dimensional nature.

Regardless of the above limitation, the studies have shown that the combustion process of gas in a dual-fuel engine can be managed by an appropriate setting of the combustion process of the liquid fraction. Also, it has been shown that the key control parameter in this case is the burning rate of diesel fuel. Management of the characteristics of diesel fuel combustion is possible indirectly by controlling the injection pressure (quality of spray), fuel temperature and the time and sequence of injections. It can be therefore concluded that the regulation strategies, currently employed for improving the performance of dual-fuel engines and providing for only the fuel injection advance angle and the dose of liquid fuel, should be modified.

SYMBOLS

Ea	activation energy, MJ/mol
G	specific fuel consumption, kg/h
H	calorific value, MJ/kg
h	enthalpy, MJ/kg
M	mass of injected fuel, kg
N	number of moles, mol
P	chemical power in the specific fuel, kW
p	pressure of the medium in the cylinder, bar
Q	heat (introduced to or dissipated from the system), J
$\overline{R}$	gas constant, J/(mol·K)
S	per cent weight of the specific fuel, %
T	temperature of the medium in the cylinder, K
Tr	torque, Nm
U	internal energy of the system, J
V	volume of the medium in the cylinder, m^3
α	crank shaft rotation angle, °CA
η	efficiency, %

Subscripts

air	air
d	diesel fuel
g	gaseous fuel
i	individual combustible gaseous component (1 – methane, 2 – ethane, 3 – propane)
in	introduced to the system
lf	mixed fuel
out	dissipated from the system
id	ignition delay
ind	indicated
inj	injection
1f	mono-fuel operation

Abbreviations

CA	Crank angle degrees
CR	Common Rail
TDC	Top Dead Centre
CH_4	Methane
C_2H_6	Ethane
C_3H_8	Propane
NO_x	Nitrous oxides

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Received 20 January 2015
Received in revised form 12 May 2015
Accepted 13 May 2015