THEORETICAL AND EXPERIMENTAL STUDY OF THE IGNITION PROCESS IN CNG DIRECT INJECTION SI ENGINES

Władysław Mitianiec, Tomasz Zioło, Marcin Noga

Cracow University of Technology
Al. Jana Pawla II 37, 31-864 Krakow, Poland
tel: +48 12 6283692, fax: +48 12 6283690
e-mail: wmitanie@usk.pk.edu.pl

Abstract

The future requirements of CO_2 emission decreasing in ICE by applying of new fuels leads also to a design of new engines with direct injection of the compressed natural gas in internal combustion engines with high compression ratio. Because of higher temperature of CNG ignition the SI engines have more effective ignition system than conventional engines. The gas motion, turbulence, charge temperature, spark gap, electrical resistance of the gas and obviously electrical energy of the coil have a big influence on the ignition and burning process in the combustion chamber. The paper includes theoretical and experimental investigations of ignition process in the high charged SI engines with direct CNG injection and in the caloric chamber. The influence of the "tumble" and "swirl" on the sparking is shown by modelling of this process. The paper includes also the thermal efficiency of the ignition process with given values of radiation, conductive and ionization losses.
The experimental and mostly simulation tests showed some important factors which influence on the ignition and combustion process of the compressed natural gas. The thermal efficiency of the ignition system increases linearly with the initial pressure in the chamber. The energy consumed by the charge is only a small part of the total energy delivered by the secondary circuit of the coil. The test of the influence of the gas motion on the ignition and combustion process of methane was carried out in CFD environment.

Keywords: *Transport, SI four-stroke engine, fuelling*

1. Introduction

A much bigger ignition temperature for the natural gas (640 – 670°C) than for gasoline vapours (220°C) is required. For this reason the gasoline-air mixture needs much lower energy for ignition than CNG-air mixture. However higher pressure during compression process in the SI engine with higher compression ratio and higher charging causes also higher temperature that can induce the sparking of the mixture with help of the high-energy ignition system. Because of lower contents of the carbon in the fuel the engines fuelled by the natural gas emit much lower amount of CO_2 causing the less heat effect on the earth. Till now only some laboratory experiments there are conducted with the high-energy ignition system for spark ignition engines with direct CNG injection. There are known the ignition systems for low compressed diesel engines fuelled by CNG by the injection to the inlet pipes.

The flammability of the natural gas is much lower than vapours of gasoline or diesel oil in the same temperature. At higher pressure the sparkover 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 near the spark plug. The flammability of the mixture depends on the concentration of the gaseous fuel and turbulence of the charge near the spark plug. Maximum of the pressure and velocity of its increase in the cylinder for given rotational speed depends on the ignition angle advance before TDC. The beginning of the mixture combustion follows after several crank angle rotation. During this period a certain chemical reactions in the mixture follow to form the radicals that can induce the combustion process. The energy in the spark provides a local rise of the temperature of several thousand Kelvins, which causes any fuel vapour present to be raised above its auto-ignition temperature. The auto-ignition temperature determines the possibility of the break of the hydrocarbon chains and the charge has sufficient internal energy to oxidize the carbon into CO_2 and water in vapour state.

2. Ignition thermal efficiency

Only a small part of the delivered energy from the second circuit is consumed by gaseous medium, which is observed by increase of the temperature ΔT and thus also internal energy E_i. The thermal efficiency of the ignition system is defined as ratio of the increase of internal energy and energy in the secondary circuit of the ignition coil:

$$\eta_{th} = \frac{\Delta E_i}{E_2} = \frac{\Delta E_i \cdot E_1}{E_2 \cdot E_1} = \eta_o \cdot \eta_e \,, \tag{1}$$

where E_1 is the energy in the primary circuit and η_0 is the total efficiency and η_e is the electric efficiency of the ignition system. The increase of the internal energy in volume V with initial pressure p_1 can be determined as follows:

$$\Delta E_i = m \cdot c_v \cdot \Delta T \ . \tag{2}$$

Assuming a constant mass and individual gas constant R, the temperature after ignition can be defined from the gas state equation as follows:

$$T_2 = T_1 \cdot \frac{p_2}{p_1} \,. \tag{3}$$

At small change of the gas temperature from T_1 to T_2 the volumetric specific heat c_v has the same value. In such way it is possible to determine the increase of the internal energy:

$$\Delta E_i = \frac{p_1 \cdot V}{R \cdot T_1} \cdot c_v \cdot \left(T_2 - T_1\right) = \frac{p_1 \cdot V}{R \cdot T_1} \cdot c_v \cdot \left(T_1 \cdot \frac{p_2}{p_1} - T_1\right). \tag{4}$$

After simplification this equation takes the form:

$$\Delta E_i = \frac{V}{R} \cdot c_v \cdot \left(p_2 - p_1\right) = \frac{V}{R} \cdot c_v \cdot \Delta p \,. \tag{5}$$

The increase of the internal energy depends on the sparking volume, gas properties and a pressure increment in this volume. Because of constant volume and known R and c_v the unknown value is only the increment of the pressure Δp.

The experimental tests on five ignition systems with different technical solutions were carried out by measurement of the primary and secondary voltage and current to determine the delivered energies to the charge. The spark plug was located in the small chamber filled by the nitrogen or air with different initial pressure. The variation of the thermal efficiency in dependence of the pressure is shown in Fig.1.

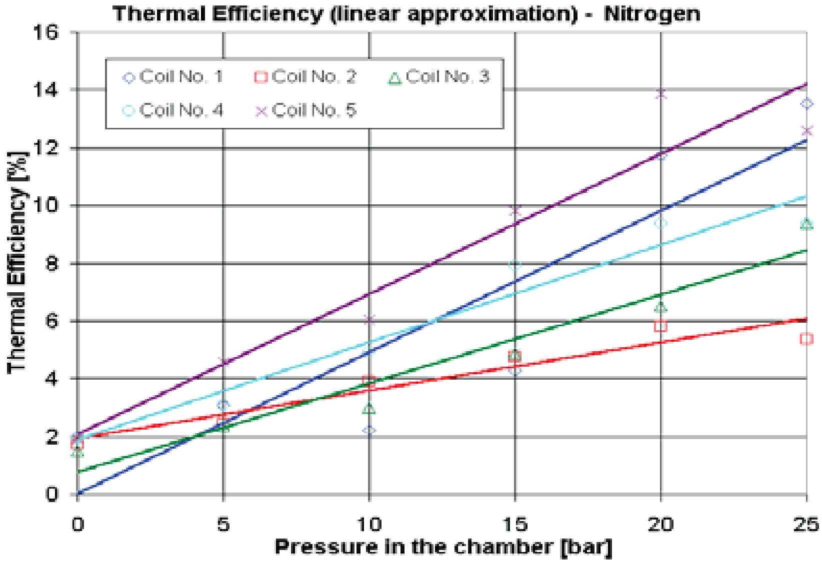

Fig. 1. Thermal efficiency of five tested ignition systems

The higher pressure of the charge requires higher thermal efficiency, thus much more energy should be delivered to the charge. Making use if the experimental tests and the results from the calculations of the heat transfer, radiation, ionization and kinetic energy from the electrodes to the charge the balance of the all energies were determined. The values of the individual energies delivered by the secondary circuit of the ignition system are shown in Fig.2. Only 7% of the total energy is consumed by the charge independently on the initial pressure.

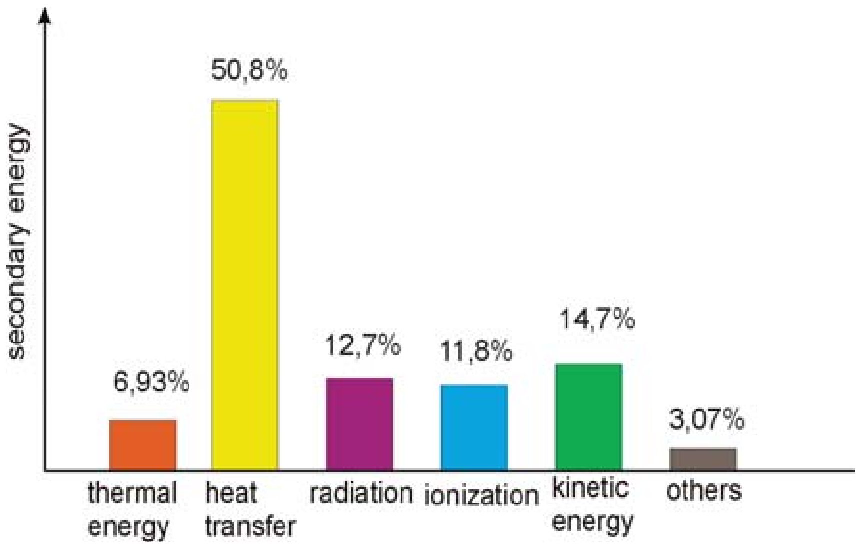

Fig. 2. Balance of the energies delivered by the secondary circuit of the ignition system

3. Combustion in turbocharged CNG engine

At the full load the SI CNG engine requires a big pressure ratio in the turbocharged system with the intercooler. One predicts of the pressure in the inlet system about 2 bars, which needs the change of the engine timing. The sparking of the CNG mixture with the high pressure during compression stroke causes the requirements of higher energy of the ignition system. The simulation of the combustion process was carried out by program KIVA3V [3] on one-cylinder engine with capacity 400 cm3. The mesh of the engine with the flat piston contains 62 838 vertices and 62 811 cells. It was assumed, that the injector had only one nozzle with flow area amounted 2 mm^2 and is shown in Fig.3.

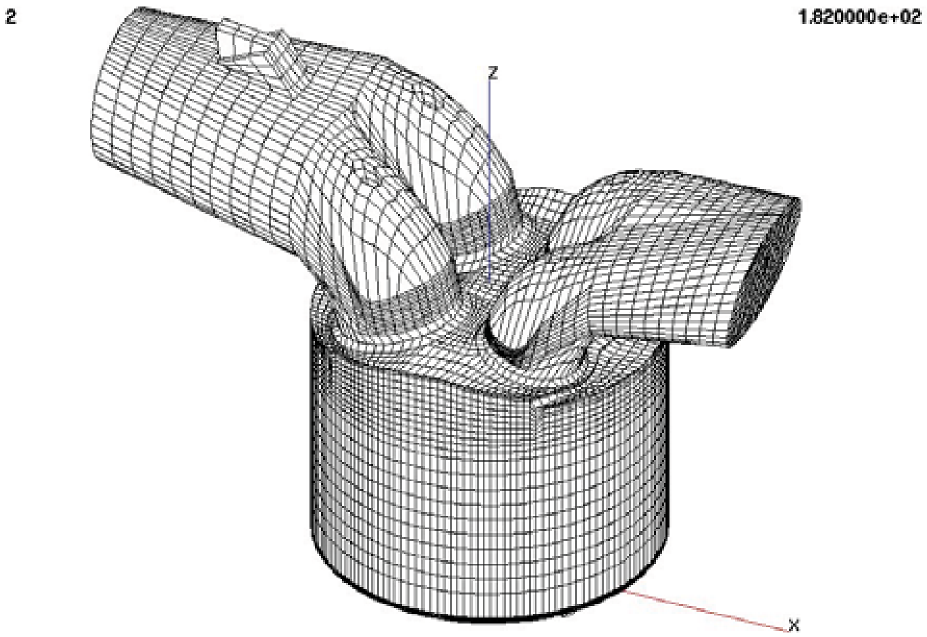

Fig. 3. Mesh of the CNG cylinder

Before the 3-dimensional simulation the additional computer program based on zero-dimensional code enabled the calculation of the working parameters of the naturally aspirated engine fuelled by CNG at different rotational speed and full load. The results of these calculations were taken as the initial parameters: pressure and temperature in the cylinder and gas velocity in the ducts for CFD calculations. Variation of the calculated cylinder pressure and temperature are shown in Fig.4 at 4000 rpm and WOT.

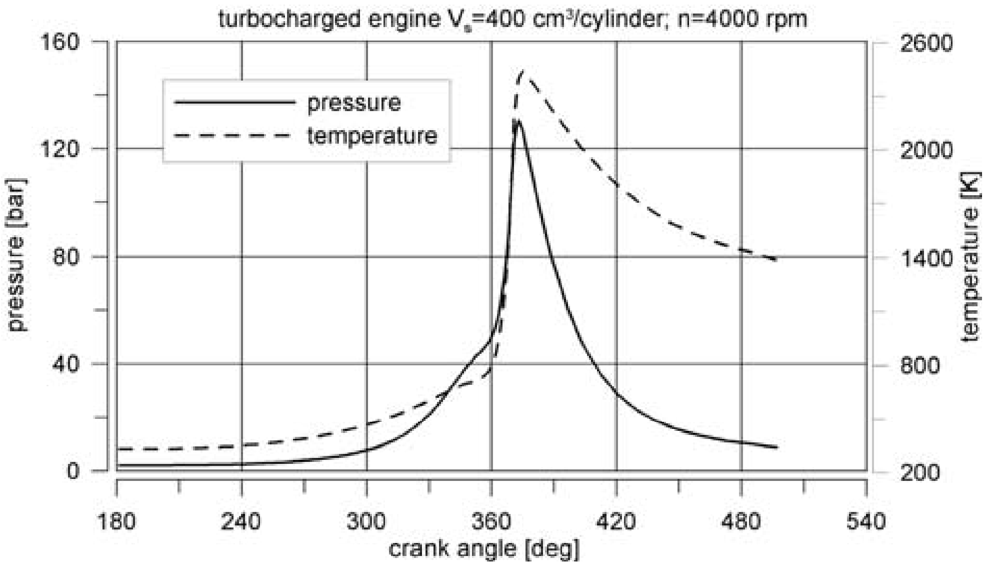

Fig. 4. Pressure and temperature in the cylinder of SI turbocharged engine at 4000 rpm and WOT with the compressor pressure ratio 2,0

The fuel consumption during the combustion process lasted very short only 35 deg of the crank rotation. The CNG dose amounted 0,045 g and mass of the air amounted 0,871 g, which corresponds to λ=1,14 in the premixed charge. The variation of the fuel consumption is shown in Fig.5 and indicated on the full combustion of CNG. The start of ignition was at 8 deg BTDC when the pressure inside the cylinder amounted 42 bars. The beginning of the combustion started at 5 deg BTDC and very quickly the combustion reaction proceeds radial from the spark plug. The temperature in the cylinder reaches maximum value 2750 K and its distribution is shown in Fig.6

at 5 deg CA BTDC and 10 deg ATDC. The higher temperature causes formation of NOx and the mean mass concentration reaches value 3200 ppm at 40 deg CA ATDC (Fig.7).

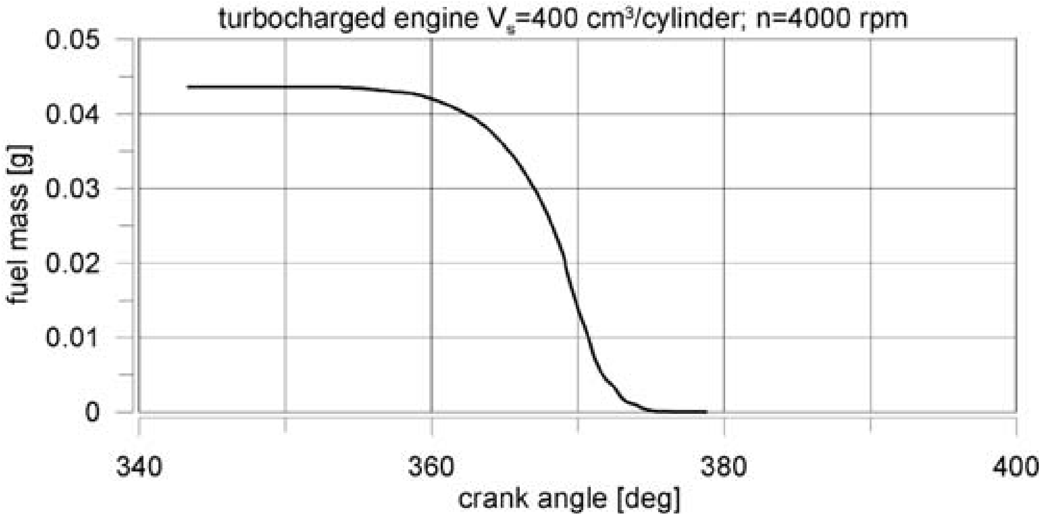

Fig. 5. CNG consumption in the cylinder of SI turbocharged engine at 4000 rpm and WOT with the compressor pressure ratio 2,0

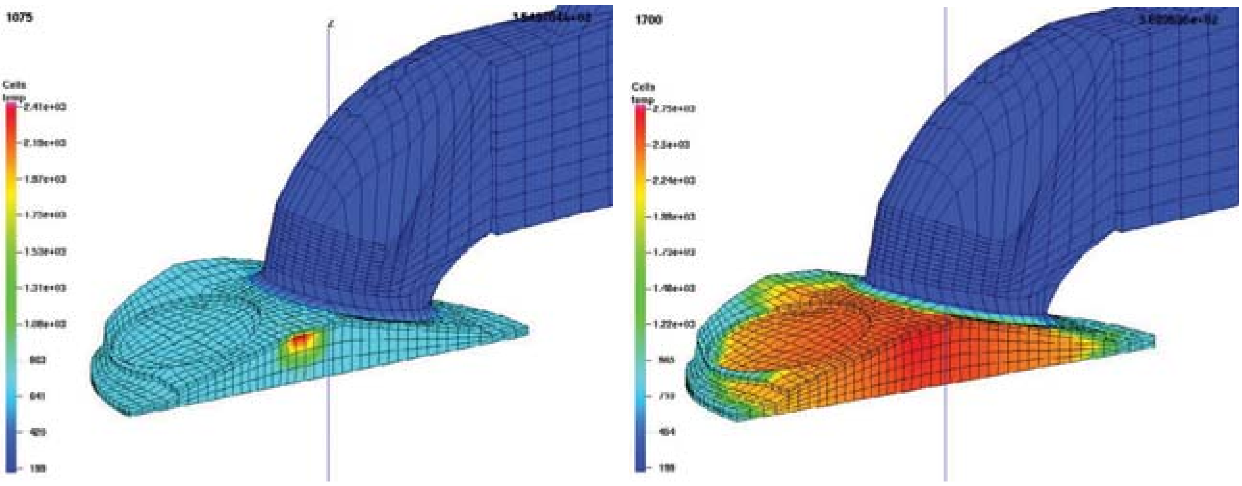

Fig. 6. The charge temperature in the cylinder at 5 deg BTDC and 10 deg ATDC

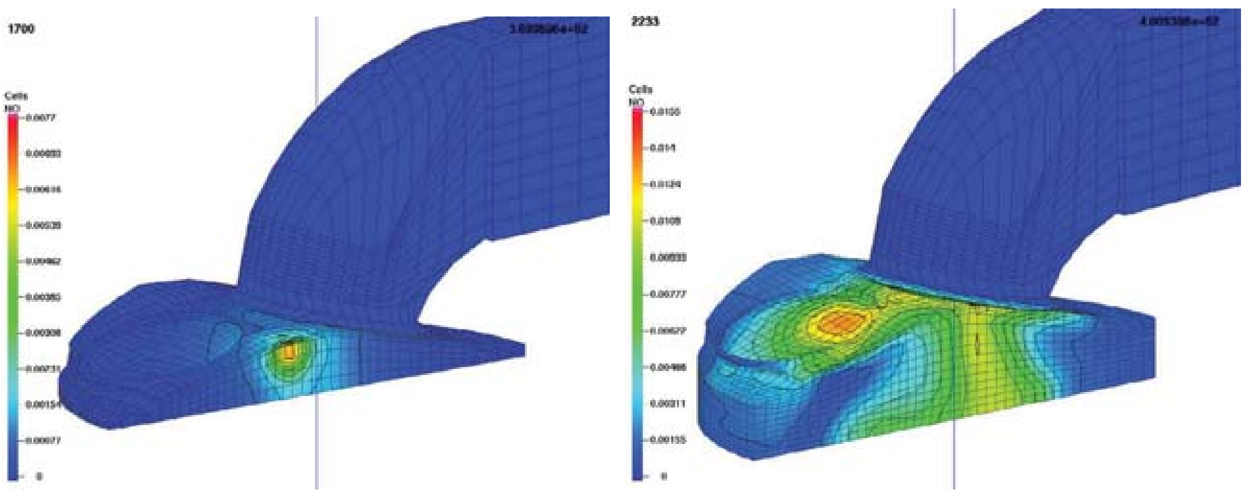

Fig. 7. Formation of NOx *at 10 and 40 deg CA ATDC*

4. Simple chemical reaction scheme

The ignition process of methane can be performed in the caloric chamber filled by the mixture of the defined air-fuel ratio and pressure with given initial velocity of the charge. The propagation of the flame after the ignition is observed by change of the gas density on the boundary flame. The thickness of the flame depends on the mixture composition and for the leaner mixture the width of the flame is wider. In the caloric chamber velocity of the gas is small and for that the simple chemical reaction scheme (SCRS) describe by Spalding [1] can be used. All chemical reactions involve the release or absorption of proportionate amounts of energy. The amounts of energy involved are deducible from the thermodynamic properties of the reactants; they are thus independent of the rates at which the reactions proceed. The SCRS is a "model", which is a simplification of reality and still represents reality in practically important respects, and facilitates thought and computation. The SCRS postulates that combustion does proceed via

$$fuel + oxygen \rightarrow products,$$

without intermediates. This reaction is taken as irreversible and this means that the rate of the reverse reaction is presumed to be very low. The energy released (heat of combustion) is taken as independent of temperature and this implies equality (even constancy) of specific heats of reactants and products. The enthalpy of a mixture of fuel, oxidant and products can be taken as given by:

$$h = c_p \cdot T + H \cdot m_{fu} . \tag{6}$$

Two options are provided, namely:

a) the mixing-controlled reaction-rate option,
b) the kinetically-controlled reaction-rate option.

Experiment shows that reaction rates are often more greatly affected by local turbulence than by chemical-kinetic factors. The eddy-break-up model developed by Spalding [1] rests on the hypothesis that only turbulence and fuel concentration affect the rate.

The first version is described by reaction rate given below:

$$\omega = -\mathrm{const} \cdot \rho \cdot m_{fu} \cdot |\mathrm{grad}\ u|. \tag{7}$$

The second version was takes into account the kinetic energy κ and dissipative kinetic energy ε and the combustion rate is performed as follows:

$$\omega = -\mathrm{const} \cdot \rho \cdot m_{fu} \cdot \varepsilon/\kappa. \tag{8}$$

Both work fairly well; but they are only approximate. If the fuel and oxidant are fully pre-mixed, and the distribution of fuel is uniform, it becomes possible to solve just one equation, termed the "reactedness" (RCTD). The model takes into account the mixture properties, temperature of burned and unburned mixture and is based on the empirical constants implemented in the program code of the special routines. The described simple combustion models are implemented in CFD program Phoenics [9].

5. Simulation of ignition in Schlieren chamber

The initial simulations of the CNG combustion was carried out on the model of the Schlieren chamber used foe the experimental tests. The chamber had the volume equalled 100 cm3 with diameter D=80 mm and width B=20 mm. The initiation of the ignition followed in the centre of the chamber by two thin electrodes. The chamber was filled by natural gas at 5 bars and λ=1,4. The initial temperature of the charge amounted 300 K, so this required much more electrical energy than for firing engine. The simulation was carried out for the same thermodynamic parameters as in the experiment in the Phoenics program for 50 ms in the transient operation. The ignition energy was simulated as additional internal energy in the centre of the combustion chamber. The RCTD model for fully premixed charge was used. The simulation was carried out in the cylindrical coordinates (NX=36, NY=25 and NZ=11).

The constants for the simulation with using of the RCTD model were taken from the experiments on the Schlieren stand. The combustion process in the chamber lasted a long time (above 50 ms), because of absence of the gas motion. The pressure variation is shown in Fig.8. The chemical reaction (RCTD) taking place in the chamber are shown in the Fig.9 for the reaction time 8 and 50 ms. The propagation of the reaction is radial and the thick boundary of the combustion (about 8 mm) is observed because of the lean mixture.

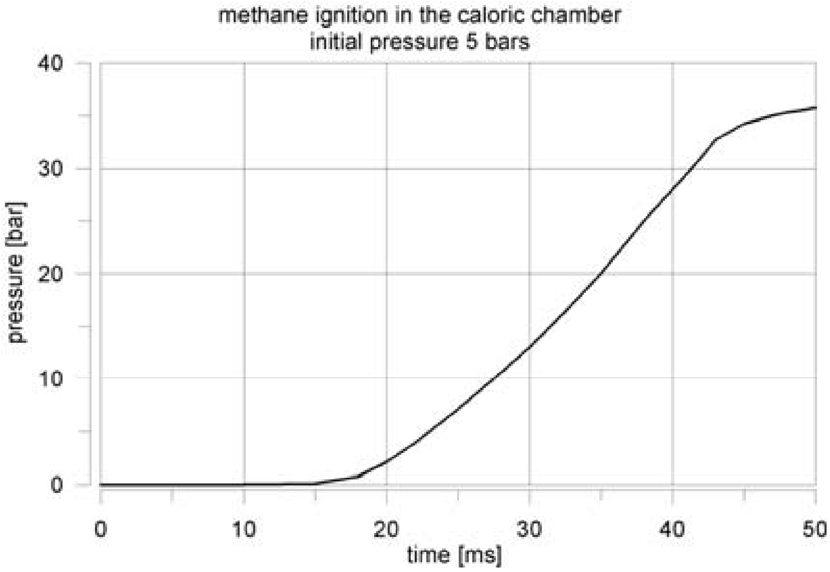

Fig. 8. Increase of pressure in the chamber after ignition

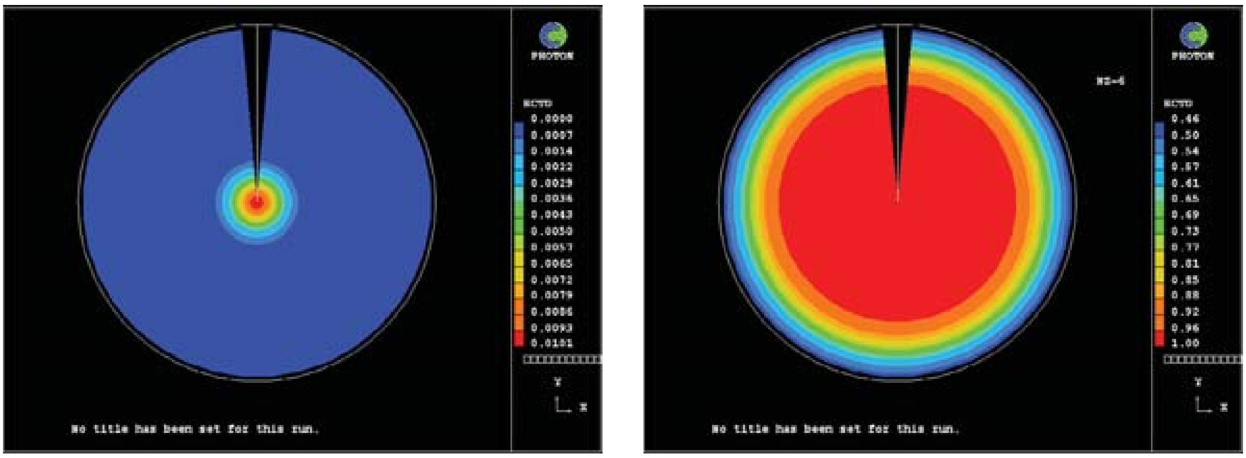

Fig. 9. Propagation of the combustion reaction after 8 ms and 50 ms

Experimental test on the Schlieren stand done by Sendyka and Noga [10] showed also radial propagation of the flame defined by the change of the charge density. Figure 10 shows the films of the flame propagation in the chamber at 7 and 40 ms after start of the ignition. The flame is distorted by touching into the quartz glass in the chamber. The comparison of the simulation and measurement flame propagation shows a good agreement of the results by using RCTD combustion model.

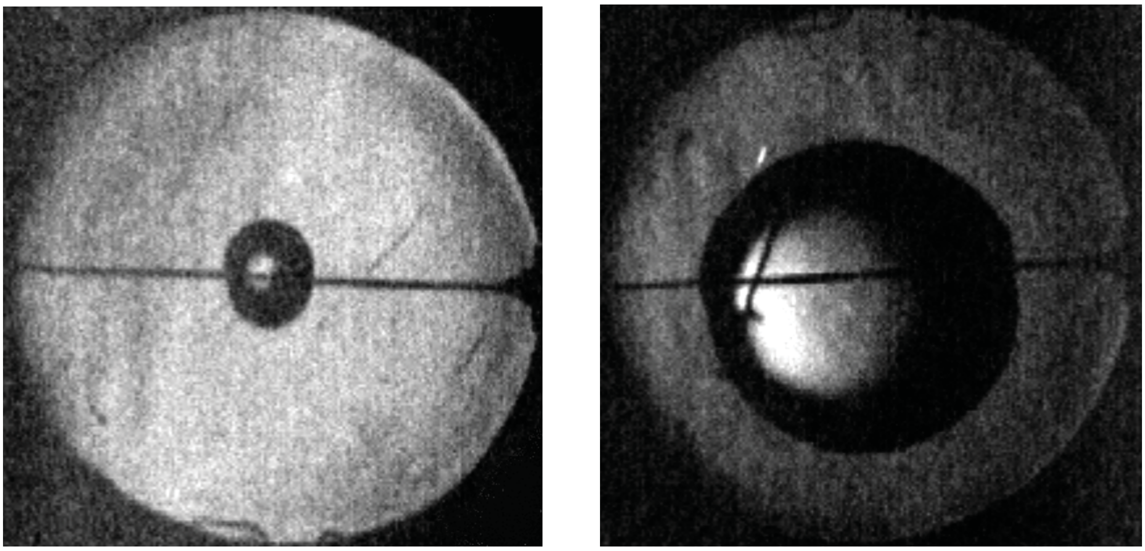

Fig. 10. Schlieren stand – boundary of the flame after 7 ms and 40 ms [10]

6. Tumble and swirl motion

The most important factor influencing on the ignition is the charge motion through the spark plug. Two kinds of motions were considered: swirl and tumble caused by valve and inlet profile, combustion chamber and squish. The combustion process is strongly connected with turbulence of the charge and only small part is the laminar speed of the total combustion velocity. Simulation was carried out in the rectangular space wit central location of the spark plug. The mesh contained 20925 cells with rectangular prism (NX=31, NY=27 and NZ=25) and the calculations were done in transient conditions (500 intervals in time t=5 ms). The spark plug was located in the centre of the calculation space and the object of the electrodes was created in the "*.*stl"* format in CAD system and then transformed in the Phoenics object.

At the first the ignition of CNG was simulated with „initial tumble" ω_y =250 rad/s and p=20 bars. The charge with velocity about of 15 m/s flew through the gap of the spark plug causing the propagation of the flame inside the chamber. The simulation took into account another SCRS model called "the kinetically-controlled reaction-rate". The charge motion is connected with high turbulence and this causes also the higher combustion rate. The distributions of the combustion products in the space are shown in Figures 11 and 12 at 5 and 1,2 ms after start of the ignition, respectively. After short time (about 1 ms) the whole charge is burned in the calculation space.

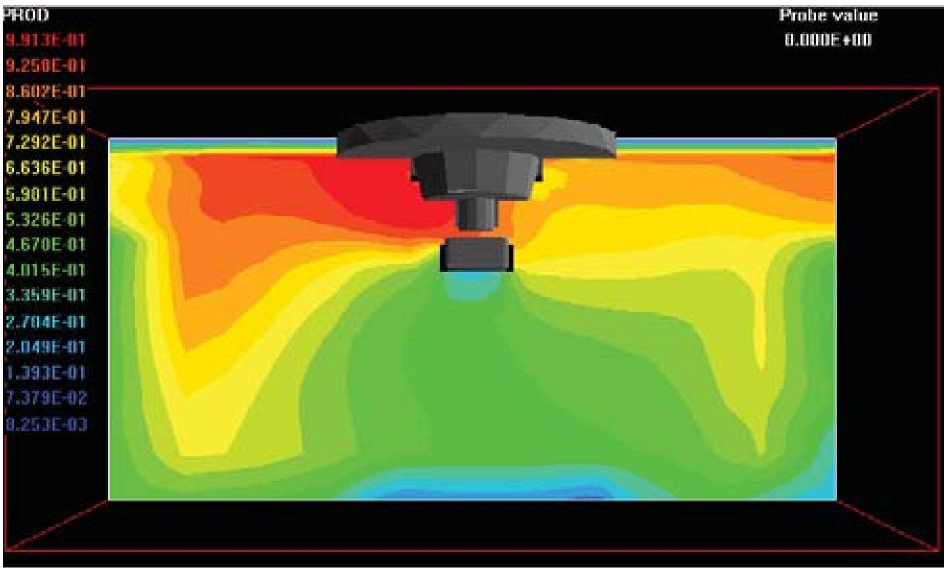

Fig. 11. Distribution of the combustion products with the charge tumble after 0,5 ms

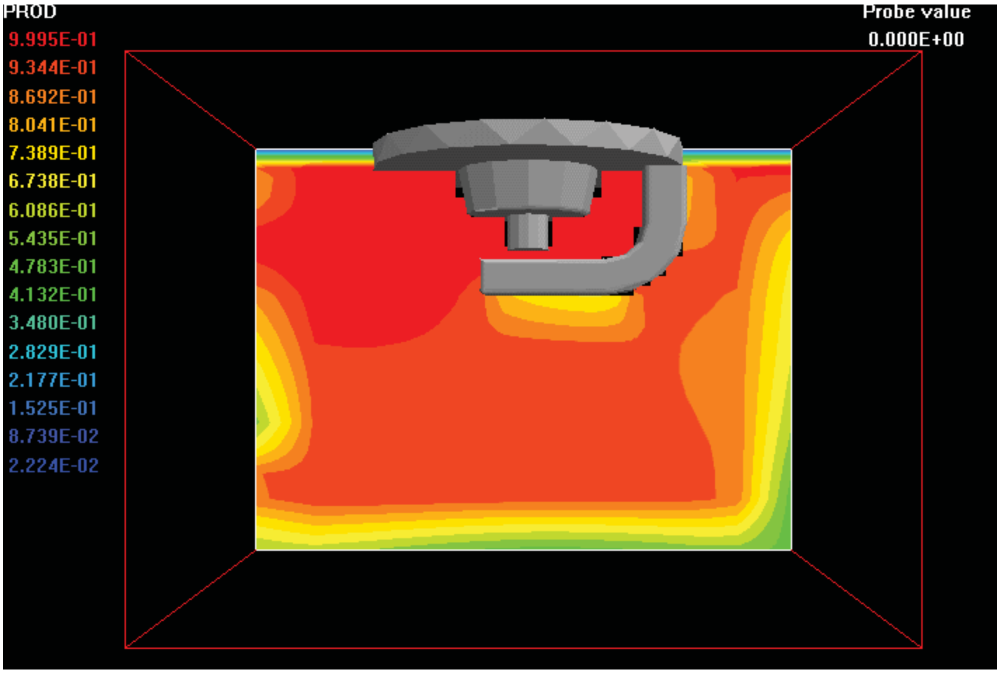

Fig. 12. Distribution of the combustion products with the charge tumble after 1,2 ms

The "tumble" motion of the charge in the space is observed by the distribution of the velocity vectors shown in Fig.13. The higher flow velocity is between in the gap of the spark plug.

The other simulation was carried out for the central swirl around the spark plug with swirl velocity 15 m/s on the mean radius 1,5 cm. The same combustion model was used as in the previous case. In this case the interaction of the electrode shape is seen – the propagation of the flame is faster in the opened site of the electrodes (Fig.14). The swirl in the chamber influences on the irregular propagation on the flame and extends the combustion process. Even after 40 ms the combustion of the methane in not full. Velocity of the gas flow in the spark plug gap is smaller than in the "tumble" case. For this reason the propagation of the combustion products (PROD) and flame is not uniform. Fig.15 shows the distribution of the products in the calculation space after 40 ms from the start of ignition.

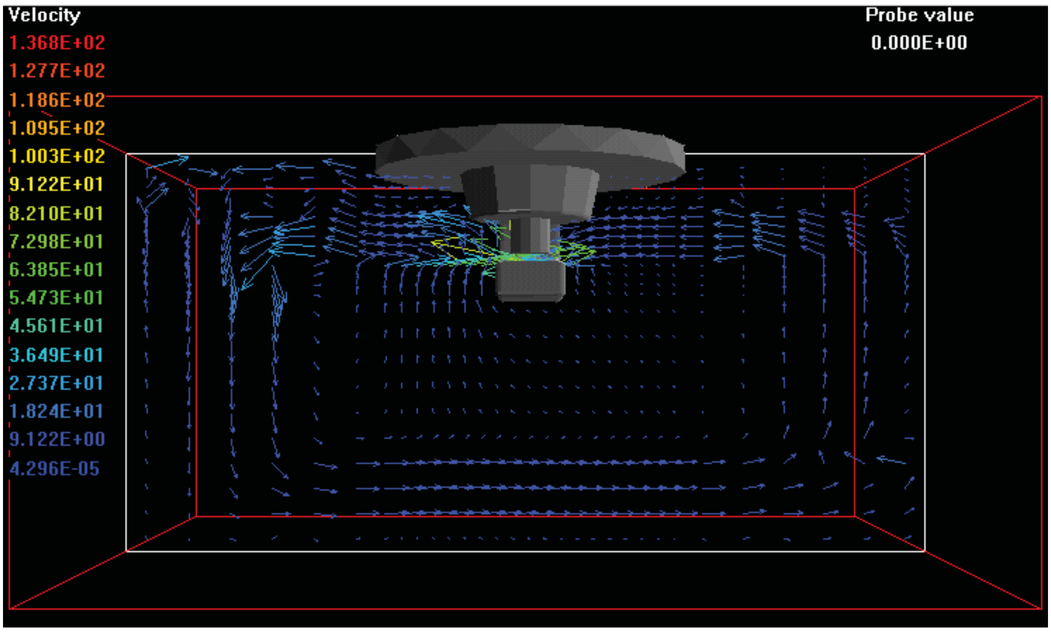

Fig. 13. Velocity vectors of the charge (tumble case) after 1,2 ms

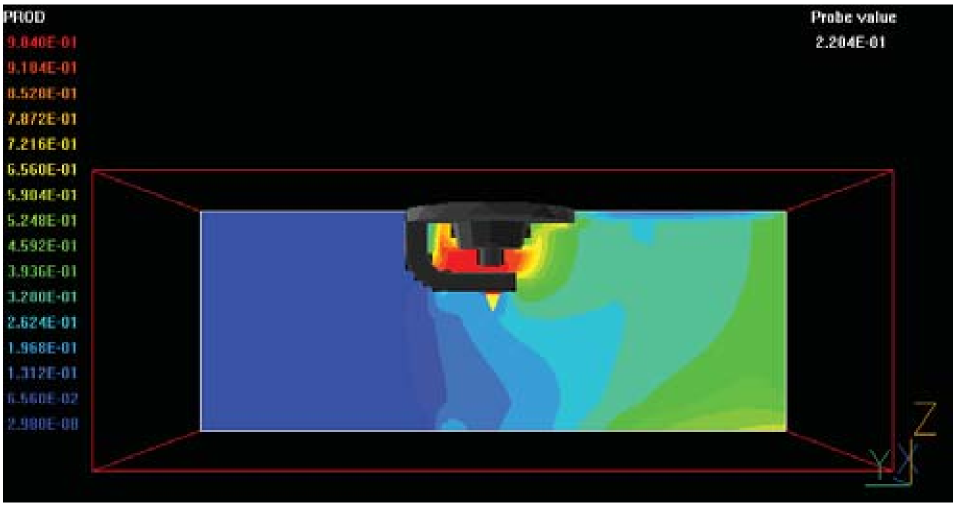

Fig. 14. Distribution of the combustion products with the swirl of the charge after 1,0 ms

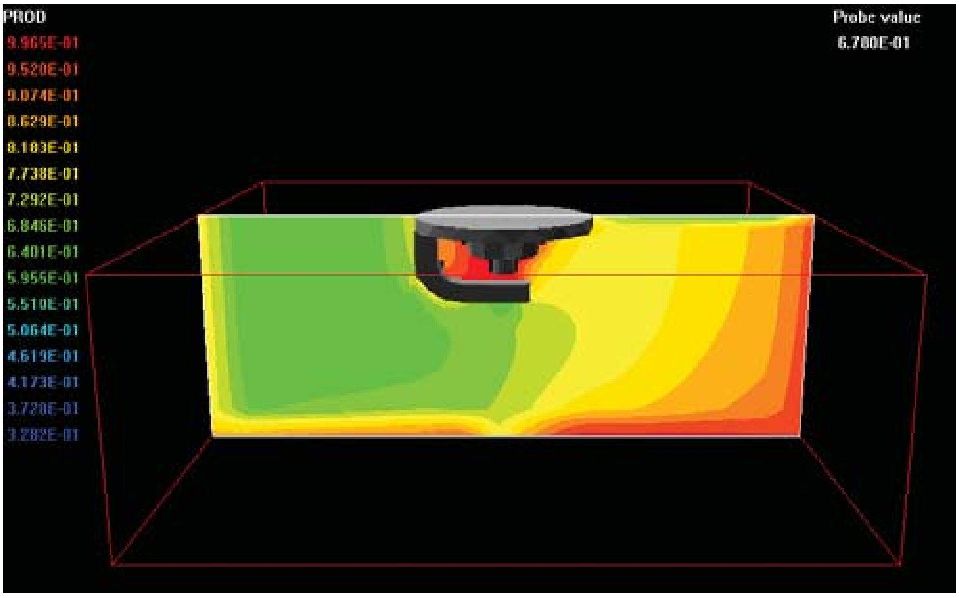

Fig. 15. Distribution of the combustion products with the charge swirl after 4,0 ms

The simulations for two cases showed the big influence of the gas motion in the chamber on the ignition and the combustion process in the chamber. The calculations of the ignition and combustion by use the simple chemical reaction system proved the superiority of the "tumble" case on the ignition process of the methane and also CNG.

7. Conclusions

The experimental and mostly simulation tests showed some important factors which influence on the ignition and combustion process of the compressed natural gas.

1. The thermal efficiency of the ignition system increases linearly with the initial pressure in the chamber. Thus higher pressure requires much more electrical energy.
2. The energy consumed by the charge is only a small part of the total energy delivered by the secondary circuit of the coil.
3. In the high charged CNG SI engine at full load with leaner mixture there is no problem with the ignition and combustion, which lasts only 35 – 40 deg CA. The problem is with the high emission of NOx, however this species can be reduced in TWC reactor.
4. The RCTD model of combustion was verified with the experiment on the Schlieren stand. The propagation of the flame is spherical and its mean velocity amounts 0,4 m/s.

5. The test of the influence of the gas motion on the ignition and combustion process of methane was carried out in CFD environment. The calculations showed the superiority of the tumble motion of the gas. With the swirl the slower combustion process proceeds. Only junction of both motions of the gas can be satisfied for the ignition and combustion of the methane and CNG.

Acknowledgement

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