Research on combustion process of gasoline homogenous charge compression ignition engine

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Abstract. The paper presents the research designed to develop a HCCI (Homogenous Charge Compression Ignition) engine starting from a spark ignition engine platform. The chosen test engine was a single cylinder, four strokes provided with a carburettor. The results of experimental research data obtained on this version were used as a baseline for the next phase of the research. In order to obtain the HCCI configuration, the engine was modified, as follows: the compression ratio was increased from 9.7 to 11.5 to ensure that the air – fuel mixture auto- ignite and to improve the engine efficiency; the carburettor was replaced by a direct fuel injection system in order to control precisely the fuel mass per cycle taking into account the measured intake air-mass; the valves shape were modified to provide a safety engine operation by ensuring the provision of sufficient clearance beetween the valve and the piston; the exchange gas system was changed from fixed timing to variable valve timing to have the possibilities of modification of quantities of trapped burnt gases. The cylinder processes were simulated on virtual model. The experimental research works were focused on determining the parameters which control the combustion timing of HCCI engine to obtain the best energetic and ecologic parameters.

1 Introduction

Transport plays an important role in today's economy and society and has a large impact on economic growth and employment. It is generally agreed that transport sector is one of the major contributors with pollutants and GHG of environment.

Reducing exhaust emissions and increasing the fuel economy of internal combustion engines are of global importance. To meet the demand of economy, energy conservation, less environmental harmful exhaust emissions, especially carcinogenic nitrogen oxides (NOx) and responsibility for a greenhouse effect carbon dioxide (CO2) and higher thermal efficiency present generation engines must be characterized by: low fuel consumption; high efficiency; reliability; low price and low cost of usage. To achieve these tasks regarding diminishing fuel consumption, pollutants and CO_2 emissions the implementation of the concept of homogenous charge compression ignition engine (HCCI) is required.

The combustion technology of HCCI combines the main advantages of the gasoline engines (the homogeneous charge) with the main advantages of the diesel engines (higher compression ratios leading to a higher efficiency) determining the potential of reducing the emissions while increasing fuel economy.

Analysing the main strength and weak points of a conventional spark ignition engine and compression ignition engines, it is possible to summarize the advantages and disadvantages of HCCI engine.

Firstly, the spark ignition engines provided with a three-way catalyst can be considered as being very clean engines, problem appears during on part-load conditions. Low efficiency at part-load results from pumping losses due to specific spark ignition (SI) engine load control. SI engine is controlled by the changing of the air flow rate, adapting the fuel flow, in order to obtain the desired air fuel ratio. The SI engines operate with stoichiometric mixture ($\lambda \approx 1$) which produces a large amount of CO_2, this is responsible for the greenhouse effect. In other terms, low efficiency means high fuel consumption especially at part loads and high emissions of CO_2. Worthy mentioning the fact that in urban traffic the engines mainly run at part loads, resulting a very low level of overall efficiency on urban normal driving conditions

Secondly, the compression ignition engines are widely used in small displacement passenger cars, vans, trucks, buses, and locomotives, as well as large high power marine and power-generation applications. Load control is achieved by changing the amount of fuel injected in engine cylinders; the air flow rate is almost the same for all load conditions for naturally aspirated CI engines or, in the case of turbocharged engines, the air mass flow increases with engine load. Compression ignition (CI) engine in opposition to SI engine has no pumping losses and, thereby, has higher part-load efficiency. Naturally aspirated or supercharged engines operation in combination with high compression ratio caused that the part-load efficiency is higher for a CI than for a SI engine. The CI engines are the important source of particulate emissions, 0.2% – 0.5% from the fuel mass injected in the

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cylinder is emitted as small (~ 0.1μm diameter) particles which consist primarily of soot with some additional absorbed hydrocarbons. Particulates are mainly formed in the rich region of combustion area and NOx - in the hot temperature burned gas regions. Latest technologies used on CI engines for exhaust gas after treatment are very complex systems and expensive.

The HCCI could give 50% improvement in engine efficiency at part load compared to spark ignition engines and 30% compared to compression ignition engines, zero particulate matters and very low NOx emissions (< 10 ppm – compared to >500 ppm for diesel engines), as a result of using a diluted air-fuel mixture. In HCCI engine, the soot formation due to the use of homogeneous charge is not of a main concern.

The HCCI engines having a simplified construction, do not require:

- sophisticated shapes of inlet duct (cylinder head) to generate specified turbulence in combustion chamber;
- high pressure injection system;
- very complex aftertreatment exhaust gas systems.

The HCCI engine can be regarded as a hybrid of both types (SI and CI) of engines with attractive properties as: higher specific heat ratio of very lean mixture; higher cycle efficiency caused by using higher compression ratio; closer to ideal cycle due to the faster combustion.

The HCCI engines can operate on gasoline, diesel fuel, and most of alternative fuels.

2 Combustion process in HCCI engine

The combustion process in HCCI engine is different from conventional engine as SI and CI engines because of lacks of flame propagation. Ignition of homogenous air-fuel mixture occurs simultaneously at whole volume of combustion chamber and burns at the same time.

The chalange of HCCI engine is to control the start of combution timing and the combustion parameters as comubustion heat release rate and combustion temperatures.

The gasoline HCCI combustion is achieved by controlling the levels of the cylinder charge temperatures, pressures, and mixtures composition through the following parameters: the EGR rate, air-fuel ratio, compression ratio, intake temperature, inlet manifold pressure, the injection timing, fuel properties, the air-fuel homogenization, the heat transfer to the engine.

Because the entire mass of air fuel of combustion chamber burns simultaneously, the combustion process is caracterized by high values of heat-released rate, cylinder's pressures and a rapid increased of cylinder's temperatures, these parameters can be controlled by air-fuel mixture dilution with burnt gases.

In case of gasoline HCCI engine, the management of burnt gases for HCCI combustion can follow two strategies:

Residual gas trapping, known as negative valve overlap strategy (NVO). The great amount of residual gas is trapped in cylinder when the exhaust valve is closed early, during the exhaust stroke. In order to keep the trapped burnt gases in cylinder and to avoid the back flow of these gases into inlet manifold, the intake valve is opened later, after TDC at the beginning of the inlet stroke.

This strategy involves the recirculation of burnt gases after they left the engine cylinder, and the valve timing permit the positive valve overlap. The burnt gases recirculation can be realized using the so-called internal gas recirculation (IEGR) method. The method consists in sucking the exhaust gases existing in the exhaust system by reopening for a short period of time the exhaust valve during the intake stroke or by closing of the exhaust valve later during the inlet stroke.

With increasing amounts of exhaustgas recirculation (EGR), the combustion process becomes slower, resulting in lower peak pressure and lower rate of heat release and therefore longer combustion rates. Both the combustion and gross indicated efficiencies increased with increasing EGR.flows.

Other strategies applied to control the combustion of HCCI engines is the fuel injection process by injector location, fuel-spray patterns and by varying injection timing and duration.

In the gasoline HCCI engine development, three injection patterns may be implemented, as follows:

a) injection during negative overlap duration to determine fuel reformation;
b) injection during to intake stroke to form homogenous air-fuel mixture;
c) Injection during compression stroke to form stratified air-fuel mixture.

Research regarding combustion process shows that varying the charge in homogeneity, a method for controlling the combustion duration in HCCI engines can be used.

For combustion process Vibe function is used to aproximate the rate of heat released rate during the combustion process:

$$\frac{dx}{d\alpha} = a \cdot (m+1) \cdot y^m \cdot e^{-a \cdot y^{(m+1)}} \quad (1)$$

$$dx = \frac{dQ}{Q} \quad (2)$$

$$y = \frac{\alpha - \alpha_0}{\Delta \alpha_c} \quad (3)$$

Where:

Q - the total heat released,
α – crankshaft position,
α_0 - crankshaft position at the beginind of combustion process,
$\Delta\alpha_c$ – Combustion process duration,
m - Shape parameter,
a - the Vibe parameter (6.9 for a complete combustion).

To obtain the mass-fraction-burned equation (2.1) has to be integrated:

$$x = \int \frac{dx}{d\alpha} \cdot d\alpha = 1 - e^{-a \cdot y^{(m+1)}} \quad (4)$$

Mass-fraction burned for each engine's cycle represents a normalized quatity on a scale from 0 to 1 (0 until the start of combustion, 1 after the complete fuel burned during the combustion process).

Mass-fraction burned is used to appreciate the release of chemical energy as function of crankshaft position.

To obtain a more accurate approximation of the heat release a combination of two or more Vibe functions has to be used. For the gasoline HCCI engines, where the heat is released in a single stage, the best results were obtained by using a combination of three Vibe functions:

$$\frac{dx}{d\alpha} = pv_1 \cdot a \cdot (m_1 + 1) \cdot y^{m_1} \cdot e^{-a \cdot y^{(m_1+1)}} + pv_2 \cdot a \cdot (m_2 + 1) \cdot y^{m_2} \cdot e^{-a \cdot y^{(m_2+1)}} + pv_3 \cdot a \cdot (m_3 + 1) \cdot y^{m_3} \cdot e^{-a \cdot y^{(m_3+1)}} \quad (5)$$

and for mass-fraction burned:

$$x = \frac{\left(1 - e^{-a \cdot y^{(m_1+1)}}\right) + \left(1 - e^{-a \cdot y^{(m_2+1)}}\right) + \left(1 - e^{-a \cdot y^{(m_3+1)}}\right)}{pv_1 + pv_2 + pv_3} \quad (6)$$

Where *pv1*, *pv2* and *pv3* are the shares of the corresponding Vibe functions.

3 Experimental setup

To collect the experimental data in this research a single cylinder four-stroke naturally aspirated engine was used. The test engine having the specifications presented in Table 1 was connected to a SIEMENS 1PH6167 4NG00 dynamometer, in order to obtain the engine torque and power. The HCCI test engine was realized starting from a SI engine equipped with a carburettor as a platform. In order to obtain a HCCI operation cycle some modifications were implemented to the engine. The compression ratio was increased from 9.7:1 to 11.5:1, the fuelling system was changed, replacing the carburettor by direct injection system provided with an electromagnetic actuating injector. The injector was fitted by replacing the spark plug.

Table 1. Engine's specifications

Engine type	**SI**	**HCCI**	**[-]**
No. of cylinders	1	1	[-]
Fuel	Gasoline	Gasoline	[-]
Displacement	652	652	[cm^3]
Compression Ratio	9.7:1	11.5:1	[-]
Bore	100	100	[mm]
Stroke	83	83	[mm]
Con-rod length	149	149	[mm]
Valve position	DOHC	DOHC	[-]

To modify the quality of cylinder's mixture in order to control the combustion process, the engine was provided with a variable gas distribution system using a camshaft phaser. The original camshaft was replaced by a camshaft with new redesigned cams' profile. These cams provide a short valves opening period and a lower valve lifts to offer a safety clearance (valve-piston) when the valve timings are changed during the research work. The standardized equipment was used for tests.[1,2,3,4,5,6,7].

During the tests, one could register data regarding: ambient, intake and exhaust pressures, injection and cylinder pressure, oil pressure, ambient, cylinder, oil, water temperatures, torque, the valves timing and pollutant emissions (HC, CO, NOx).

The inlet air has the possibility to be conditioned in conformity with research requirements to obtain the control of combustion timing. The inlet and exhaust pressure was measured through KISTLER 6041A and KISTLER 4075A10 transducers. The cylinder's pressure data were obtained with a KISTLER 6041A piezoelectric transducer. The information regarding crankshaft's position was obtained by using a transducer type ROD 426A.

The fuel consumption was measured by using a fuel-mass flow meter and the air-flow was measured by using a laminar anemometer.

The engine test incorporated a full electronic and manual control.

The equipment used for pollutant measurements were NDIR Pierburg for O_2/N_2 and CO, CO_2, CLD Pierburg for NO/NO_x, HFID AVL for HC.

The "BremoXP" soft-ware was used for thermodynamic analysis of combustion process.

4 Experimental results and discussions

The main problem of gasoline HCCI engine is the control of combustion timing.

The auto-ignition process is a result of various complex chemical, physical and energetic processes that take place during the auto-ignition process. It was observed that before the main ignition of the mixture, a first-stage smaller-scale reaction, called "cool flame" was also present. By simulating the cylinder's processes, it was found that the pressure required to initiate the first-stage reaction was a linear function of the compression pressure at TDC, while depending on the fuel, the required compression pressure to initiate autoignition decreased with increasing fuel concentration. To increase the compression pressure to improve auto-ignition properties, the engine's compression ratio was rised from 9.7:1 to 11.5:1.

To control the start of combustion timing (SOC) and to moderate the pressure rise rate during the combustion process internal exhaust gas recirculation (IEGR) was used, and met the both requirements, to rise the mixture temperature and to realize the mixture's dilution, exploiting the burnt gases effects: heating, dilution, heat capacity and chemical activity. The amount of trapped residual gases (TRG) in the cylinder can be changed, by changing the valves timing of the engine which determine modifications of temperature, pressure, and composition of the cylinder mixture at inlet valve closing (IVC), achieving a stable HCCI combustion. The negative valve overlap strategy (NVO) was adopted for this experimental research. The amount of residual gas was trapped in cylinder when the exhaust valve is closed early, during the exhaust stroke. In order to keep the trapped burnt gases in cylinder and to avoid the back flow of these gases into inlet manifold, the intake valve is opened later, after TDC at the beginning of the inlet stroke.

The test was realized for exhaust vale closing (EVC) values: 639[CA] *(81[CA BTDC])*, 644[CA] *(76[CA BTDC]*), 646[CA] *(74[CA BTDC])* and 649[CA] *(71[CA BTDC]*), (Fig.1.) and (Fig.2.).

The main goal of the experiments was to investigate the possibility of controlling combustion phasing and combustion duration using various internal exhaust gas recirculation (IEGR) fractions.

By increasing the advance of closing of the exhaust valve timing from 71[CA BTDC] 649 [CA] to 81[CA BTDC] 639 [CA], the peak of heat release rate rises by 31 [%]. These increases determine a vigorous combustion process due to the rise quantity of trapped burnt gases.

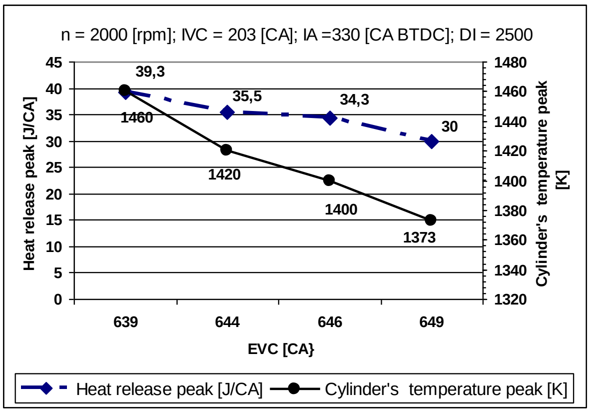

Fig.1. Influence of the exhaust valve closing timing on heat release rate peak and cylinder's temperature peak

Cylinder's temperature peak increases from 1373 [K] to 1460 [K] (Fig.2.) representing 6.34 [%] for the analyzed interval, despite of the rise of the air-fuel mixture dilution with burnt gases, then, the NOx emission increases from 17.86 [ppm] to 25.72 [ppm] representing 44 [%] (Fig.2.) due to the influence of cylinder's temperature.

Diminishing the quantity of burnt gases in the cylinder's mixture determinates the moving of combustion in the expansion stroke, the angle of heat release rate peak moves from 363 to 367 [CA] and the angle of peak of cylinder's temperature change from 371 [CA] to 377 [CA].

The influence of the intake valve's closing was realized for 204, 210 and 215 [CA], maintaining constant the following parameters: EVC at 649 [CA], the injection advances at 330 [CA], the injection duration ID=2550 [µs] and the engine speed n = 2000 [rpm], in these cases, the quatities of burnt gases and fuel remainig constant in the cylinder.

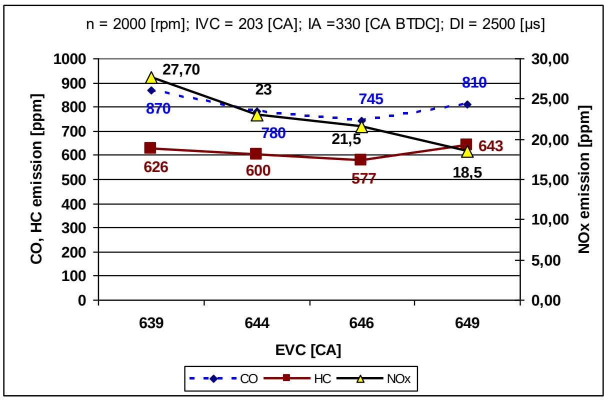

Fig.2. Influence of the exhaust valve closing timing on CO, HC and NOx emission.

By changing of values of IVC, the intake valve's closing timing determines the modification of the fresh-air content in the cylinder's mixture, in fact, the quality of mixtures are changed. Increasing the values of retarding the inlet valve's closing from 204 [CA] to 210 and 215 [CA], the data of experimental research show that the peak of heat release rate decreases from 29,8 [J/CA] at IVC = 204 [CA] to 25,1[J/CA] at IVC = 210 [CA] (-15,8%) and to 23,8 [J/CA] at IVC = 215 [CA] (- 20,1%), (Fig.3.).

The peaks of heat release rate are placed at the same crank angle. The peak of cylinder's temperature decreases from: 1377 [K] to 1347 [K] and, 1320 [K] for established IVC values. The pollutant emissions follow the values of cylinder's temperatures, HC increases from 650 [ppm] to 681 [ppm] and 775 [ppm], CO increases from 800 [ppm] to 1000 [ppm] and 1445 [ppm], and NOx decreases from 17, 5 [ppm] to 15, 2 [ppm] and 15, 6 [ppm], (Fig.4.).

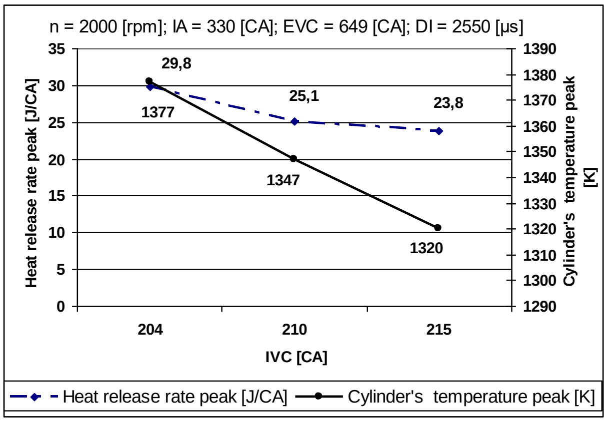

Fig.3. Influence of the inlet valve closing timing on heat release rate peak and cylinder's temperature peak

Fuel injection into the cylinder at different stages of the engine cycle allows HCCI combustion timing to be advanced by improving mixture ignitability or retarded by increasing fuel stratification, creating the possibility of expanding the low- and high-load operating limits.

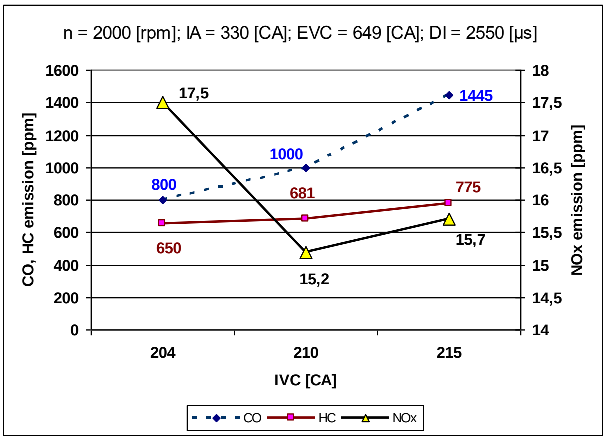

Fig.4. Influence of the exhaust valve closing timing on CO, HC and NOx emissions

Direct injection can be a good way to control HCCI combustion, but it depends heavily on the type of fuel and the timing of the direct injection.

For this research, the early direct injection of fuel in the intake stroke produced near identical results to a premixed charge. The values of injection advances were, as follow: 325 [CA]; 335[CA]; 345[CA]; 350[CA], or 35 [CA ATDC] 25 [CA ATDC] 15 [CA ATDC] 10 [CA ATDC] of the engine cycle. These starts of injection point were placed during the intake stroke after the exhaust valve had been closed, but before the intake valve had been opened.

In order to keep constant the quantity and quality of air-fuel burnt gases mixture during injection process, the following parameters were established at constant values: engine speed n =2000 [rpm], intake valve's closing timing IVC = 203 [CA], exhaust valve's closing timing EVC = 644 [CA], injection duration ID = 2500 [µs].

By incresing the fuel injection advance from 325{CA] to 350 [CA] the fuel was injected in hotter burnt gases, the jet dropplets evaporate and mixes with the burnt gases which have a remainig quntity of oxigen, hydrocabon molecules had same reforming chemical reactions which improve the ignition and combustion of cylinder mixture. After the opening of the intake valve, the burnt gases and the evaporated fuel will mix with the air, and this mixture will be homogenized during the intake and compression strokes.

For this reason, the peak of heat's release rate increases from 25.5 [J/CA] to 31.3 [J/CA] representing 22.75[%], (Fig.6.). The improvement of combustion process detemine an increased peak of cylinder's temperatures from 1327 [K] to 1373 [K], this representing 3.7%. The cylinder's temperature is an important parameter of NOx formation, (Fig.5.)

By increasing the injection advance, the NOx emission rises from 8.6 [ppm] at IA=325 [CA] to 14.4 [ppm] at IA=350 [CA] representing 67.44 [%], HC and CO emissions increases with decreasing o fuel injection advances (Fig.6.).

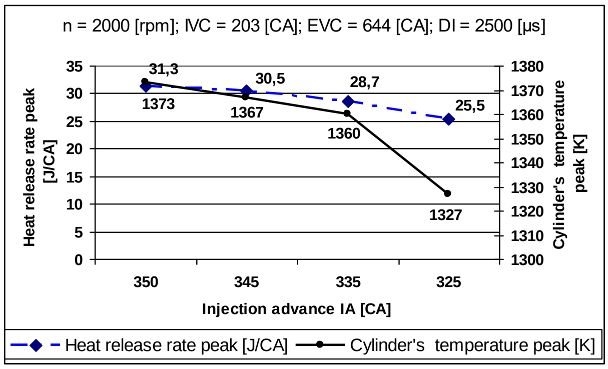

Fig.5. Influence injection advance on heat release rate peak and cylinder's temperature peak

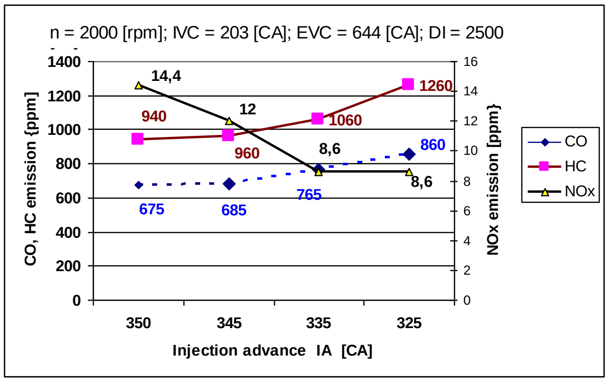

Fig.6. Influence of the injection advance on CO, HC and NOx emissions

The influence of engine's speed on the combustion processes was realized for 1900, 2000, 2100, 2200 and 2300 rpm, keeping constant the same parameters which maintain constant content of the cylinder.

On the one hand, the engine's speeds have an important influence on the heat exhange process which decreases at the increasing of engine's speed; on the other hand, the homogenization process of cylinder's content is intensified. By increasing of engine's speed, the combustion process is improved, it is represented by an increased heat release peak from 23,5 [J/CA] to 31,0[J/CA], representing +21,6%, (Fig.7.)

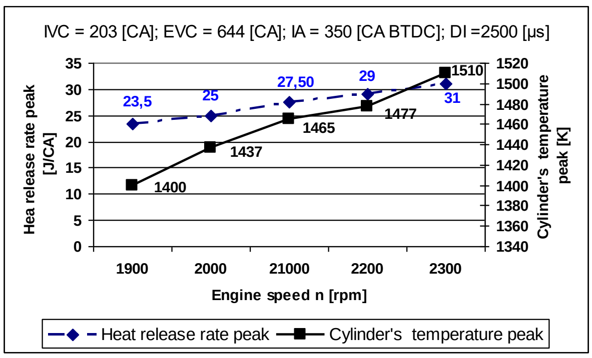

Fig .7. Influence of engine's speed on heat release rate peak and cylinder's temperature peak

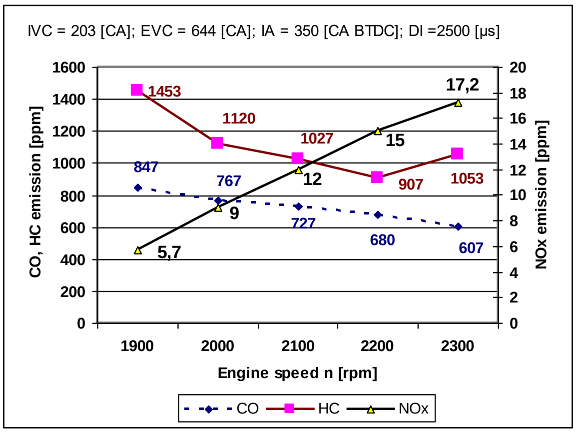

Fig.8. Influence of the engine's speed on CO, HC and NOx emission

The influence of the fuel quantity injected (engine's load) on cylinder's processes was realised by modyfing the fuel injection duration, the used values being 2400 [μs], 2500 [μs], 2600 [μs] and 2700 [μs], keeping constant same parameters, as presented in Fig.10.

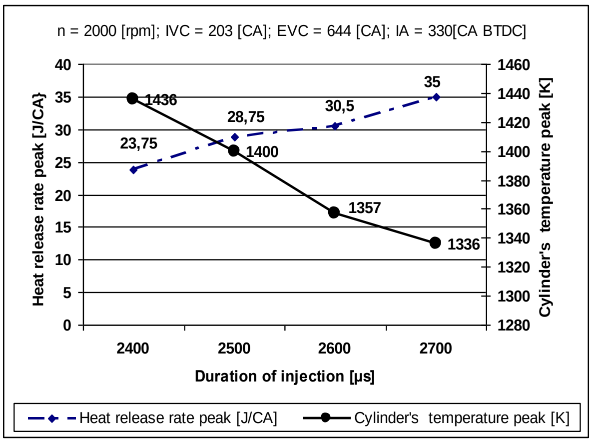

Fig.9. Influence of injection duration (engine's load) on heat release rate peak and cylinder's temperature peak

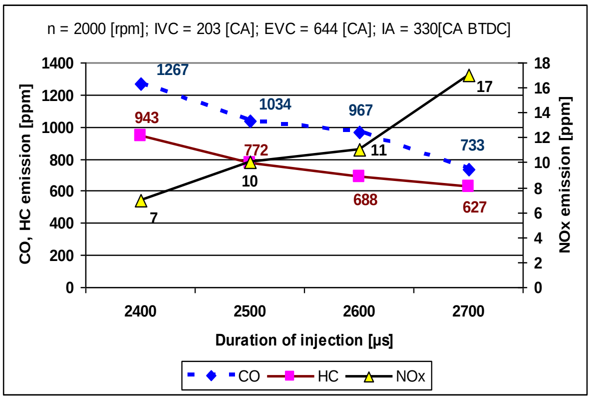

Fig.10. Influence of injection duration (engine's load) on CO, HC and NOx emission

5 Conclusions

The amount of trapped burnt gases has a great influence on the combustion process.The amount of residual gas trapped in the cylinder was changed by modifying the exhaust valve's closing timing earlier during the exhaust stroke. During the experimental research, the negative overlap strategy to control the engine's combustion processes by residual gas trapped in cylinder was adopted.

The intake valve's timings were chosen in order to keep the trapped burnt gases in the cylinder and to avoid the back flow of these gases into inlet manifold; the intake valve is opened later, after the TDC at the beginning of the inlet stroke. The closing of the intake valve was adopted after the TDC to maintain the volumetric efficiency in an acceptable range. For this reason, the intake and exhaust valves timing needs to be tuned and depends on the engine's load and speed.

The obtaining of homogenous mixture in HCCI needs to select carefully the advance of injection (start of injection). Within the experimental tests, the point (SOI) was placed during the intake stroke after the exhaust valve had been closed, but before the intake valve had been opened. However, injection close to TDC improved the combustion efficiency of very low fuel load mixtures

The influence of the operational parameters as: engine speed and engine load on HCCI combustion were tested.

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