IGNITION AND COMBUSTION PROCESS IN HIGH CHARGED SI ENGINES WITH DIRECT INJECTION OF CNG

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Abstract

The paper concerns the direct injection in SI internal combustion engines by compressed natural gas in order to decrease the global CO_2. The increase of the working parameters of the spark ignition engines in comparison to the diesel engines would be done by applying a high charge pressure and with direct injection of CNG for two modes with the homogenous charge and stratified charge. Decreasing of NO_x emission is possible by fuelling the engine by lean mixtures (λ = 1.2 - 1.6). Ignition of the mixture at high compression pressure above 40 bars requires applying of a strictly defined dose of gas fuel. Sparking of the high compressed mixture can be done only by high voltage ignition system. The paper describes the total problems of injection, ignition and combustion CNG in high compressed spark ignition engines with simulation results in one-cylinder four-stroke SI engine. There are also compared the test and calculation results in the caloric chamber. The paper gives dependences of compression ratio and air-fuel ratio for lean burning of CNG.

1. Development of IC engines fuelled by natural gas

Recently in automotive industry the applying of compressed natural gas in the spark ignition internal combustion engines is more real than never before. There are known many designs of the diesel engines fuelled by the natural gas, which is injected into inlet pipes. 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. Applying of two-stage turbocharging systems or mechanical chargers can fulfil the higher charging. 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 time bigger than in diesel engines with direct fuel injection. For these reasons the maximum value of combustion pressure can reach 180 bars. The increased compression pressure requires also a big pressure of CNG injection. Till now on the automotive market there are not the gas injectors with sufficient flow rate through the nozzles for the low loads with injection during compression stroke enabling obtaining of the stratified charge and also for full loads at high engine speeds. Naturally aspirated SI engine filled 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.

2. Ignition and combustion of cng

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 that can induce the sparking of the mixture by applying 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 our earth. 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.

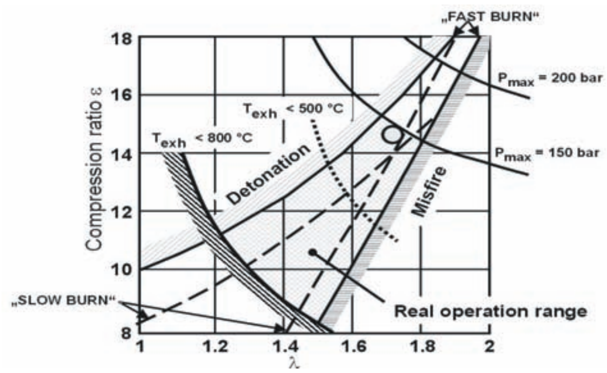

Fig. 1. The range of combustion limits for lean CNG mixture [3]

The flammability of the natural gas is much lower than vapors 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]. 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. During the cold start-up the ignition process of the CNG mixture is much easier than with gasoline mixture because of whole fuel in the gaseous state. 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 every conditions of the engine loads and speeds.

If we assume that the electrical energy E is delivered during period τ to a certain small volume V near spark plug with the temperature of the charge T_1 and pressure p_1 with concentration of CNG fuel adequate to relative air-fuel ratio λ it is possible to calculate the change of the charge temperature in this space. On the basis of the law of gas state and balance of energy the specific internal energy u of the charge in the next step of calculation is defined.

$$u_i = u_{i-1} + dE \,. \tag{1}$$

Delivery of electrical energy to the local volume results on the increase of local internal energy and change of the temperature that can be determined as follows:

$$m \cdot c_v \cdot T_i = m \cdot c_v \cdot T_{i-1} + de \qquad \text{or} \qquad m \cdot c_v \cdot \frac{dT}{dt} = \frac{de}{dt} \,. \tag{2}$$

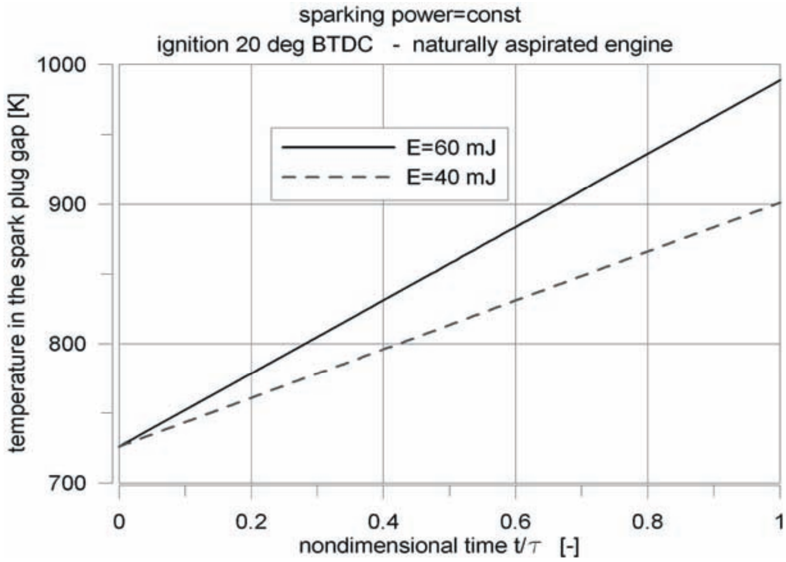

Fig. 2. Increment of the local temperature in the region of the spark plug for two ignition systems with constant sparking power

If the total electrical energy amounts E and duration of sparking lasts τ (1.8 ms) then for the case of constant power of the spark plug the local power is E/τ for whole period τ of the sparking. At assumption of specific volumetric heat c_v as constant at small period τ, the temperature of the local charge is simple to obtain by integration of the equation (2) as function of time t ($t = 0 \,..\, \tau$)

$$T = T_1 + \frac{E}{m \cdot c_v} \frac{t}{\tau} \,. \tag{3}$$

For assumed sparking region with diameter 5 mm and height 2 mm and the given concentration of the air and fuel (CNG) in the mixture the gas constant amounts R=296.9 J/(kg K). The calculated mass of charge in the region amounts 2.92e-7 kg. As shown in the Fig.2 the final temperature in the region is the same for two considered variations of ignition power. If the volume of the sparking region decreases the local temperature will increase, however ignition of the mixture depends on concentration of the fuel in the air.

3. Direct injection of compressed natural gas

Because the natural gas contains many hydrocarbons with changeable concentration of the individual species the heat value of the fuel is not constant. It influences also on the ignition process depending on lower ignition temperature of the fuel and energy induced by secondary circuit of the ignition coil. The future gas engines will be equipped with high pressure direct injection systems. Like the gasoline direct injected engine also the gas engine can operate mainly in two modes, which can create: a homogenous charge for full loads and a stratified charge for part loads. Injection of the gas fuel for full load takes place during the induction stroke after opening of inlet valves and this operation do not require so high injection pressure. For part load the fuel is injected during the compression stroke forming a bigger concentration of fuel near spark plug located in the central axis of the cylinder head. The timing of injection should correlate with the piston position BTDC and engine speed in order to enable the adequate stoichiometric mixture near the spark plug during the ignition. This mode requires higher injection pressure than in the first one. The stratification of the charge depends on the location of the injector and the angles of injection nozzles. The most important problem is to fulfil the dose of fuel especially for the gas injection during compression stroke for high speed operation.

4. Calculation mesh of SI engine with CNG direct injection

The first step of 4-stroke SI engine testing has been done by conducting the simulation of injection in the stratified and homogenous modes and combustion process in order to check the assumed injection, ignition and charging parameters. The Cracow University of Technology will provide tests with direct injection of CNG on one-cylinder motorcycle 4-stroke engines SUZUKI DR-Z400S adopted for this target. This modern engine with capacity of 400 cm^3 and bore/stroke ratio = 90mm/62.6mm and compression ratio ε=12 will be equipped with gas injector located between two inlet channels. The cylinder with a pentroof combustion chamber and 4 valves has the spark plug located in the middle of the cylinder head. The assumed parameters of the gas injection are presented in Table 1.

Table 1. Injection parameters

	Stratified charge	Homogenous charge
Dose of CNG fuel	0.0098 g/cycle	0.045 g/cycle
Beginning of the injection	105 deg CA BTDC	55 deg CA ATDC
Duration of the injection	40 deg CA	120 deg CA
Ignition point	9 deg CA BTDC	9 deg CA BTDC
Cone of fuel jet	50 deg	50 deg
Number of nozzles	1	1

The whole process of the engine work was done in 3- dimensional space by using the program KIVA3V [2]. The source code of the program was modified by author in order to apply it for gas injection, because the original version was released for liquid fuel injection only. The overall geometry of the engine and additionally cross section through the valves for calculations is presented in Fig.3 The mesh 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.

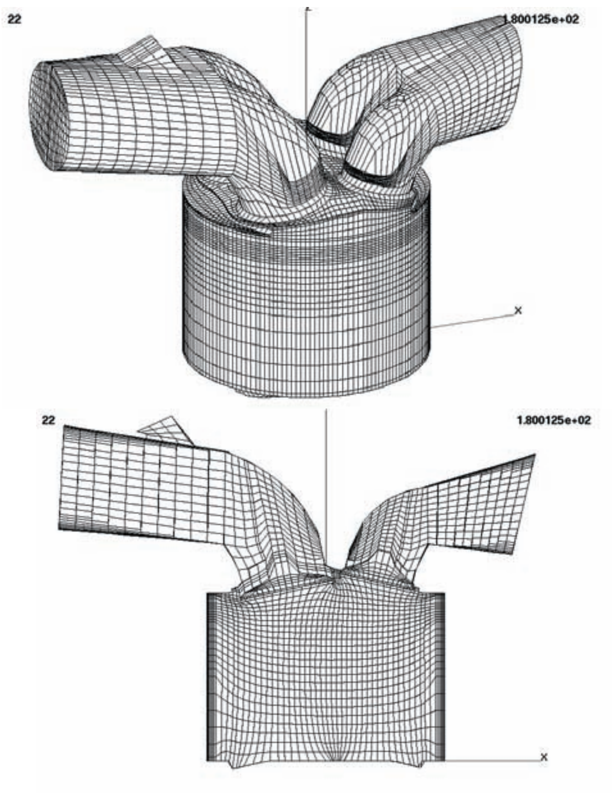

Fig. 3. Calculation mesh of motorcycle engine SUZUKI DR-Z400S

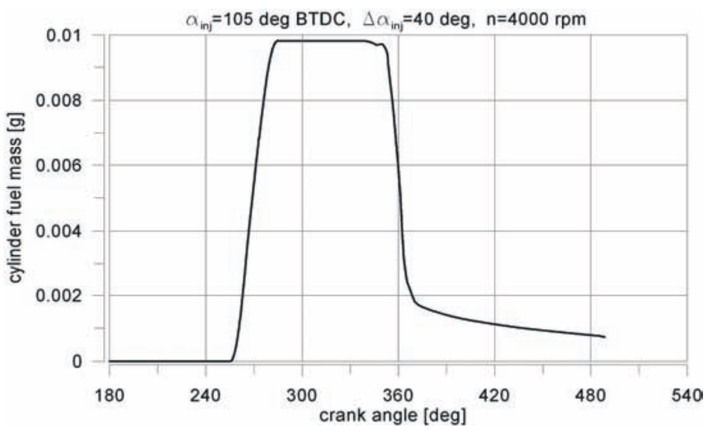

Fig. 4. Mass of CNG in the cylinder with injection dose of fuel 0.0098 g/cycle at 4000 rpm

The injection of CNG during compression stroke was simulated at the injector setting under the angle of 70 deg to the cylinder wall. Variation of the injected CNG to the cylinder and its decrease during combustion process is shown in Fig.4. Because of small dose of fuel the kinetic reactions during the combustion process proceeds very short. The rest of fuel located in the regions near wall burns very slow and this process proceeds until the exhaust valves open. For homogeneous charge the fuel injection takes place during induction stroke and for this case in the simulation it was assumed that all mixture components are distributed with the same concentration.

5. Propagation of fuel jet

Two possibilities of fuel direct injection for charge stratification were analysed: with almost vertical fuel jet (26 deg of injection angle) and parallel fuel jet 70 deg of injection angle. In the first case the fuel moves very early to the piston head (Fig.5) and only small amount reaches the spark plug region. With the parallel direction of the injected fuel stream most of the fuel flows through the combustion space and after reaching of the opposite wall rebounds from it (Fig.6). Next the fuel gas washes the cylinder head and reaches the spark plug.

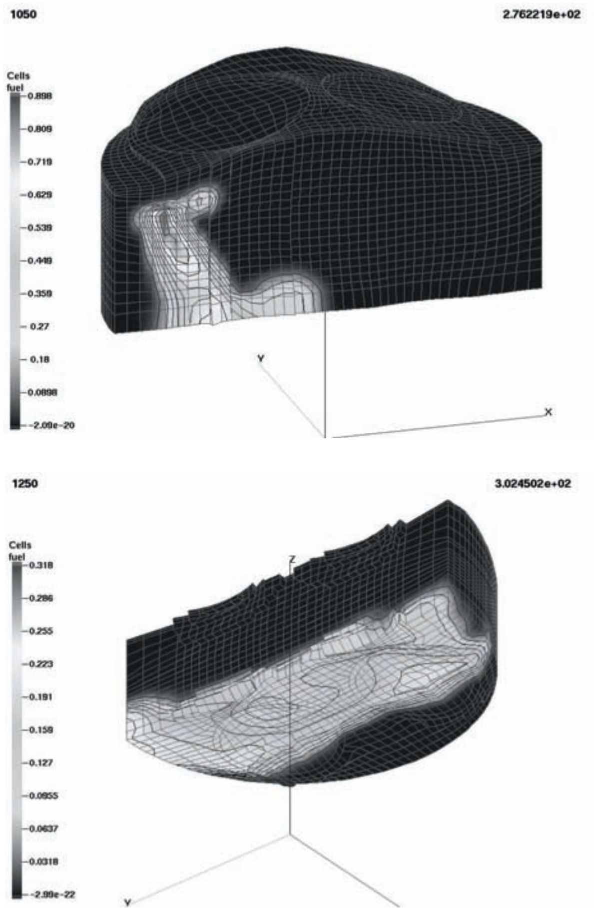

Fig. 5. Distribution of CNG during injection at 84 and 58 deg BTDC with injection angle 26 deg

The CNG injection should start early for higher rotational speeds in order to get the mixture with proper concentration of fuel in the ignition region. In the stratified charge mode the fuel jet can be guided by the wall (by the piston bowl) or guided by the air as an effect of swirl or tumble of the charge caused by induction stroke.

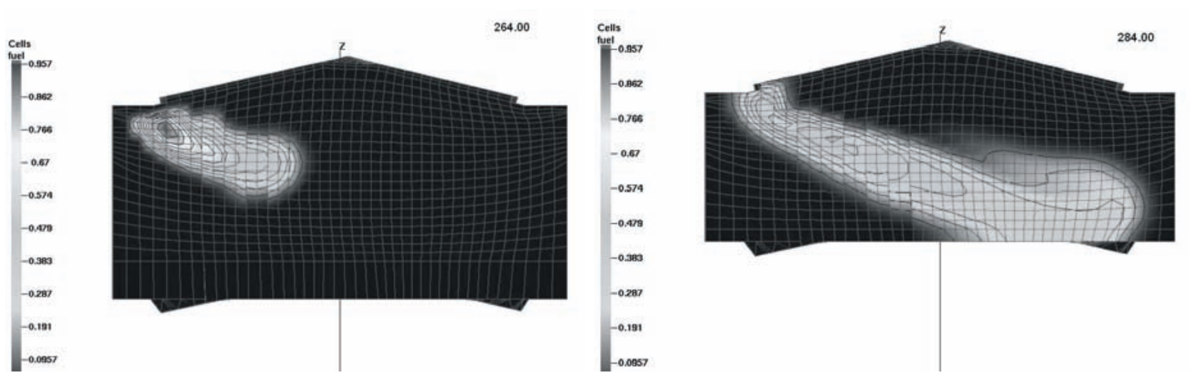

Fig. 6. Distribution of CNG during injection at 96 and 76 deg BTDC with injection angle 70 deg at 4000 rpm

The injector with one nozzle does not enable a good distribution of the fuel inside the combustion chamber and a bowl in the piston head is needed for direction of the fuel to the spark plug. The beginning and duration of the injection with the proper dose of fuel are the main parameters that should be chosen for stratified charge in the experimental engine tests.

6. Engine parameters

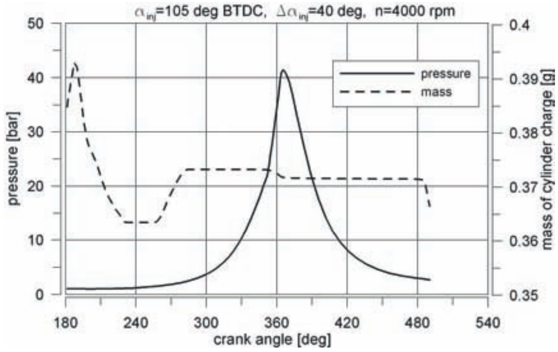

Fig. 7. Cylinder pressure and charge mass at stratified charge mode in the naturally aspirated engine

The cylinder pressure in a naturally aspirated engine fuelled by CNG in the stratified mode is on the same level as SI gasoline engine (Fig.7). Mass of the cylinder charge decreases at the end of the opening of the inlet valves about 7% and after CNG injection increases as a result of inflow of the gaseous fuel. The combustion of the natural gas takes place in a long time and in the considered case until the exhaust valves opened (Fig.4).

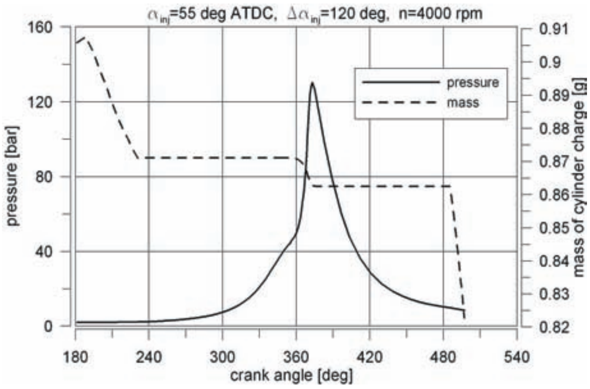

Fig. 8. Cylinder pressure and charge mass at homogeneous charge mode in the high-charged SI engine

In the engine with high charging ratio (γ=2.0) working in homogeneous mode the maximum combustion pressure amounted 130 bars and the whole fuel was burnt in 30 deg CA (Fig.8). Because of pressure fluctuation in the inlet pipe the charge in the cylinder decreases and the maximum pressure also has a lower value than it results from the theoretical considerations. The temperature distribution in the cylinder for the stratified charge after ignition is shown in Fig.9 at 6 deg CA BTDC and 18 deg CA ATDC, respectively. The lean and rich mixture regions limit the ignition and combustion processes.

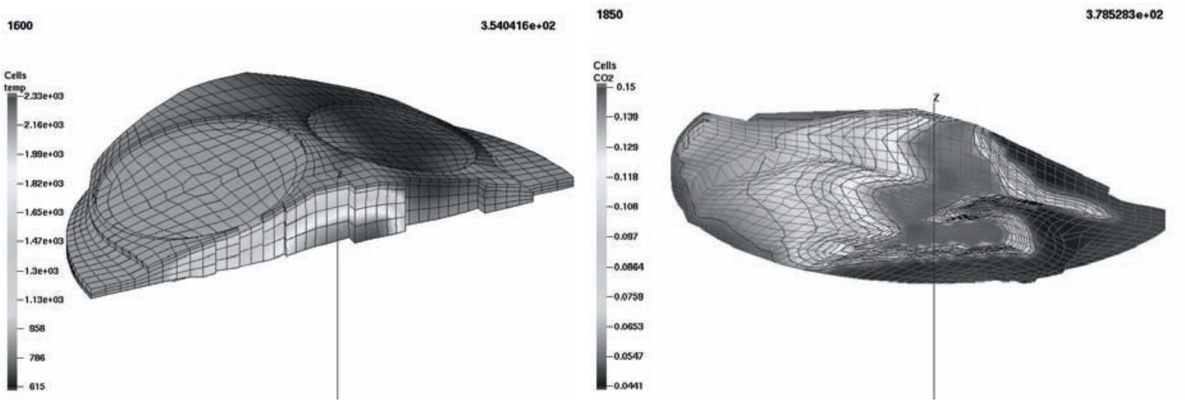

Fig. 9. Distribution of temperature inside cylinder at 6 deg BTDC and 18 deg ATDC with injection angle 70 deg at 4000 rpm

7. Emission of combustion products

Applying of CNG in the engine influences on lower concentration of CO_2, CO and hydrocarbons. However at high charging of the engine the combustion process takes place in higher temperatures and this increases NO_x concentration in the exhaust gases. Formation of the combustion products is dependent on the temperature. In Fig.10 the distribution of CO_2 in the cylinder at 18 and 129 deg ATDC is shown for stratified charge mode.

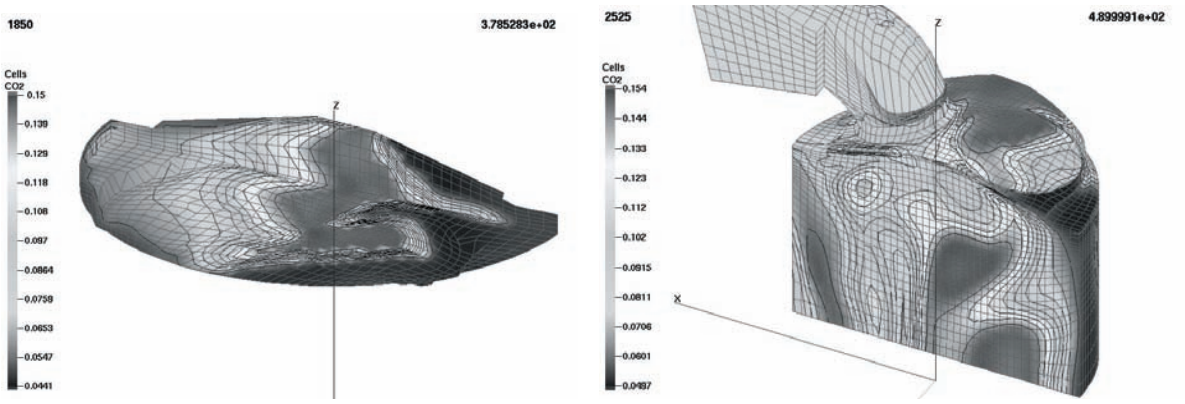

Fig. 10. Distribution of CO_2 at 18 and 129 deg ATDC with injection angle 70 deg at 4000 rpm

Concentration of CO_2 in the cylinder with the stratified charge increases rapidly during combustion process, however the maximum volumetric concentration do not exceed 3.5%, that is shown in Fig.11a. For homogeneous charge (Fig.11b) the increase of CO_2 concentration is proportionally less (7%) relative to fuel dose per cycle (0.045g). In this mode all fuel is burnt during the first 30 deg CA from occurring of the ignition.

a) b)

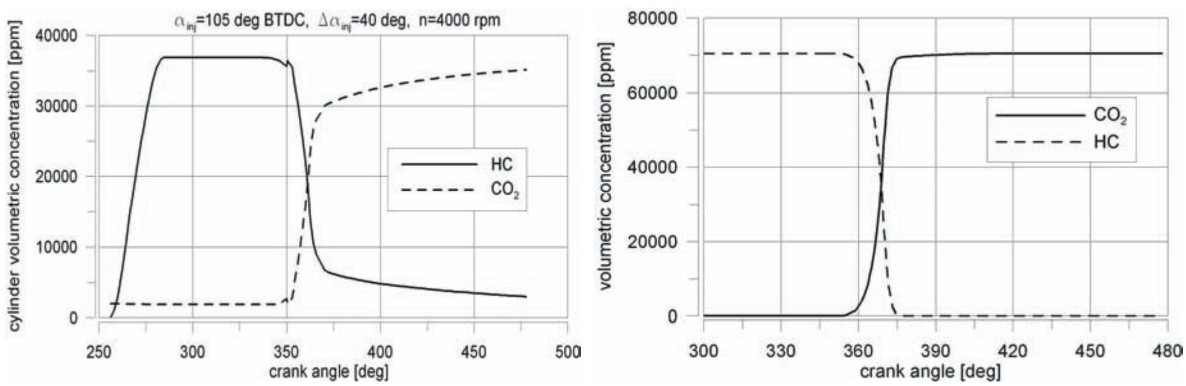

Fig. 11. Concentration of HC and CO_2 for stratified charge (a) and homogeneous charge (b)

For the first case after the combustion process there is a signified amount of unburned fuel in the cylinder, which is shown in Fig.11a. The charge stratification causes a formation of CO in some regions because of a local reach mixture, which is presented in Fig.12. However, the global amount of CO decreases during the combustion and expansion process. In this case there is seen an increase of nitrogen oxides above 1600 ppm (Fig.13a). The NO concentration is in the "frozen state" during the expansion stroke. The combustion of a homogeneous charge takes effect on a decrease of CO emission and on a higher concentration of NO as an effect of the higher charge temperature.

The simulation of the combustion process at 4000 rpm showed that volumetric concentration of NO is above 3000 ppm (Fig.13b), which is not acceptable by the environmental regulations. The slower combustion process can reduce the amount of NO for this engine operation. The experimental tests of combustion CNG were conducted in a caloric chamber for homogeneous mixtures with λ=1.0 – 1.6 and with different high initial pressures.

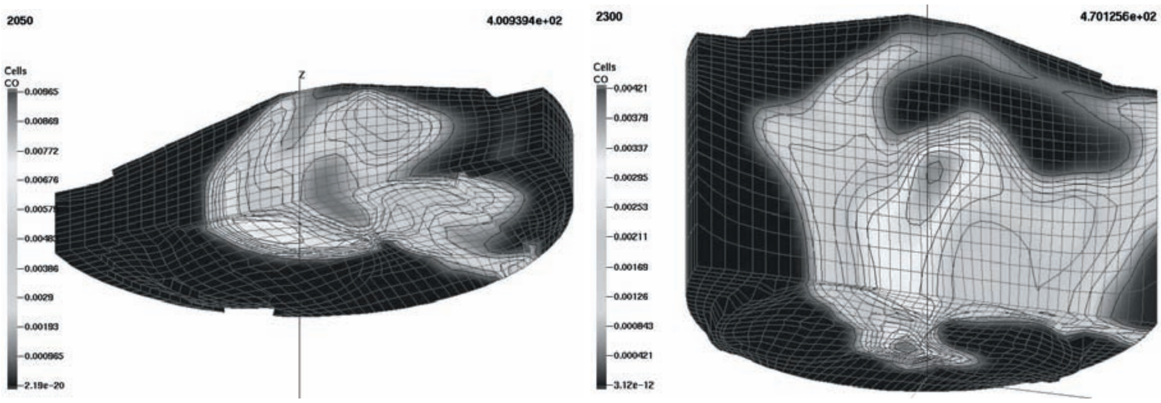

Fig. 12. Concentration of CO at 40 and 110 deg ATDC in stratified charge

a) b)

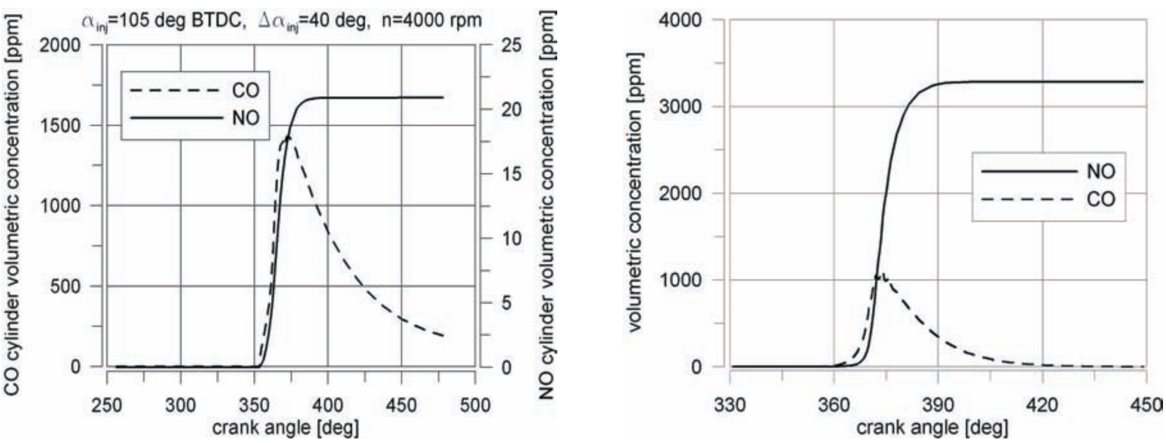

Fig. 13. Concentration of CO and NO in cylinder in the cylinder with stratified charge (a) and with homogeneous charge (b)

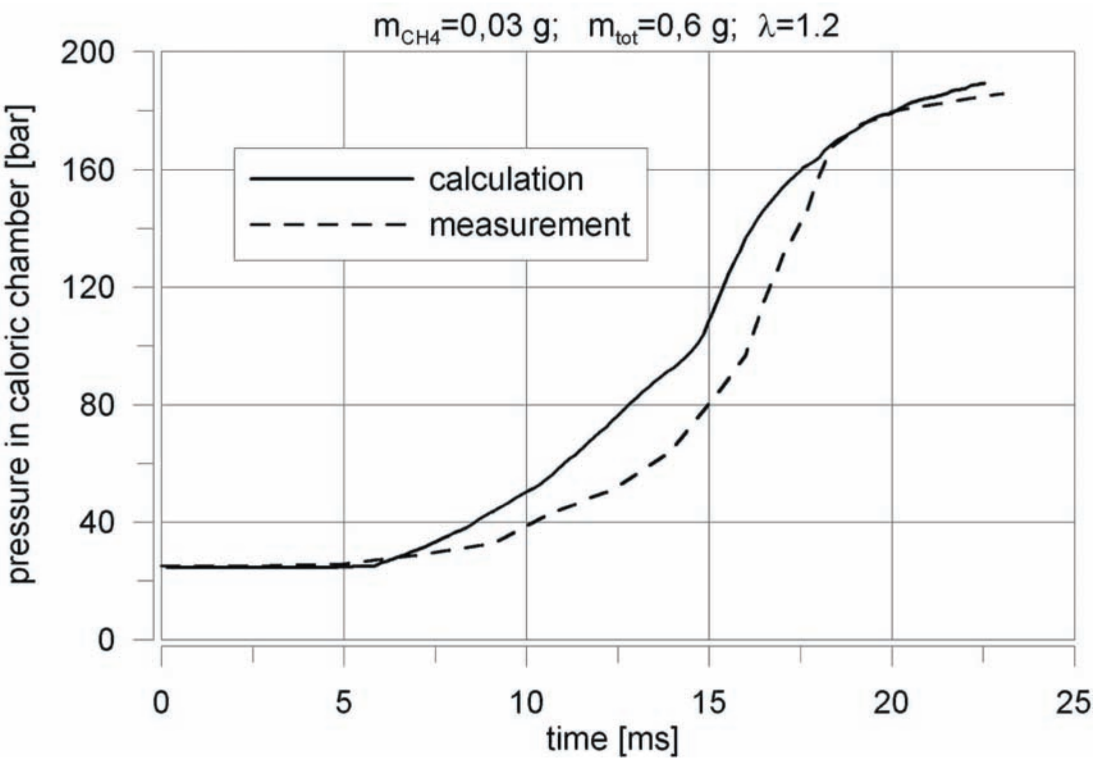

Fig. 14. Comparison of measured and calculated pressure in the caloric chamber

In Fig.14 the comparison of measured and simulated pressure variations in the caloric chamber is presented. In the test rig the pipes were closed by hydraulic valves with an additional volume and it influenced on the measured pressure. The calculation was done at assumption of any additional pipes. The both cases were considered at relative air-fuel ratio equal 1.2 in the caloric chamber with volume 20 cm^3 and initial pressure 25 bars.

8. Conclusions

In the paper there are presented some problems considering the direct injection and combustion process of the compressed natural gas in high charged SI engines.

1. The tendencies in development of European SI gas engines are directed on decrease of fuel consumption and possibilities of higher loading at lower rotational speed by applying of high charging and doing these type of engine more competitive to the diesel engines.
2. Direct injection of compressed natural gas in two modes to obtain a lean mixture at chosen injection parameters decreases CO and CO_2 emission.
3. Realisation of CNG direct injection for high loads requires a significant technical development of a new type of gas injector and utilization of the initial charge tumble or swirl at stratified charge operation.
4. The combustion of the homogeneous charge can cause a big increase of nitrogen oxides and lower emission of HC at stratified charge operation.
5. The injection pressure should be higher than 35 bars for both injection modes.
6. Location and settings of the gas injectors influences much more on the possibilities of mixture ignition for stratified charge at lower loads. On the basis of conducted simulation a good solution would be making a bowl in the piston or direction of the fuel stream by an additional air stream during fuel injection.
7. The high charging of SI engines fuelled by CNG influences on the durability of the crankshaft mechanism, cylinder sleeves and cylinder head.
8. Direct gas injection in high charged SI engines required the ignition systems of high voltage in the secondary circuit and longer time of sparking particularly for the stratified charges.

9. References

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