IGNITION OF CNG MIXTURES IN SPARK IGNITION ENGINES

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Abstract

Nowadays in automotive industry, the applying of gaseous fuels and particularly compressed natural gas both in SI and CI engines is more frequent. Application of CNG in the spark ignition internal combustion engines is more real than never before. The paper describes a problem of ignition of CNG mixtures formed in SI engines with direct injection, which requires higher energy supplied by the ignition systems. The main topic of this problem was determined on the basis of the work in NICE project. The paper presents the mathematical model of ignition of CNG mixture and results of calculations. One presents dependencies between different thermal parameters on requiring electric energy. The paper shows also results of experimental results carried out in the caloric chamber done in Cracow University of Technology and chosen results of testing different ignition systems. It was modelled ignition process of the air-methane mixture in the part of combustion chamber with real spark plug. Simulation process was carried out for simple kinetic chemical reaction at initial pressure 20 bar and temperature 900 K. The paper presents calculation CFD model with high density tetrahedral mesh. Transient modelling enables obtaining of the flame spreading in the considered chamber by presenting temperature distribution and kinetic reaction rate for four times. Mass fractions of methane, carbon dioxide and water vapours are shown in dependence on time in the diagrams. The simple combustion model indicated high temperature above 3500 K in the ignition kernel at the beginning of sparking.

Keywords: *transport, combustion engines, fuelling, CNG ignition*

1. Introduction

Recently in automotive industry, the applying of compressed natural gas in the spark ignition internal combustion engines is more real than never before. Many designs of the diesel engines fuelled by the natural gas, which is injected into inlet pipes, are known. Because of the bigger octane number of the natural gas the compression ratio of SI engines can be increased, which takes effect on the increase of the total combustion efficiency. However, in diesel engines the compression ratio has to be decreased for homogeneity of the mixture, which flow into the cylinder. Such mixture cannot initiate the self-ignition in traditional diesel engines because of higher value of octane number. Direct injection of the compressed natural gas besides of the defined dose of the fuel delivered by the injectors requires also high energy supplied by the ignition systems. A natural tendency in the development of the piston engines is an increase of the air pressure in the inlet systems by applying of high charge level by applying of a turbo-charging or mechanical charging. The high engine charging causes a higher compression pressure during the moment of sparking and also the higher temperature. The liquid form of CNG requires the storage temperature near -162° C and much heat to vaporization after direct injection from the cylinder charge.

Naturally aspirated SI engine fuelled by the natural gas has lower value of thermodynamic efficiency than diesel engine. The experiments conducted on SI engine fuelled by CNG with lean homogeneous mixtures ($\lambda \approx 1.4$) show that in many cases it causes a faulty ignition and for this reason the better solution is the concept of the stratified charge with CNG injection during the compression stroke. The world automotive corporations have worked on a new direct injection of

compressed natural gas in spark ignition engines with high charging bigger than in diesel engines. It is assumed that charging ratio reach value about 2.5 which is not met in automotive engines. The increase of the SI engine compression ratio in comparison to the standard engines for naturally aspiration requires the increase of durability of all engine elements. By applying of high supercharging system the pressure of the end of compression process can be higher than 40 bars and is almost 1.5 times bigger than in diesel engines with direct fuel injection. For these reasons the maximum value of combustion pressure can reach 180 bars. The ignition problem of CNG mixtures was considered by many researches in the past [1, 5, 7]. It corresponds also to the theory of ignition of different air-fuel mixtures given by [2-4]. The experiments conducted on SI engine fuelled by CNG with lean homogeneous mixtures ($\lambda \approx 1.4$) show that in many cases it causes a faulty ignition and for this reason the better solution is the concept of the stratified charge with CNG injection during the compression stroke.

2. Ignition problem in SI engines

A bigger ignition temperature for the natural gas (640–670°C) than for gasoline vapours (220°C) is required. For this reason for ignition of the gasoline-air mixture much lower energy is needed than for ignition of CNG-air mixture. However, higher pressure during compression process in the engine (higher compression ratio) causes also higher temperature, which can induce the sparking of the mixture by applying of the high-energy ignition system. The mixture of the fuel and oxygen ignites only above the defined temperature. This temperature is called as the ignition temperature (self-ignition point). It is depended on many internal and external conditions and therefore it is not constant value. Besides that for many gases and vapours there are distinguished two points: lower and higher ignition points (detonation boundary). These two points determine the boundary values where the ignition of the mixture can follow. The flammability of the natural gas is much lower than vapours of gasoline or diesel oil in the same temperature. At higher pressure the sparking is more difficulty than at lower pressure. During the compression stroke the charge near the spark plug can be determined by certain internal energy and turbulence energy. Additional energy given by the spark plug at short time about 2 ms increases the total energy of the mixture in this region. The flammability of the mixture depends on the concentration of the gaseous fuel and turbulence of the charge near the spark plug.

For direct injection of CNG for small loads of the engine in stratified charge mode the burning of the mixture depends on the pressure value at the end of compression stroke and on the relative air-fuel ratio. These dependencies of the CNG burning for different mixture composition and compression ratio are presented in Fig. 1 [3].

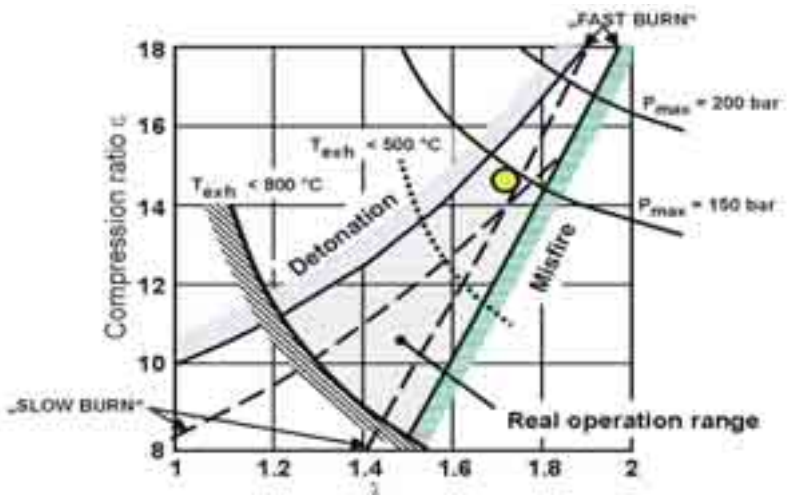


Fig. 1. The range of combustion limits for lean CNG mixture [3]

The burning of CNG mixture can occur in very small range of the compression pressure and lean mixture composition and maximum combustion pressure reaches near 200 bars. For very lean mixtures and higher compression ratios the misfire occurs, on the other hand for rich mixtures and high compression ratios the detonation is observed. Today with new ignition systems with electronic or capacitor discharge the secondary voltage can reach value 40 kV in some microseconds. The higher voltage in the secondary circuit of the transformer and the faster spark rise enable that sparking has occurred when the spark plug is covered by liquid gasoline. With fuelling of the engine by CNG the sparking process should occur in all conditions of the engine loads and speeds.

3. Ignition process in CNG mixtures

The main target of the ignition system is to deliver to the charge certain energy in order to increase the temperature which will begin the decomposition of the hydrocarbon bonds and next the kinetic reaction in the combustion process. The secondary circuit of the ignition coil delivers to the electrodes of the spark plug energy E_2 about 60 mJ. The ignition of a stoichiometric air-fuel mixture ($\lambda = 1$) requires approximately 0.2 mJ of spark energy. Very rich or lean mixtures can require as much as 3 mJ. Most coils are capable of storing 60 to 120 mJ of energy. Spark duration for a coil storage ignition runs between 1 and 3 ms [3].

Power is lost across the distributor gap, via leakage of the wires insulators and any in-line resistances such as the wires and resistors inside the spark plugs themselves. The biggest losses of energy are caused by heat transfer to the electrodes that causes the cooling of the ionization arc and by the radiation of the ionization kernel. Experimental and theoretical studies divide the spark ignition into three phases: breakdown, arc and glow discharge. They all have particular electrical properties. The plasma of temperature above 6000 K and diameter equal the diameter of the electrodes causes a shock pressure wave in several microseconds.

At a very early stage a cylindrical channel of ionization about 40 μm in diameter develops, together with a pressure jump and a rapid temperature rise. Since the ratio between the initial temperature of the mixture and the temperature of the spark channel is much smaller than unity, the diameter d of the cylindrical channel is given approximately by the following expression [7]:

$$d = 2\left[(\gamma - 1)\frac{E_{bd}}{\gamma \cdot \pi \cdot h \cdot p}\right]^{1/2}, \tag{1}$$

where: γ is ratio of the specific heats, h is the spark plug gap and p pressure. E_{bd} represents the breakdown energy to produce the plasma kernel.

After a forming of the plasma between the electrodes the heat source $\dot{q}_e$ of the mixture medium can be calculated directly from the electrical current in the secondary coil circuit I, which changes during time [8]:

$$\dot{q}_e = \sigma \frac{I^2}{\left(\int_0^R 2\pi r \sigma(r,z)dr\right)^2}, \tag{2}$$

where r and z are the coordinates of the ionization volume.

4. Experimental test of ignition systems

Experimental test with ignition of CNG mixtures with different excess air ratios were proved in the laboratory of the Chair of Internal Combustion Engines in CUT. Both currents and voltages both in the primary and secondary circuits of ignition coil were measured by special high voltage probe were measured in order to compare thermal efficiency of considered ignition systems. The

experiments were carried out for CNG mixtures and also in the chamber filled with nitrogen and clean air. The tests indicated a higher electric energy in the secondary circuit of the ignition coil for nitrogen than for air. An example of course of current and voltage measured for the secondary circuit of the ignition coil is shown in Fig. 2 in the CNG mixture at air excess ratio λ=1.4 and at initial pressure 30 bar. It is observed a temporary peak of high voltage during several microseconds reached value about 30 kV and sudden increase of the current above 600 mA.

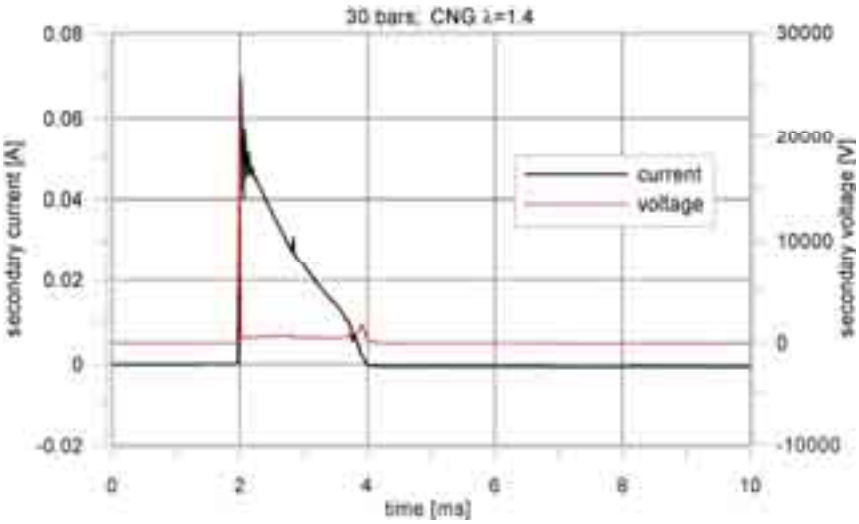


Fig. 2. Course of current and voltage in secondary coil circuit of ignition system

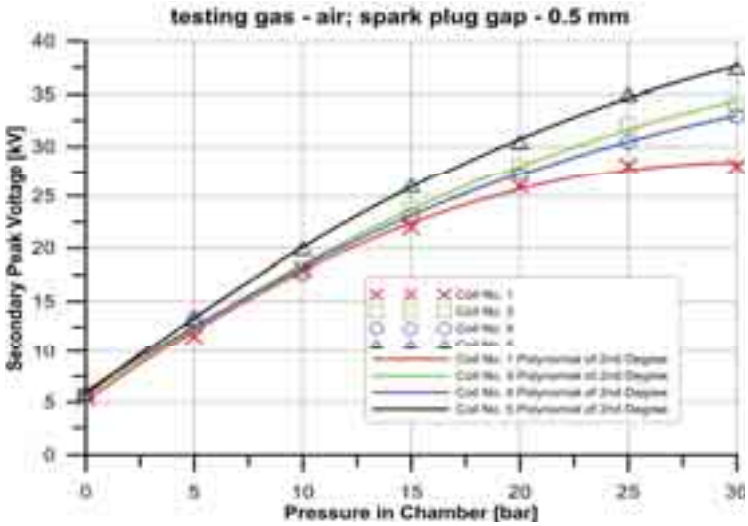


Fig. 3. Peak voltage in secondary circuit for different ignition coils in dependence of air pressure

A linear decrease of the secondary current is observed during 4 ms. The peak of high voltage depends on the initial pressure in the combustion chamber. For example a variation of the peaks of high voltage in the different ignition coils in dependence on initial pressure in the combustion chamber is shown in Fig. 3. With increasing of initial pressure the ignition (jump of spark) requires higher voltage in the secondary circuit in the ignition coil. For the charge with initial pressure about 30 bar the ignition coil should give output of voltage above 35 kV. Despite of higher voltage at higher initial pressure the ignition system delivers lower electric energy with increasing of gas pressure in the combustion chamber. Fig. 4 presents courses of energy in the secondary circuits of considered ignition systems but only for chamber filled with air. Most of the ignition systems can deliver above 60 mJ, however, at initial pressure about 25 bar they gave only from 25 to 55 mJ of electric energy.

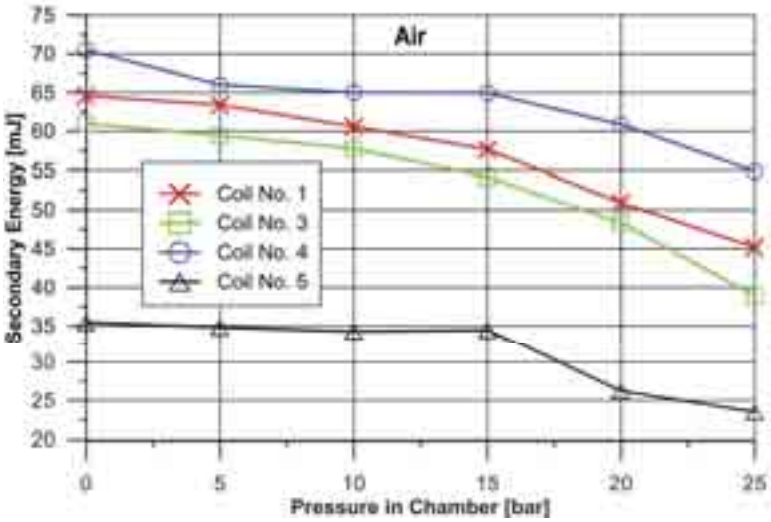


Fig. 4. Electric energy in secondary circuit in dependence of air pressure for different ignition coils

5. Computational ignition model

Ignition of CNG mixture requires adequate value of air excess ratio according to initial pressure in the combustion chamber as shown in Fig. 1. Initiation of ignition process can be simply now observed by simulation of chemical and thermodynamic process in CFD programs such as Fluent, Star-CD, Phoenics, Kiva etc. These programs enable a using of precise geometry of combustion chamber with the spark plug. In this work a part of the combustion chamber with a lower part of the spark plug was modelled. The geometry of the combustion chamber with the real dimensions of the central and side electrodes is shown in Fig. 5a. The geometry was created in CAD system and imported to pre-processor of CFD program. The simulation chamber with radius 15 mm has 10 mm height at piston position at sparking initiation.

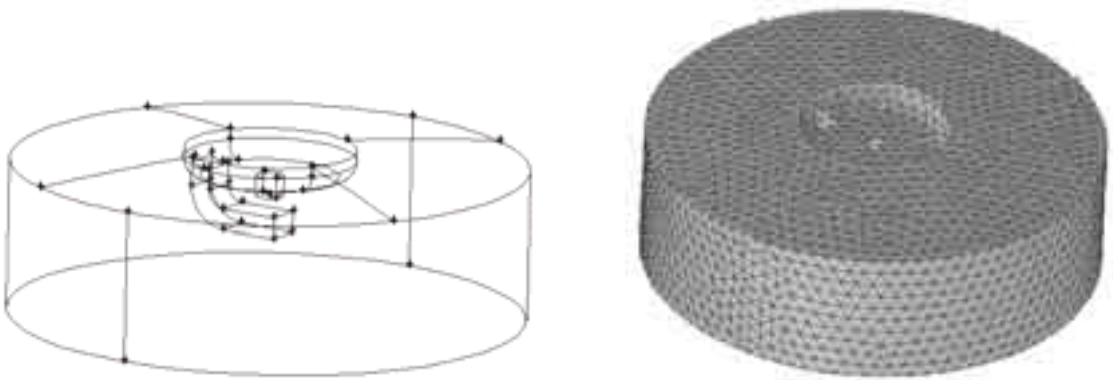


Fig. 5. Simulation model: a) geometry of combustion chamber, b) mesh of combustion chamber volume

The mesh of the considered chamber model was created by using only tetrahedral elements with shortest edges near electrodes 0.3 mm and 0.5 mm for the rest volume of the combustion chamber. It was assumed 1 mm spark gap between electrodes and the initial radius of the spark cylinder 0.25 mm with delivered total ignition energy 60 mJ during 1 ms. The mesh contains above 45000 tetrahedral elements. Because natural gas contains almost 94% of methane, therefore this component was only considered in the chemical reaction. An influence of ionization of the methane-air mixture was omitted and only simple chemical reaction was taken into account in the form:

$$CH_4 + 2O_2 \Rightarrow CO_2 + 2\,H_2O. \tag{3}$$

The simple chemistry allows carrying out a calculation of the thermo-chemical processes in this volume with so big mesh density in several hours. The kinetic reaction was modelled by using Aarhenius reaction at temperature T for the charge with gas constant R in the following form:

$$k_f = A_f T^{n_f} e^{-\frac{E_A}{RT}}, \tag{4}$$

where:
A_f – constant of reaction (A_f= 2.119·10^{11}),
E_A – activation of energy for one mol of reactants or products (E_A = 2.027·10^8 J/kmol),
n – exponent of temperature T (n = 0).

Formation of nitrogen oxides was not modelled and five chemical components such as CH_4, O_2, CO_2, H_2O and N_2 were only considered.

6. Simulation results of CNG mixture ignition

The constant step of calculations equal 10 μs was assumed with maximum 20 iterations per time step. According to the gas parameters at beginning in the combustion chamber obtained from 0-D model the initial pressure was assumed as 20 bar with initial temperature equal 900 K. Mass fraction of methane amounted 0.055, which corresponds to λ = 1. Turbulence κ - ε model was taken into account with laminar-finite rate of combustion.

Chosen results of CFD calculations are presented below for the first period of initiation of combustion by sparking during 1.3 ms, causing almost spherical flame propagation inside the combustion chamber. Fig. 6 presents consumption of methane during this period. About 0.2 of total methane mass was consumed during 1.3 ms in the considered volume. Oxidation of methane takes part on a big participation of the water vapours in the exhaust gases, much higher than during oxidation of gasoline. Variation of mass fraction of water in the charge during ignition process is presented also in Fig. 6. During oxidation of methane the amount of carbon dioxide is growing and mass fraction of this species increases with the growing of the flame radius (Fig. 7). After 1.3 ms from the beginning of ignition 3% of total mass considers CO_2. Mean temperature of the charge in the considered volume in a function of time is presented in Fig.9. During this small period mean temperature increases to above 1200 K (increment above 300 K).

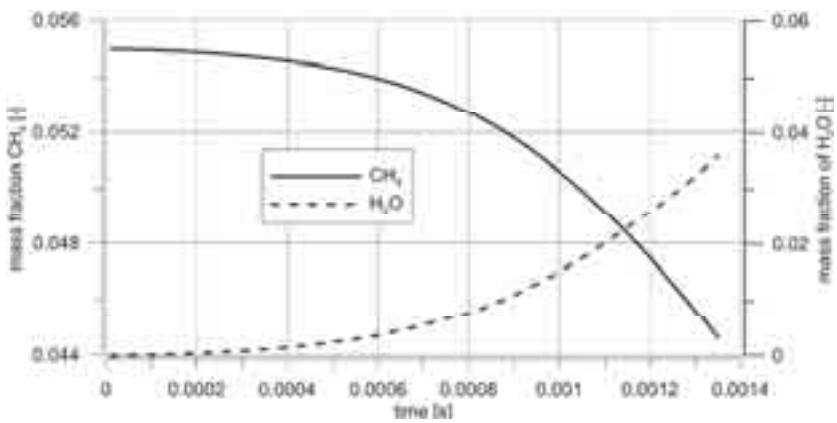


Fig. 6. Decreasing of methane mass fraction in combustion chamber during ignition process

Distribution of temperature in the charge inside the combustion chamber in the cross section going by the cylinder axis along the electrodes is shown in Fig. 9 and 10 for four times at: 100 μs, 700 μs, 1 ms and 1.3 ms, respectively. It is seen an influence of the side electrode on the flame propagation in the first period of ignition. At the beginning of the sparking temperature in the ignition kernel reaches temperature above 3500 K and after 1.3 ms maximum temperature is only

2800 K. Distribution of temperature is depended on kinetic reaction of methane oxidation and turbulent diffusion. The model does not take into account the initial gas motion in the chamber and therefore a spherical shape of the flame was obtained from simulation.

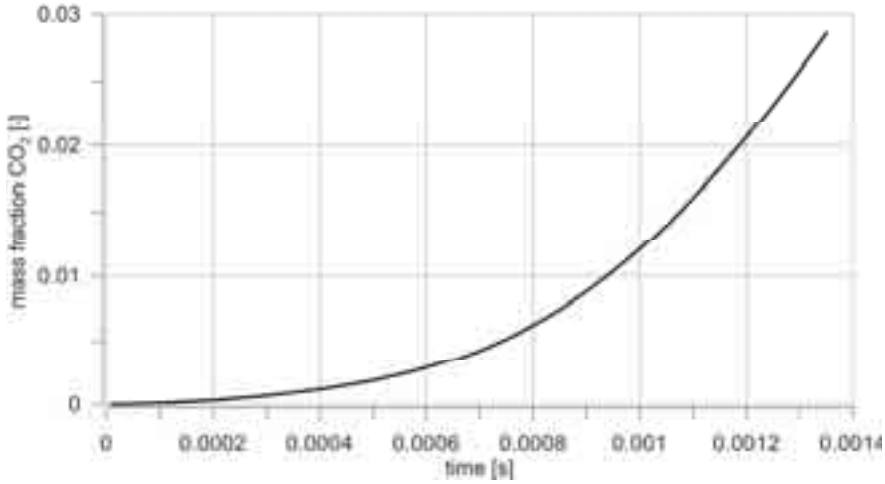


Fig. 7. Increasing of carbon dioxide mass fraction during ignition process

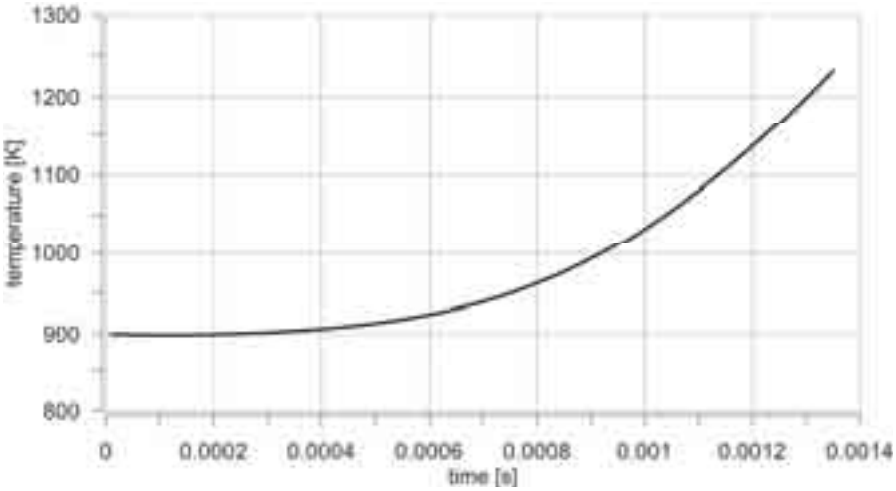
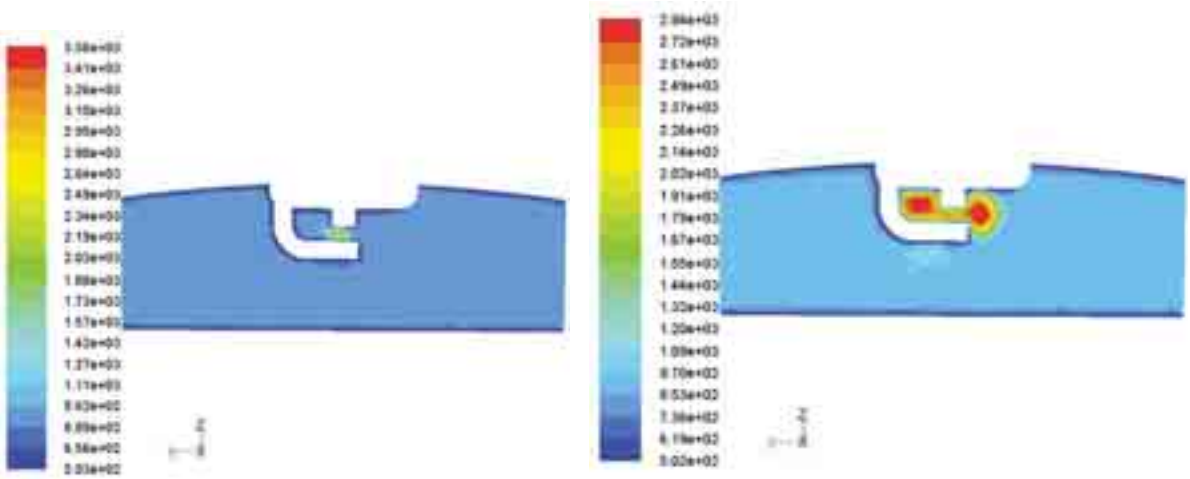
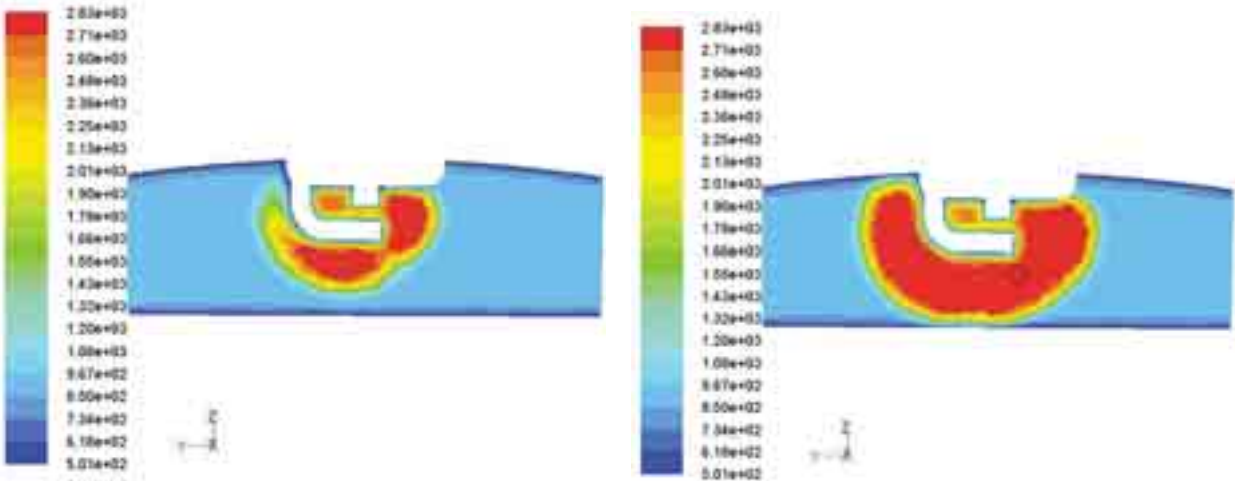


Fig. 8. Variation of mean charge temperature in combustion chamber during ignition process



a) 100 µs b) 700 µs

Fig. 9. Temperature contours in combustion chamber at: a) 100 µs, b) 700 µs



a) 1 ms b) 1.3 ms

Fig. 10. Temperature contours in combustion chamber at: a) 2 ms, b) 1.3 ms

7. Conclusions

The charged CNG spark ignition engines should be equipped with the high energy ignition system above 60 mJ given during period above 1 ms.

Higher pressure of the air-methane mixture required also higher electric energy in the secondary circuit of the ignition coil and maximum high voltage peak increases with the charge pressure.

Determination of thermal charge parameters during ignition requires a detailed chemical reaction model with ionization

Laminar-finite rate combustion model considered in simulation gives a spherical movement of the flame and influence of the side electrode on direction of flame propagation is observed.

The next step of simulation of CNG-air mixture should include ionization effects as a result of current field in the electrodes' gap influencing on development of temperature.

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