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ZERO-DIMENSIONAL 2-PHASE COMBUSTION MODEL IN A DUAL-FUEL COMPRESSION IGNITION ENGINE FED WITH GASEOUS FUEL AND A DIVIDED DIESEL FUEL CHARGE

ZERO-WYMIAROWY 2-FAZOWY MODEL SPALANIA W DWUPALIWOWYM SILNIKU O ZAPŁONIE SAMOCZYNNYM ZASILANYM PALIWEM GAZOWYM I DZIELONĄ DAWKĄ OLEJU NAPĘDOWEGO*

The problem of using alternative fueling sources for combustion engines has been growing in importance recently. This is connected not only with the dwindling oil resources, but also with the growing concern for the natural environment and the fight against global warming. This paper proposes the concept for a zero-dimensional model of a multi-fuel engine, enabling the determination of thermodynamic system parameters based on the basic geometric and material object data (complete model). The basic problems in the creation of this model and the modeling of the accompanying subprocesses have been outlined and the methodology of the numerical solution of the obtained mathematical description has been proposed. The basic characteristics of the developed model are: the application of an original model of liquid fraction injection based on normal distribution, a new Assanis correlation for computing the diesel fuel self-ignition delay period in the presence of gas, first-order chemical reaction kinetic equations for describing the course of combustion for combustible components of the gas/air mixture, the implementation of a self-consistency procedure in modeling heat exchange and the effect of exhaust recirculation, the inclusion of both a single liquid fuel injection and the possibility of performing computations for a divided charge.

***Keywords**: dual-fuel engine, engine combustion process, mathematical model, computer simulation, heat exchange*

W ostatnim czasie problem wykorzystania alternatywnych źródeł zasilania silników spalinowych zyskuje szczególnie na znaczeniu. Związane jest to nie tylko z kurczącymi się zasobami ropy naftowej, ale również z coraz większą troską o środowisko naturalne oraz walką z globalnym ociepleniem. W niniejszej pracy zaproponowano koncepcję zero-wymiarowego modelu silnika wielopaliwowego, umożliwiającego, wyznaczenie parametrów termodynamicznych układu w oparciu o podstawowe dane geometryczne i materiałowe obiektu (model kompletny). Nakreślono podstawowe problemy w zagadnieniu tworzenia takiego modelu i modelowania podprocesów towarzyszących oraz zaproponowano metodykę numerycznego rozwiązywania uzyskanego opisu matematycznego. Podstawowe wyróżniki opracowanego modelu to: zastosowanie autorskiego modelu procesu wtrysku frakcji ciekłej opartego na rozkładzie normalnym, nowej korelacji Assanisa do obliczenia okresu zwłoki samozapłonu oleju napędowego w obecności gazu, jednostopniowych równań kinetyki reakcji chemicznej do opisu przebiegu spalania składników palnych mieszaniny gaz-powietrze, implementacja procedury samouzgodnienia w modelowaniu procesu wymiany ciepła i wpływu recyrkulacji spalin, uwzględnienie zarówno pojedynczego wtrysku paliwa ciekłego jak i możliwość prowadzenia obliczeń dla dawki dzielonej.

***Słowa kluczowe**: silnik dwupaliwowy, proces spalania w silniku, model matematyczny, symulacja komputerowa, wymiana ciepła.*

1. Introduction

The current research on compression ignition combustion engines is primarily oriented towards the use of alternative renewable fuels, reduction of toxic air emissions and the development of new engine diagnostic methods [5, 6, 15]. One of the increasingly often-addressed problems in engine research is the possibility of dual-fueling engines with the main fuel charge fed as a gaseous fuel.
An increased interest in dual-fuel compression ignition engines in recent years has contributed to the development of both experimental and model testing on this fueling method.

One of the first valuable models simulating the combustion process in the full cycle of a dual-fuel engine was proposed by Raine in 1990 [24]. The operation of this model was limited only to the computation of the system's efficiency parameters.

Mansour et al. [14] proposed a simple, zero-dimensional model of combustion in a CNG-fueled engine with a pilot diesel fuel charge. Heat exchange with the chamber walls in this model was limited to convection and a number of other simplifying assumptions were adopted. The authors used a Wiebe function to describe energy release during diesel fuel and natural gas combustion in the first program

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step. The calculated pressure and temperature values were used as the input data for detailed kinetic computations, for which the commercial CHEMKIN package was used. The integration of complex combustion mechanisms with processes occurring in the dual-fuel engine, with the adopted simplifications, seems justified only in detailed research on exhaust toxicity, for which this model was used. In the assessment of efficiency and the power attained by the engine, the adopted simplifications in the description of individual subprocesses do not justify the need to apply very accurate combustion models, which require the use of dedicated computational environments.

Abd Alla [2] presented a model of pure methane combustion in a dual-fuel engine, calling it quasi-two zone. Leaving the nomenclature aside, the main distinguishing feature of this model was the application of detailed reaction kinetics for the description of gaseous phase combustion. A scheme of 178 elementary reactions, in which 41 chemical species participated, was used in this model, a Wiebe function was used to describe liquid phase combustion. The model description included the compression phase through combustion until the outlet valve opening. A detailed submodel of exhaust component formation was added to the basic thermochemical model. The basic advantage of this approach was the creation of connections between the progress of combustion and time and thermodynamic load parameters. This model showed quite good agreement with the experiment and a high predictive capacity over a wide range of engine operating conditions.

Stelmasiak [25] proposed a relatively simple and complete approach to the problem of modeling dual-fuel systems. The zero-dimensional model assumed the separate combustion of two phases – liquid and gaseous. It is assumed in this model that the start of combustion is the same for both fuels and is determined by the ignition delay period. The combustion end angles for diesel fuel and CNG were different and set by the modeler. The heat release rate and the concentrations of the homogeneous gas/DF/air mixture components were imposed by a Wiebe function, separately for gas and diesel fuel. The empirical formula of Prakash [23] was used to predict diesel fuel ignition lag, taking into account the presence of natural gas in the cylinder.

Papagiannakis et al. used a gradually-improved model of a dual-fuel engine burning natural gas and diesel fuel in a number of their papers [9, 10, 11, 17, 18]. The authors used the fundamental laws of conservation of mass, energy and momentum for combustion process simulation. They proposed a two-zone description of the load in the cylinder space with division into the unburnt load zone and the exhaust zone. The fuel combustion front was described as a cone surface propagating from the point of injection. The propagation occurs in the direction normal to each cone surface with the speed defined by the jet model of Hiroyasu et al. [8]. The energy released during combustion depends, according to the authors, only on the propagation speed of the front, behind which the homogeneous product zone immediately changes into the homogeneous exhaust zone. The effect of gas composition on the combustion process is neglected in this approach and the combustion rate is correlated with experimental data. The model was verified for a single-cylinder CI engine – Lister LV1 modified for dual-fuel operation. Good agreement with measurement results was demonstrated and the model was successfully used for testing the effect of the initiating charge size on engine operation parameters [19]. The basic model limitations result from its operating range. The simulation starts at the combustion start point, which must be given as an input parameter; similarly, the mean cylinder temperature and pressure during the cycle must be calculated from experimental data.

Perini et al. [21] developed a two-phase, quasi-dimensional model of a SI engine operating on LPG admixed with hydrogen and Ma et al. [13] proposed a mathematical simulation of a hydrogen engine. A dual-fuel engine operating on LPG with a hydrogen admixture was modeled using equilibrium reactions by Lata and Ashok Misra [12]. Wang [26] proposed a similar model for CNG and hydrogen mixtures based on a CFD code and chemical reaction kinetics.

Agarwal and Assanis [1] proposed a complex, multi-dimensional model of natural gas combustion under conditions typical for compression ignition engines. The model used detailed chemical reaction kinetics.

The problem of the mathematical modeling of dual-fuel systems is still innovative. Complete models are still lacking which, including all processes occurring in the cylinder, are able to predict with sufficient accuracy changes in thermodynamic working medium parameters, using only the basic object data available from its technical documentation. Among the available models, an even lower number has undergone full experimental verification, which limits their reliability. Moreover, there is a lack of research results on the effect of gas composition on the course of combustion in a dual-fuel engine. A properly constructed and verified model can significantly contribute to the filling of these deficiencies.

2. Assumptions of the model of combustion in a dual-fuel CI engine

In dual-fuel engines, a full description of the combustion process is much more difficult compared to single fuel engines. The cylinder contains gaseous fuel mixed with air, into which liquid fuel is injected near TDC. Extremely different mechanisms dominate in the combustion of these two fractions, both of a physical (mass and energy transport processes) and chemical nature, resulting from the kinetics of fuel reaction with oxygen. All processes must therefore be analyzed, taking into account not only their effect on the gaseous and liquid fuel combustion rate, but also in regard to the proportions of individual-fuels.

On the other hand, the liquid and gaseous phases are combusted at the same time, in the same space and the presence of one fraction affects the combustion of the other. The mechanisms of this influence are not yet well-known.

The examination of all these elements allowed the model assumptions to be formulated, bearing in mind both the accuracy of its results and the complexity level, affecting the autonomy (the amount of necessary experimental data) and the computational speed.

Analyzing natural gas combustion in the engine, the following assumptions were made, which characterize the postulated model. It is assumed that a homogeneous gas/air mixture is in the cylinder during compression and temperature and pressure distribution at each point of the cylinder space is constant. This assumption leads to a zero-dimensional model. At the same time, the working medium is treated at each moment as a mixture of semi-ideal gases, mixed in any proportions. The specific heat of the mixture at each computational point is the sum of functions approximating data from thermodynamic tables for each component [4]. The approximating polynomials are functions of temperature. The changes occurring in the modeled space are treated as quasi-static. In the developed model, heat exchange with the cylinder walls was based on convection between gas and the surface of the given component and a zero wall temperature gradient was adopted, assuming that it was constant. The heat removed from the system is the sum of three fluxes passing through the surface of the piston head, the cylinder head and the time-variable cylinder wall surface. Increased heat exchange as a result of flame radiation was additionally taken into account in the combustion process.

The developed model takes into account the effect of the mass of the injected pilot charge on the thermodynamic state of the system, through a simplified fuel injection submodel based on normal distribution. This original approach allows the model to operate independently of the detailed experimental data on the course of the pressure drop and the injector needle lift for the tested object. It is assumed in the built model that combustion starts at the moment of diesel fuel

ignition, which initiates the gas mixture ignition. The two fuels are combusted separately and the emitted heat is the sum of heat released in the combustion of both fuels. The course of diesel fuel combustion is described by empirical equations for the ignition delay period and process rate – as represented by a Wiebe function. This is a simplified approach, but justified for dual-fuel engines because of the small pilot charge size.

Basic research on combustion processes proves that methane combustion is a very complex process and consists altogether of a system of 132 chemical reactions, whose rates are interrelated by the concentration of individual reactants. The implementation of such a complex model in a complete dual-fuel engine working cycle is unjustified due to the applied simplifications in the description of the other subprocesses (heat exchange, fuel injection etc.). A satisfactory representation of the combustion process can be obtained using even one methane combustion macroreaction [20]. In the proposed model, a system of three interrelated global reactions, including the combustion of the most important combustible components used to fuel gas engines, was implemented for the description of gaseous phase combustion.

It is assumed that the developed model with the adopted assumptions will enable precise examination of the effect of gas composition on the character of combustion in a dual-fuel engine.

3. Basic equations of the developed model of fuel combustion in a dual-fuel engine

According to the discussed assumptions, the cylinder load is a homogeneous mixture of air, natural gas, diesel fuel and exhaust at each moment of time. The proportions of individual components change with the stages of the injection and combustion of combustible components. With such assumptions, starting from the energy conservation law, the basic equations of the model can be introduced in a differential form:

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

The first equation expresses the first law of thermodynamics in a differential form for an open system, where: dU – increase in the internal energy of the system, p, V – are the pressure and volume of the working medium in the cylinder, respectively, Q_{out} represents the heat exchanged with the cylinder walls, Q_{in} – the heat fed to the system, h_{ON} – is diesel fuel enthalpy and M_{ON} – the mass of injected fuel. The second equation of the system represents an equation of state, where: R is the universal gas constant, n – medium moles in the cylinder and T its temperature. All the quantities in the system (1) are functions of the crank angle.

Heat exchange with the walls was included in the developed model as the sum of three fluxes passing through the cylinder wall and head and the piston head. Because the main part of heat is removed in CI engines as a result of convection between gas in the working space and the given surface, it can be written:

$$\frac{dQ_{out}}{d\alpha} = \frac{1}{\omega} \cdot h_c(T,p,V) \cdot \left[A_g \cdot (T - \bar{T}_g) + A_t \cdot (T - \bar{T}_t) + A_s(\alpha) \cdot (T - \bar{T}_s) \right] \tag{2}$$

A simplified model of heat exchange in a combustion chamber wall and on the side of the coolant is included in equation (2) by expressions for the mean value of cylinder piston (t), head (g) and wall (s) temperatures:

$$\bar{T}_{gts} = \frac{\bar{h}_c \bar{T} + k_{gts} T_{ch}}{\bar{h}_c + k_{gts}} \tag{3}$$

These temperatures were computed as the weighted mean from the mean temperature $\bar{T}$ of the medium in the chamber with the mean heat transfer coefficient $\overline{h_c}$ and coolant temperature T_{ch} with the coefficient of heat transmission through the respective wall $k_{g,\,t,\,s}$:

$$k_{gts} = \frac{1}{\dfrac{1}{h_{ch}} - \dfrac{\delta_{gts}}{\lambda_{gts}}} \int_{\alpha_A}^{\alpha_E} h_c d\alpha \tag{4}$$

where: λ_{gts} – heat transfer coefficient of the wall material, δ_{gts} – wall thickness.

Heat transfer occurs under a number of conditions – variable cylinder gas pressure and temperature and with locally variable rates of medium flow in the cylinder. These conditions are taken into account in the model by adopting an appropriate heat exchange coefficient h_c. Washini's formula was used for this purpose, in the form modified by Hohenberg [9]:

$$h_c = 130 \cdot \frac{\left(p \cdot 10^5\right)^{0,8}}{V^{0,06} T^{0,4}} \cdot (C_{fr} + 1,40) \tag{5}$$

where: $C_{śr}$ – mean piston speed.

The thermodynamic medium parameters change during liquid fuel injection into the engine's combustion chamber. The effect of the fuel mass flux supplied to the cylinder must therefore be taken into account. In direct ignition engines, temperature change as a result of stream evaporation can be neglected, because of a low ratio of the injected fuel volume to the working medium volume. This assumption is all the more justified for dual-fuel engines with diesel fuel charges of up to fifteen percent of the main charge. Another paper by the authors [16] discusses using complex hydraulic models to describe the diesel fuel mass flux in zero-dimensional models. It was demonstrated that a simple, complete model based on normal distribution with normalization to the total fuel charge could be used in this case without an observable loss of accuracy:

$$\dot{M}_{ON}(\alpha) = \begin{cases} 0 & \alpha_{pw} > \alpha < \alpha_{kw} \\ \dfrac{hB_{ON}}{\sqrt{\pi}} e^{-h^2(\alpha - \alpha_{\max})} & \alpha_{pw} < \alpha < \alpha_{kw} \\ 10^{-3} B_{ON} & \alpha_{pw} = \alpha = \alpha_{kw} \end{cases} \tag{6}$$

In a dual-fuel engine, the air/gas mixture ignites from the energy released from the combustion of the pilot diesel fuel charge. Test results show that even for a pilot charge lower than 1% of the total energy, the ignition energy can be by around 35 J higher in a dual-fuel engine than the electric pulse energy in a spark engine [25]. It can thus be assumed that in a dual-fuel engine the start of the combustion of both fractions (liquid and gas) is the same and the problem of ignition moment determination comes down to the accurate modeling of pilot charge ignition lag. This process, however, takes place under conditions significantly different from those in a classical compression ignition engine. Increasing the concentration of combustible gases in the cylinder causes, in most cases, a longer diesel fuel self-ignition delay. In an earlier paper [22], the authors examined the possibility of us-

ing ignition delay correlations verified for traditional CI engines. The equation proposed by Assanis [3] was used in the developed model to determine self-ignition lag:

$$\tau_{id} = 2,4\varphi^{-0,2} p^{-1,02} \exp\left(\frac{E_a}{\bar{R}T}\right) \tag{7}$$

It was demonstrated in the paper [22] that this correlation, taking into account cylinder load composition, describes self-ignition delay in dual-fuel engines better than others.

Diesel fuel combustion was represented in the developed model with a Wiebe function [7]. This allows the effect of the course of diesel fuel combustion on gaseous fraction combustion to be tested; however, this requires prior model validation for single fueling.

The natural gas combustion model was based on first-order oxidation macroreactions for the main combustible mixture components: methane (CH_4), ethane (C_2H_6) and propane (C_3H_8). This leads to a system of three global reactions of the form:

$$C_nH_{2n+2} + \left(\frac{5}{4}n + \frac{1}{2}\right)O_2 \rightarrow nCO_2 + (n+1)H_2O \tag{8}$$

for n = 1...3. The expression for the global reaction rate in this case will assume the form:

$$\frac{d[C_nH_{2n+2}]}{dt} = A_n \exp\left(-\frac{E_{a_n}}{\bar{R}T}\right) \cdot [C_nH_{2n+2}]^{a_n} [O_2]^{b_n} \tag{9}$$

The constant values present in equation (9), for individual gases, were collected in Table 1.

Table 1. Values of the constants included in equation (9). Activation energy Ea (kcal/mol).

No	Fuel	A	E_a	a	b
1	CH_4	$8,3 \cdot 10^6$	30	-0,3	1,3
2	C_2H_6	$1,1 \cdot 10^{12}$	30	0,1	1,65
3	C_3H_8	$8,6 \cdot 10^{11}$	30	0,1	1,65

The solution of each equation of the system (9) fully determines the course of heat release in the gas combustion process at any moment.

4. Methodology of the model's numerical computations

The developed mathematical model was implemented in the MathWorks MATLAB environment. The basis for creating the software program was the assumption of its modular structure. Individual submodels (e.g. of heat exchange, fuel injection, ignition delay, heat release during combustion, change in medium moles etc.) were treated as separate functional m-files called by the main program built around the basic system of differential equations for state variables (1). This approach guarantees program clarity, which facilitates its verification and optimization. The modular structure ensures, moreover, project expandability, by easy exchange of individual submodels with others – more detailed – when these are available.

The operation of the program starts with the entry of the basic input data into the global memory, i.e. engine geometry, engine material data (necessary for heat exchange computations), engine operating parameters etc. The thermodynamic load parameters (λ_o, T_o, p_o) at the start of compression are then computed, from which the computation of cylinder temperature and pressure during the entire cycle starts.

The computational cycle of the main program was divided into 3 main processes: compression, combustion and expansion. The system of equations (1) for compression can be reduced to a single, nonlinear first-order differential equation for cylinder medium temperature as a function of the crank angle. Developing the expression for the internal energy of the system and heat exchanged with the walls (2), we obtain in this case:

$$\sum n_i C_{vi} \frac{dT}{d\alpha} + \bar{R}\frac{n(\alpha)}{V(\alpha)} \cdot \frac{dV(\alpha)}{d\alpha} \cdot T + \frac{1}{\omega} \cdot \left[(130 \cdot 10^4)\frac{\bar{R}^{0,8} \cdot n^{0,8}}{V^{0,86}} \cdot (C_{lr} + 1,40)\right] \cdot \cdot T^{0,4} \cdot [A_g \cdot (T - \bar{T}_g) + A_T \cdot (T - \bar{T}_t) + A_c(\alpha) \cdot (T - \bar{T}_c)] = 0 \tag{10}$$

Due to the complexity level of the above equation, using the dedicated MATLAB package functions for solving differential equations was dropped in favor of a solution by the iterative method. The temperature derivative can be expressed, based on its definition, as the difference quotient limit:

$$\frac{dT}{d\alpha} = \lim_{\Delta\alpha \rightarrow 0} \frac{T(\alpha + \Delta\alpha) - T(\alpha)}{\Delta\alpha} \tag{11}$$

In the numerical solution approximation, the quantity $\Delta\alpha \rightarrow 0$ can be replaced with a finite value, obtaining instead of the differential equation (10) the corresponding difference equation, in which the successive temperature values are calculated based on the values computed in the previous step. The accuracy of the computations will be higher, the lower the value of the computational step is. The parameters for the next angle $\alpha_A + \Delta\alpha$ are calculated this way starting from the values T_o, p_o, corresponding to the crank angle α_A falling on the inlet valve closure. The other quantities in the equation (10) are calculated by complementary subprograms: the moles of individual components $n_i(\alpha)$ and the total load moles $n(\alpha)$ from the solution of the kinetics equations (9). The volume $V(\alpha)$ is calculated from rod-crank system geometry etc.

The procedure continues until the angle for which liquid fuel pumping starts is reached. Starting from this point, the pilot charge injection submodel (6) is included. For the same point, the ignition angle α_p is computed based on the calculated temperature and pressure values, according to the adopted model (7). The compression procedure continues until the calculated α_p value is reached, starting from which combustion starts. In this loop, computations are performed analogously as for compression including the heat flux released during combustion, with variable instantaneous values of load moles depending on the rotation angle. The combustion end angle is defined by the moment when combustible components burn out.

Expansion proceeds according to an analogous equation as for compression with the moles of individual components determined at the combustion end point. The parameters for expansion are calculated until the angle corresponding to the outlet valve opening is reached, which ends the model operation.

It should be noted that the heat exchange equation used in the model (2) requires the mean temperature and the mean heat exchange coefficient for the cylinder load over the entire cycle for the computation of cylinder wall, head and piston head temperatures (3). It is obvious that these values are not known until the instantaneous values of these parameters are calculated. In the simplest approximation, the above mean values can be set as model operation parameters, based on the modeler's experience or the results of experimental tests on the given engine. The first solution causes high uncertainties of computation results and the second leads to considerable limitation of model autonomy. These problems were solved by running the discussed

main program (Fig. 1) in a loop leading to the autocorrelation of the results. The main program is called by an external m-file with set initial values of the mean temperature T_{sr} = 600 K and the heat exchange coefficient h_{csr} = 800 W/m^2. After computations are performed by the main program (cylinder pressure and temperature changes), new mean values are calculated and the results are recalculated in the next loop. The procedure is repeated until the results become self-consistent, i.e. until the moment when the difference between the mean values from the two latest loops does not exceed the set threshold. This approach considerably lengthens computations, but ensures the required autonomy without loss of accuracy. Preliminary calculations showed that with set initial parameters, running the main program 3–4 times is sufficient for the difference not to exceed the value of 1 (K and W/m^2, respectively) for both quantities.

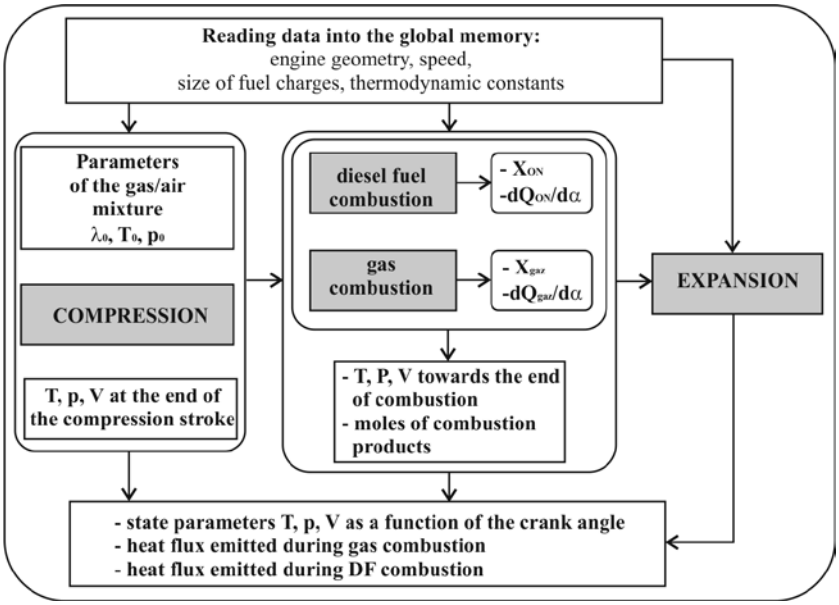

Fig. 1. Computational structure of the main program (excluding submodels)

The exhaust residue percentage in the load sucked into the cylinder is determined similarly. In the first computational step, the exhaust residue percentage is not included because the moles of individual exhaust components have not yet been calculated by the model. The calculations from the first loop are included in the next loop. The procedure of updating the computation results for the moles is repeated until the model self-consistency procedure ends.

5. Simulation capacity of the developed model

The presented model was subjected to the procedure of experimental verification by comparing the calculated pressure changes with the results of indicator tests on a dual-fueled ADCR engine. The verification tests of the model were conducted on the tested engine both for a single diesel fuel charge and for a divided charge (pilot and main charge injection). Good agreement of the computation results was obtained during model verification (Table 2). Example pressure changes recorded in the engine's combustion chamber and the changes obtained from the developed model are shown in Fig. 2.

Detailed discussion of the methodology and broader results of the verification tests will be presented in further papers by the authors.

The developed model in its present specification can be used as a research tool in two main ranges, i.e. autonomous operation and semi-autonomous operation, in the analysis of experimental changes.

In the autonomous operation range, only the basic technical data of the tested engine and the parameters characterizing the working point are used as the input data. Each input parameter can be used as an independent variable, which enables the simulation of different real processes. Moreover, it is possible in the model to insert the calculation of some parameters and their introduction into the computational program as an independent variable, e.g. the ignition delay angle. With such model applicability, the effect on the course of combustion in a dual-fuel engine can be tested for such parameters as:

- engine geometry;
- injection start angle (of both a single and a dual charge);
- temperature of the medium sucked into the cylinder;
- consumption (of air, liquid fuel, gas);
- ignition delay angle;
- gaseous fuel composition;
- diesel fuel combustion time.

In the semi-autonomous operation mode, the model can be successfully used for the analysis of real changes obtained during the performed experiments. For example, cylinder pressure changes do not have to be calculated by the model, they can be introduced there from an external file. In this case, the model can calculate all the other thermodynamic load parameters for the real cycle.

The proposed model, regardless of the mode of operation, based on available data without additional experimental tests on the data, enables the calculation of thermodynamic cylinder load parameters as a function of the crank angle (Fig. 3). The basic changes generated by the computational program are:

- cylinder pressure changes;
- cylinder temperature changes;
- cylinder volume changes;
- changes in the (total) heat flux removed through the surface of the cylinder, piston head, cylinder head, changes in DF, CH_4, C_2H_6, C_3H_8 combustion;

Table 2. Range of conditions for which the model was verified and a synthetic summary of verification results

Engine operation type	verification parameters				maximum error		
	n	M_{ob}	P_{max}	U_{on}	instantaneous		in cycle
	RPM	Nm	bar	%	bar	%	%
compression and expansion	750 3400	0	27 42	0	1,1	5	6,6
DF operation non-divided charge	2300 3400	50 200	49 92	100	6,5	8	8,4
DF operation divided charge	1500	20-200	36-87	100	3,7	6	6,8
DF+CNG operation non-divided charge	3400	50-200	35-48	80	6,5	11	6,1
DF+CNG operation divided charge	1500	50 150	40 72	16 80	8	15	6,2

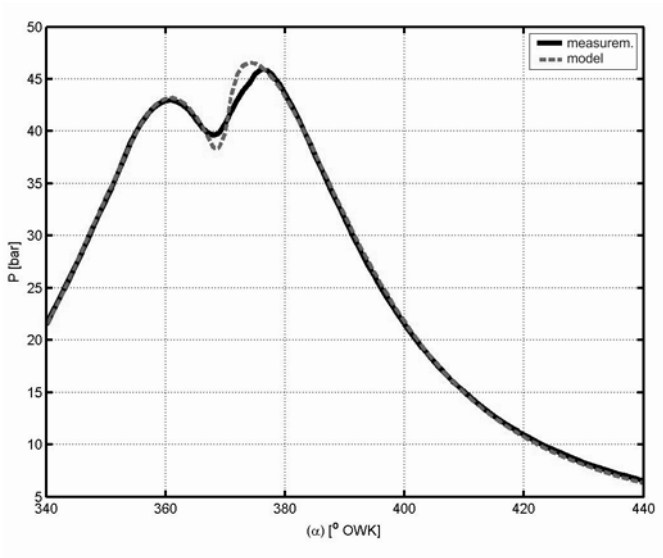

Fig. 2. Comparison of model computation results for dual-fuel operation with the mean recorded pressure changes in the combustion chamber for: n = 1500 (RPM), U_{on} = 70 (%) – divided charge

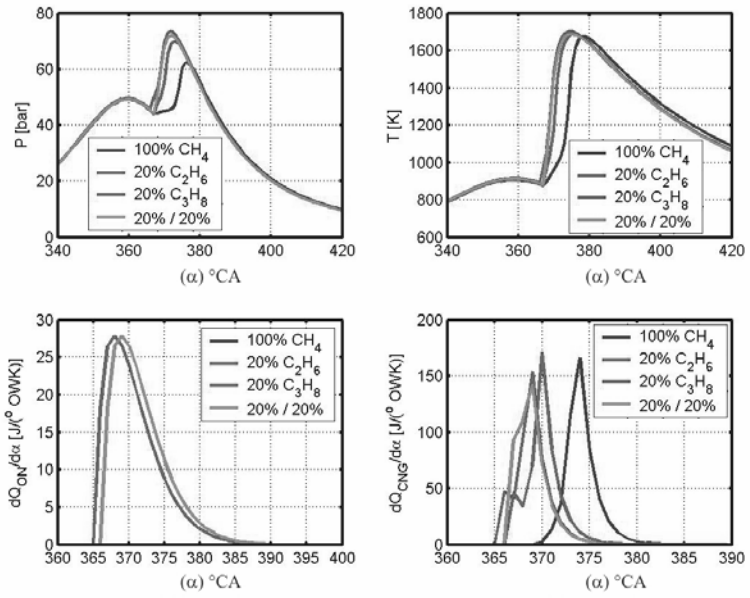

Fig. 3. Example results of simulation computations using the model, for different methane contents relative to combustible gas admixtures and a constant diesel fuel percentage U_{on} = 30%

The model uses around fifteen submodels to determine these relationships, which compute a number of additional parameters in real time. The results of these partial computations can also be used in the testing procedure in inference. The most important additional parameters computed by the program include:

- excess air coefficients for the cylinder mixture (λ_{ON}, λ_{CNG}, λ_A);
- ignition delay angle (and combustion start angle);
- mean temperatures (of the load, cylinder, piston head, cylinder head);
- specific heat changes (for CO_2, CO, N, O_2, H_2, H_2O, CH_4, C_2H_6, C_3H_8);
- changes in the moles in the cylinder (for CO_2, O_2, H_2O, CH_4, C_2H_6, C_3H_8, DF) – under the adopted combustion model.

Different parameters and synthetic characteristics can be calculated based on the generated changes, including:

- *P*/*V* characteristics;
- phase portrait of the engine;
- mean indicated pressure;
- indicated power and thermal efficiency of the engine.

6. Summary

The developed zero-dimensional simulation model of the working cycle of a dual-fuel engine fed with gaseous fuel with an initiating diesel fuel charge includes the compression, combustion and expansion phases. The main characteristics of the proposed model are:

- application of an original approach to modeling liquid fuel injection, based on normal distribution;
- application of a new Assanis correlation for describing diesel fuel self-ignition delay in the presence of gas;
- application of chemical reaction kinetics equations for describing the course of combustion for combustible components of the gas/air mixture in the zero-dimensional dual-fuel engine model;
- implementation of a self-consistency procedure in modeling heat exchange and the effect of exhaust recirculation, which allows a high degree of model completeness to be maintained;
- inclusion of both a single liquid fuel injection and the possibility of performing computations for a divided charge.

The basic advantages of this model include:

- a high degree of completeness, which permits autonomous computational program operation, using only the basic technical data of the tested engine;
- a wide spectrum of model applicability;
- relatively fast computational time, which enables model use on standard PC computers;
- a modular structure, ensuring easy modification and expandability of the model. The model can be developed by adding new submodels or changing the existing ones when more accurate correlations are available, without the need to change the main program;
- the model was subjected to thorough verification in several stages, which ensures its high reliability.

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