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MODELLING OF THE THERMAL CYCLE OF SI ENGINE FUELLED BY LIQUID AND GASEOUS FUEL

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Summary. Results of numerical analysis of methane and gasoline combustion in SI engine are presented in the paper. Work parameters of engine fuelled by gasoline and methane lean mixtures (excess air factor equals $\lambda = 1.8$) are compared.

Keywords: combustion engine, modelling, lean mixture, NO_x, CO_2.

INTRODUCTION

In recent years, air quality has become a particularly severe problem in many countries. Growing concern with exhaust emissions from internal combustion engines has resulted in the implementation of strict emission regulations in many industrial areas such as the Europe and United States. Therefore, how to reduce hazardous emissions and greenhouse gases from engines has now become a research focus.

Improving fuel economy, using a fuel with higher hydrogen to carbon ratio (H/C) or using a renewable fuel can all reduce CO_2 (greenhouse gas) emissions from engines [1]. Natural gas, which is primarily composed of methane (≥ 95% vol. – [2]), is regarded as one of the most promising alternative fuels due to its interesting chemical properties with high H/C ratio and high research octane number (about 120 – [3]). Also, natural gas has relatively wide flammability limits (0.59-1.97 for pure methane – [4]) [1]. The exhaust of natural gas has characteristics such as low NO_x and low CO_2, natural gas has high potential as a clean energy source [5]. CO_2 and NO_x emissions of natural gas engines can be reduced by more than 20% compared with gasoline engines at equal power [1], [6], [7].

Combustion engine researches aimed at reducing of toxic components of exhaust (lean mixtures, alternative fuels) are carried out for many years in the Institute of Internal Combustion Engines and Control Engineering. Carried out are both experimental works and three-dimensional modelling of combustion process, using KIVA-3V and AVL FIRE programs [8-19].

The paper aims at analysis of the methane-air lean mixture combustion in SI engine and comparison of engine work parameters and NO (main component of NO_x) and CO_2 emission with work parameters of engine powered by gasoline as traditional fuel.

NUMERICAL ANALYSIS OBJECT

The engine model was prepared according to the test engine data. The test engine was designed as the modified single-cylinder, high-pressure S320ER engine, which has been rebuilt in order to apply multipoint spark ignition. The application of multipoint spark ignition in the test engine allowed to fuel the engine with lean mixtures of liquid and gaseous fuels of air excess factor $\lambda \leq 2.0$, higher values of air excess factor were not possible to achieve [17], [18]. The main engine parameters are presented in Table 1. The location of spark plugs in the engine head is presented in Figure 1.

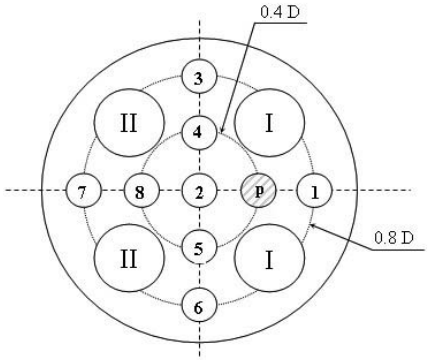
Table 1. Main engine parameters

Engine capacity	1810 cm^3
Number of cylinders	1
Cylinder alignment	horizontal
Cylinder diameter	120 mm
Crank throw	80 mm
Crankshaft length	275 mm
Piston stroke	160 mm
Compression ratio	8.5
Rotational speed	1000 rev/min
Number of spark plugs	8

The numerical modelling was performed in KIVA-3V code [20]. The software is one of the more advanced numerical models, currently used to simulate the combustion process in piston engines. Program allows calculation of three-dimensional flow in the engine chambers of any geometry, including the effects of turbulence and heat exchange with the wall. KIVA-3V is an example of a three-dimensional field model. Combustion process (way and velocity the propagation of flame front) is the result of solving the basic conservation equations, and other dependencies that determine the flow field in the region of the front, the course of chemical reactions of combustion and instantaneous thermodynamic state of the medium. The model is based on solving the equations of conservation of mass, momentum, energy and quantities of ingredients, which describe the unsteady, three-dimensional flow field of chemical reaction (combustion). These equations are three-dimensional Navier-Stokes equations for the compressible fluid mixture. The model calculates separately the thermodynamic state of exhaust gases and the thermodynamic state of the unburned charge. The model calculates the temperature of each area and the average temperature.

The geometric mesh (Figure 1 b)) describing the combustion chamber of the test engine was generated in the pre-processor of KIVA-3V package.

a) b)



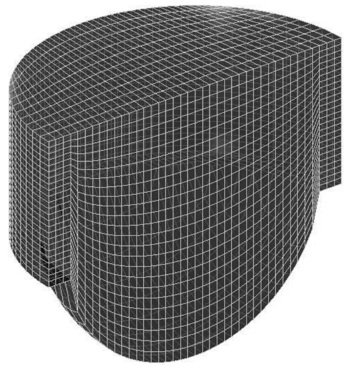


Fig. 1. *Spark plug location legend in engine head, (I – inlet valves, II – outlet valves, D – cylinder bore, p – pressure sensor) - Fig. a) and geometric mesh in cartesian co-ordinate system – Fig. b)*

COMBUSTION MODELLING

The simulation of combustion process was performed for lean mixtures of gaseous fuel (methane) and gasoline with air, for air excess factor values λ = 1.8 for central spark plug no 2 (Fig. 1 a). The ignition advance angle was set to 12° CA before TDC for both fuels. Those parameters were derived from earlier numerical and experimental research [17], [18], where the main criterion in selecting the optimum value of ignition advance angle was the indicated work and exhaust emissions.

The chemical reaction of fuel combustion model in KIVA-3V takes into account one global reaction of fuel oxidation, three reactions of kinetics NO formation, according to extended Zeldowicz mechanism and six reactions of chemical equilibrium for intermediate species [21], [22]:

Methane

$$CH_4 + 2O_2 \Rightarrow CO_2 + 2H_2O,$$
$$O + N_2 \Leftrightarrow N + NO,$$
$$N + O_2 \Leftrightarrow O + NO,$$
$$N + OH \Leftrightarrow H + NO.$$

Gasoline

$$4C_8H_{17} + 49O_2 \Rightarrow 32CO_2 + 34H_2O,$$
$$O + N_2 \Leftrightarrow N + NO,$$
$$O_2 + N \Leftrightarrow O + NO,$$
$$N + OH \Leftrightarrow H + NO.$$

The reactions of chemical equilibrium are [21], [22]:

$$H_2 \Leftrightarrow 2H,$$
$$O_2 \Leftrightarrow 2O,$$
$$N_2 \Leftrightarrow 2N,$$
$$O_2 + H_2 \Leftrightarrow 2OH,$$
$$O_2 + 2H_2O \Leftrightarrow 4OH,$$
$$O_2 + 2CO \Leftrightarrow 2CO_2.$$

The coefficients of kinetic reaction rate of NO formation are necessary to perform the calculations and they were chosen on the basis of the literature studies [23]. However, the turbulence model used in the calculation is *k–ε* model.

The numerical simulation began at 220° CA and finished at 494° CA, which corresponds to experimental engine camshaft phases, which are inlet valves closure and beginning of outlet valves opening.

The results of numerical modelling of thermal cycle of model engine operated on methane and gasoline are compared in this work. The courses of pressure, temperature, NO and CO_2 concentration (averaged values for the volume of combustion chamber) in function of crank angle are presented. The distribution of temperature and nitric oxide concentration in the combustion chamber are presented in graphical form using Tecplot 360 postprocessing software [24].

NUMERICAL ANALYSIS RESULTS

The following figures depict the distribution of temperature and nitric oxide concentration in the combustion chamber for the two analyzed fuels and courses of pressure, temperature, NO and CO_2 concentration (averaged values for the volume of combustion chamber) in function of crank angle. The temperature distribution is presented at crank angle equal 5° after top dead center (TDC). The NO distribution is presented at crank angle corresponding with the maximal concentration of this compound.

The temperature distribution as well as pressure and temperature courses (averaged values for the volume of combustion chamber) in function of crank angle for air excess factor λ=1.8 are depicted in Fig.2 – Fig.4.

a) Gasoline b) Methane

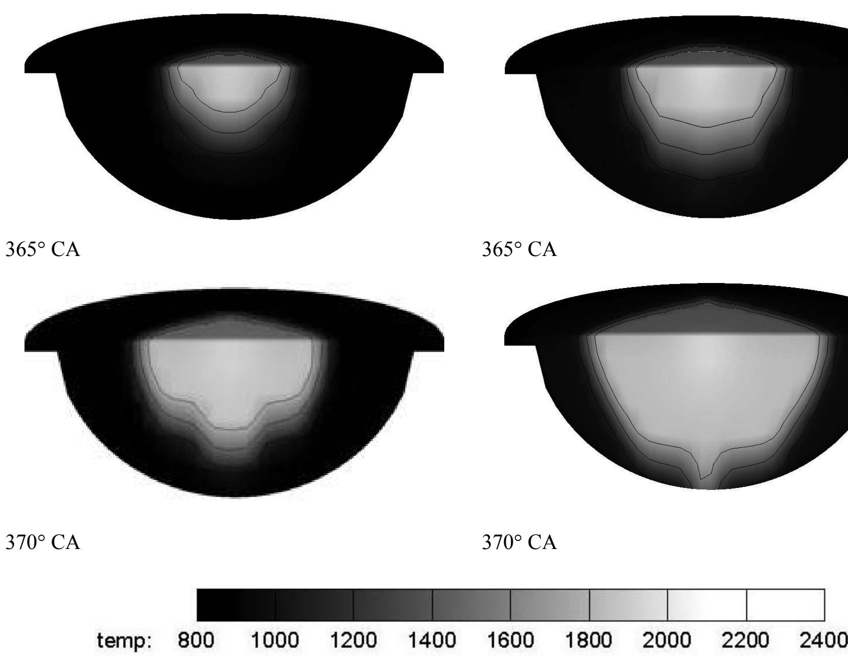


Fig. 2. Temperature distribution for 365° CA and 370° CA

Fig. 2. reveals, that the combustion process was accelerated by using methane as fuel. Greater portion of fuel was burnt and temperatures above 2000 K were reached. Such phenomenon is clearly seen in case of 10°CA after TDC piston position, where charge is burning in a large volume of the combustion chamber. The maximal instantaneous temperature in cylinder reached approx. 2200K for methane, which is approx. 100K more than in case of gasoline burning .

Fig. 3 and 4 depict pressure, heat release rate and temperature courses (averaged values for the volume of combustion chamber) in function of crank angle.

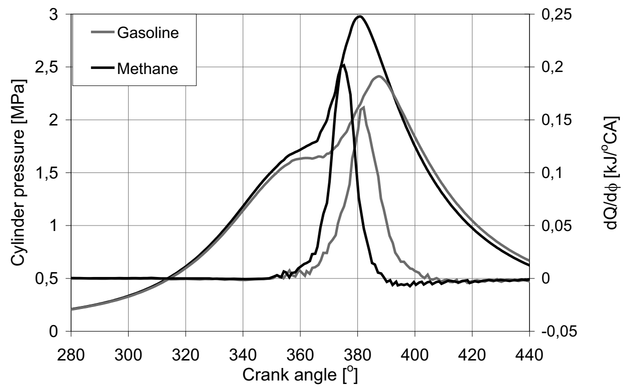


Fig. 3. In cylinder pressure and heat release rate courses for methane and gasoline

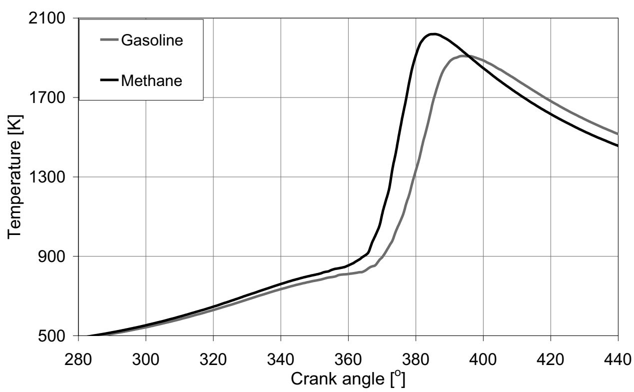


Fig. 4. In cylinder temperature courses for methane and gasoline

The pressure in the cylinder reaches its maximal value equal 2.4 MPa at 387°CA in case of gasoline burning. Using methane as fuel, causes the 24% increase in maximal pressure value to 2.97 MPa. The maximal pressure values occur earlier than in the case of engine powered by gasoline – the difference in crank angle is 6°CA. The combustion duration, defined on the basis of heat release rate, was 27° CA for methane and 34° CA for gasoline. The combustion duration was calculated as the crank angle interval from the spark timing to the end of combustion where the heat release reaches its maximum value – [25]. Above data prove that the combustion process was accelerated for the model engine powered by methane. Also it is clearly seen on a chart depicting the pressure growth speed in the cylinder – Fig. 5. For gasoline, this parameter reaches maximal value of 0.059 MPa/° at 378°CA, however for methane this factor value is almost two times higher and equals 0.122 MPa/° at 372°CA.

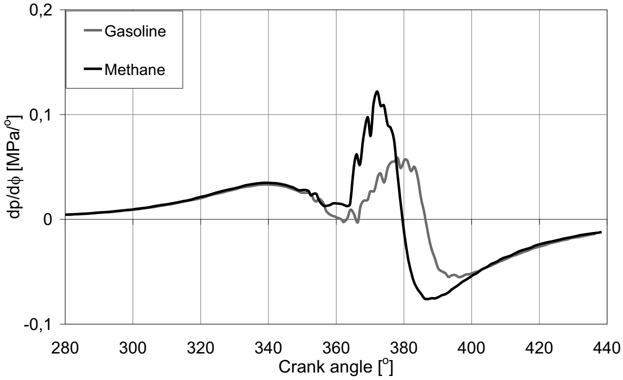


Fig. 5. Pressure growth speed courses in function of crank angle for methane and gasoline

Taking into consideration the above mentioned data, it can be stated that methane is better fuel for using in engine operating on ultra lean mixtures. The combustion process increases, which eliminates one of the main disadvantages of the use of lean mixtures - slow and very long burning.

The next objective of this paper was the numerical analysis of nitric oxide formation. Work of model engine powered by methane showed a decrease in emissions of NO, despite the higher temperature of combustion, cylinder pressure and pressure grow faster. Fig. 6 depicts the nitric oxide distribution in the combustion chamber for both fuels. The figures depict maximal values of nitric oxide concentration and are prepared in the same scale.

a) Gasoline – 422°CA b) Methane – 400°CA

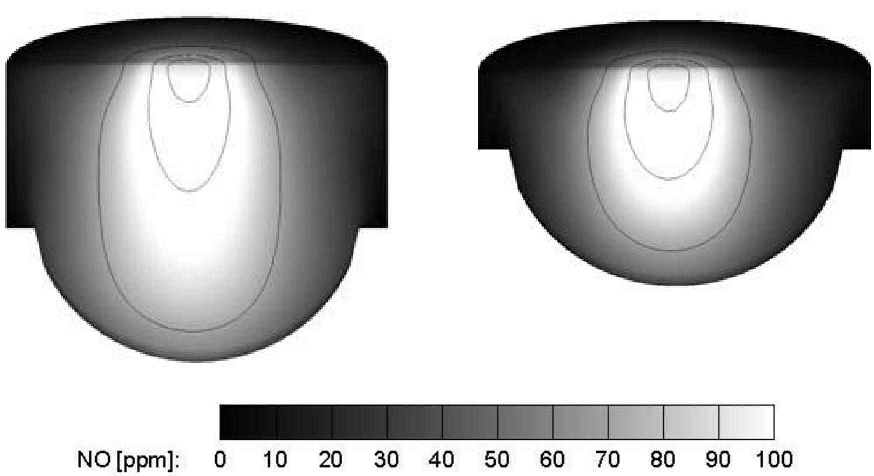


Fig. 6. NO distribution (actual maximum values) for gasoline and methane

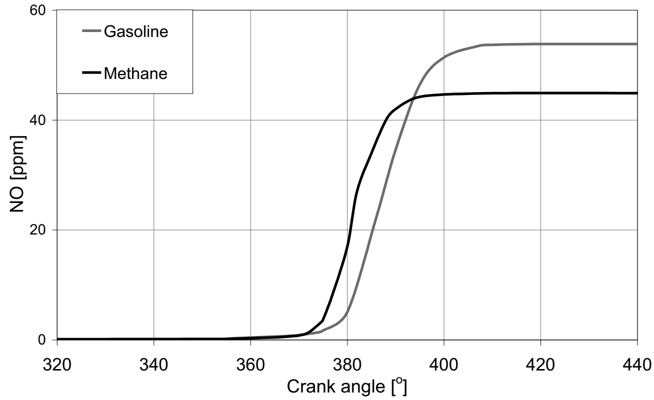


Fig. 7. Variations of NO concentration (mean values for cylinder volume) for methane and gasoline

For model engine fuelled by methane, the nitric oxide concentration (the averaged value for the volume of the combustion chamber – Fig. 7) reached its maximal value equal 45 ppm at 400°CA. However for gasoline as fuel, the NO concentration increased by 20% up to 54 ppm at 422°CA.

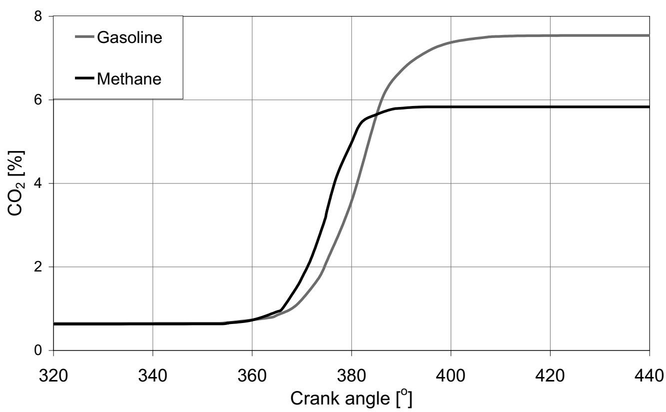


Fig. 8. Variations of CO_2 concentration (mean values for cylinder volume) for methane and gasoline

Above chart (fig.9) shows the variations of CO_2 concentration, which were occurred during modelled engine operation on methane and gasoline as fuels. The carbon dioxide emission values are mean values, calculated for the whole volume of cylinder. The maximal concentration of this compound was 7.5% at 407°CA and was obtained for engine fuelled by gasoline. However for methane as fuel, emission of CO_2 was decreased by about 23% and was 5.8% at 394°CA.

CONCLUSIONS

The results of carried out analysis proved that using methane as fuel significantly improves engine work parameters. The combustion process was significantly accelerated in comparison with engine fuelled with gasoline, combustion duration decreased by more than 20%. The pressure growth speed $dp/d\phi$ was increased more than two times. Fuelling the engine with methane also decreases the concentration of nitric oxide in exhaust gases about 20%. The concentration of carbon dioxide in exhaust gases is also significantly lower in comparison with engine fuelled with gasoline. The difference was about 23%.

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MODELOWANIE OBIEGU CIEPLNEGO SILNIKA ZI ZASILANEGO PALIWEM CIEKŁYM I GAZOWYM

Streszczenie. W artykule przedstawiono wyniki analizy numerycznej procesu spalania metanu oraz benzyny w silniku ZI. Porównano parametry pracy modelu silnika zasilanego ubogimi mieszankami benzyny i metanu, przy współczynniku nadmiaru powietrza $\lambda = 1.8$.

Słowa kluczowe: silnik spalinowy, modelowanie, mieszanka uboga, NO_x, CO_2.