Chemical and Process Engineering 2011, 32 (4), 453-471
DOI: 10.2478/v10176-011-0036-0

A STUDY OF PERFORMANCE AND EMISSIONS OF SI ENGINE WITH A TWO-STAGE COMBUSTION SYSTEM

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Lean mixture burning leads to a decrease in the temperature of the combustion process and it is one of the methods of limiting nitric oxide emissions. It also increases engine efficiency. An effective method to correct lean mixture combustion can be a two-stage system of stratified mixture combustion in an engine with a prechamber. This article presents the results of laboratory research on an SI engine (spark ignition) with a two-stage combustion system with a cylinder powered by gasoline and a prechamber powered by propane-butane gas LPG (liquefied petroleum gas). The results were compared to the results of research on a conventional engine with a one-stage combustion process. The test engine fuel mixture stratification method, with a two-stage combustion system in the engine with a prechamber, allowed to burn a lean mixture with an average excess air factor equal to 2.0 and thus led to lower emissions of nitrogen oxides in the exhaust of the engine. The test engine with a conventional, single-stage combustion process allowed to properly burn air-fuel mixtures of excess air factors λ not exceeding 1.5. If the value $\lambda > 1.5$, the non-repeatability factor COV_{Li} increases, and the engine efficiency decreases, which makes it virtually impossible for the engine to operate. The engine with a two-stage combustion process, working with $\lambda = 2.0$, the $Q_{in}/Q_{tot} = 2.5\%$, reduced the NO_x content in the exhaust gases to a level of about 1.14 g/kWh. This value is significantly lower than the value obtained in a conventional engine, which worked at $\lambda = 1.3$ with comparable non-repeatability of successive cycles (about 3%) and a similar indicated efficiency (about 34%), was characterised by the emissions of NO_x in the exhaust equal to 26.26 g/kWh.

Keywords: engine with two-stage combustion system, prechamber, excess air factor, indicated work, indicated efficiency, non-repeatability factor of the indicated work

1. INTRODUCTION

The problem of atmospheric air pollution by the exhaust gas of piston engines, particularly in highly motorised countries, is presently one of the most important aspects in the struggle for the protection of the natural environment. The necessity of limiting toxic components of the exhaust gas and reducing fuel consumption has resulted in a change in the combustion engine design and development. Reducing emissions of toxic components in exhaust gases of piston engines can be achieved by proper organisation of the combustion process, through the use of additives for fuels and by the neutralisation of exhaust and burning as a result of purification devices outside the engine (Bernhardt et al., 1976; Cupiał et al., 2003).

Among numerous harmful exhaust gas components, nitric oxides (NO_x) emissions are the most difficult to limit. High temperature and excess oxygen in the cylinder are conducive to nitric oxide formation, which transforms into nitric dioxide in the exhaust system and then in the atmosphere. The sum of these

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components is marked as NO_x. Nitric oxide creation speed increases as the temperature in the combustion chamber rises, especially above 1600 K. Lean mixture burning results in a decrease in the temperature of the combustion process and it is one of the methods of limiting nitric oxide emissions. It also increases the engine efficiency. Increasing the excess air results in a decrease in engine performance expressed by a decrease in the maximum of the indicated mean effective pressure and maximum torque and an increase in emissions of hydrocarbons in the exhaust. Conventional spark-ignition engines work properly only in a narrow range of excess air. Exceeding this range toward richer mixtures, on the one hand, is associated with the phenomenon of knocking and an increase in NO_x emissions, and exceeding this range toward lean mixtures is associated with increasing the non-repeatability of successive cycles of engine operation, misfire and an increase in emissions of HC and CO (Heywood, 1988; Kowalewicz et al., 1999).

An effective method to improve the lean mixture combustion process is a two-stage system of stratified mixture combustion in an engine with a prechamber. In such a system, the combustion chamber consists of two parts: the main chamber in the cylinder of the engine and the prechamber in the engine head connected with the main chamber by a connecting duct. A very lean mixture prepared in the engine inlet system (λ = 1.5 ÷ 3.0) is aspirated to the cylinder. However, a stoichiometric mixture (λ = 1.0) is delivered to the prechamber. The stoichiometric mixture ignition by spark discharge occurs in the prechamber and large amounts of CO and HC and slight amounts of NO_x are produced. As a result of the pressure increase, the burning charge of the prechamber is forced by the connecting duct to the main combustion chamber, where many moving ignition kernels develop. As a consequence, the lean flammable mixture, which could not be ignited by spark discharge, ignites in many regions. The ignition is fast enough to provide high engine cycle efficiency and helps to avoid disadvantages connected with combustion during the expansion stroke. At the time of the main combustion, slight amounts of NO_x are produced, and particles of CO and HC are burnt (Jamrozik, 2004).

One of the first attempts to study and analyse the effectiveness of ignition and combustion of lean mixtures in the engine with a prechamber was preliminary: the ignition system of the pilot flame torch. This system was developed and patented in 1963 and 1966 by L.A. Gussak (Gussak, 1966; Gussak et al., 1963). This system is characterised by the use of a prechamber of small dimensions (of the third inlet valve and the spark plug), which serves as a spark chamber, and lean combustion in the cylinder.

The two-stage combustion system in the engine compartment with a small spark was the subject of studies conducted since 1978 in Berkeley at the University of California by the team of Prof. Antoni K. Oppenheim (Oppenheim et al., 1978; Oppenheim et al., 1990). The research developed a system called controlled burning, in which the lean combustion in the cylinder followed the PJC generator pilot flame (pulsed jet combustion). The proposed concept of burning proved beneficial in terms of engine thermal efficiency and emissions of CO and HC.

Studies of the stratification of combustible gas charges using the sectional combustion chamber were conducted in 1990 in the UK (Charlton et al., 1990). The work included numerical simulations of the mixture creation process using the program PHOENICS and experimental studies on a stand test equipped with a single-cylinder, a turbocharger engine with a prechamber, powered by natural gas. A detailed analysis was conducted of the phenomenon of dilution to the rich fuel mixture in the prechamber during the compression process and the poor mixture injected with of the main chamber of the engine. Measurements of toxic gases have shown that as a result of a two-stage combustion of the stratified charge in the research engine, the flammability limits of lean mixture in the main chamber of the engine widened, and emissions of nitrogen oxides were reduced to very low levels compared with a conventional engine.

At the French University of Orleans, in 1999, a combustion system for lean mixtures called APIR (auto-inflammation pilotée par injection de radiucaux) was developed. In this solution, instead of the traditional spark plug, a spark plug was placed in a small prechamber, whose volume accounted for 1%

of the total volume of the combustion chamber of the engine. The prechamber was supplied with a rich mixture by an additional supply system. The prechamber was connected to the main chamber with four holes with a diameter of 1 mm each. The ignition in the chamber followed the initial spark discharge from the spark plug. The burning charge passing by the connecting holes yielded strong swirl, and the load was the beginning of intense burning in the cylinder. Research results, compared to the classic spark-ignition engine, showed a positive effect on the stability of the two-stage combustion engine, repeatability of successive cycles of engine operation, and resistance to knock and toxic components of exhaust emissions (Robinet et al., 1999).

Studies of combustion gas and lean flammable mixtures conducted in 2001 in Japan showed that the thermal efficiency improved due to changes in the compression ratio engine with a two-stage combustion system with stratified mixtures in the engine with a sectional combustion chamber (Endo et al., 2001). The work was carried out on a single-cylinder test engine to analyse the impact of the shape of the combustion chambers and the degree of turbulence of the fresh charge on the efficiency and resistance to knock in an SI engine with a sectional combustion chamber. These studies have shown that a withdrawal of the knock limit of the gas engine, by generation of turbulence due to the combustion chambers' topography, gives the possibility of increasing the engine compression ratio and improving its efficiency.

Improving the combustion process by shifting the ignition point of the mixture from the main combustion chamber to a small conventional engine did not involve a prechamber of the engine with two-stage combustion system when conducted at the Swiss Institute of Technology in Lausanne (Roethlisberger et al., 2003). Studies related to a six-cylinder SI engine powered by natural gas equipped with a non-fuel supplied prechamber were a continuation of previous numerical studies in KIVA-3V. The aim of this study was to determine the possibility of reducing toxic components of exhaust emissions with a two-stage combustion system of lean mixtures (λ = 1.6 ÷ 1.7) in the engine test for values defined by the standards of the Swiss. A detailed analysis was performed to optimise the dimensions and shape of the prechamber and location and diameter of holes connecting the prechamber with the main chamber. The study showed that the most convenient conditions for the ignition and spread of the flame front, thereby reducing toxic exhaust components, can be achieved with a prechamber with a capacity equal to 3% of the total volume above the piston in the TDC (top dead centre), a number of nozzles equal to 4 and a small total area nozzle holes (9.37 mm^2).

For many years Polish scholars have also been conducting research on the two-stage combustion system of lean mixtures in the engine with sectional combustion chamber. The results of the engine experimental investigations and the computer analysis of Pulsed Jet Combustion system are presented in the papers (Gmurczyk et al., 1992; Kesler et al., 1993; Leżański et al., 1993; Wolański, 1996; Wolański et al., 1997). Its operation is based on the introduction into the combustion chamber of a turbulent plume formed by a jest of rich combustion products issued from a small generator plug. The investigations were performed for lean combustible mixtures (λ = 1.0÷1.6) over a wide range of spark advance angles. The results of calculations as well as those obtained from experiments have demonstrated that the investigated system is advantageous. The combustion process in this system was evidently faster than that of the spark generated flame, yielding higher peak pressures and shorter pressure rise times. The advantages of the investigated system were particularly prominent for very lean mixtures.

At the Warsaw University of Technology, work was done on a two-stage combustion system with stratified gas charge in a sectional chamber with constant volume (Bocian et al. 2001; Jarnicki et al., 2001). The results of preliminary studies on the ignition of gaseous fuel injected into the combustion chamber equipped with a special spark chamber are presented. Experimental studies as well as computational simulation results showed that the use of a pre-ignition chamber ensures proper ignition of the mixture and has a positive effect on the combustion process.

Almost all of the major global automotive companies have conducted work on a two-stage combustion process of stratified mixtures. In part one, the research resulted in the implementation of a new engine design for mass production. The most well-known engine of this group was already developed in the 1970s by Honda CVCC system (compound vortex controlled combustion) (Kowalewicz, 1980). In the automotive industry, this group also includes solution companies: Ford, General Motors, Volkswagen, Walker, Eaton, Heintz, Nilov, Porsche, Toyota, and Mitsubishi (French, 1990; Noguchi et al., 1976).

Currently, most automotive SI internal combustion engines with liquid fuel and gas engines with a cylinder diameter up to about 200 mm with a stratification fuel mixture by altering the design of the combustion chamber does not allow the reduction of toxic emissions to the level imposed by the European standards (EURO IV and V) (www.dieselnet.com).

Creation and combustion of stratified mixtures in automotive engines initially implemented in the prechamber system (Honda CVCC) were abandoned in favour of targeted directed fuel injection to the combustion chamber (Mitsubishi GDI - gasoline direct injection) (Sendyka et al., 2005).

Currently, a two-stage combustion system for stratified mixtures with a prechamber is used primarily in modern, stationary, supercharged gas engines of medium and high power operating at a constant speed and focused mainly on the use of stationary, electricity generation and gas compression.

The two-stage combustion of lean gas mixtures using the sectional combustion chamber is used in modern stationary gas engines with high power ignition cylinder diameters exceeding 200 mm inter alia by the Austrian company Jenbacher AG (Diesel & gas turbine, 2003), Danish German MAN B & W Holeby (Janicki, 2000), a Finnish Wärtsilä NSD Corporation (Stenhede, 2003) and in the US by the Waukesha Engine Dresser (Jeffery, 1988) and Caterpillar Inc. (Mooser, 2003).

Better knowledge of mixture formation and combustion processes in two-stage combustion systems could have a great influence on gas engines with prechambers. The aim of this research was the analysis and better understanding of stratified mixtures formation and combustion processes in stationary gas engines of medium and high power.

The article presents the results of laboratory research on an SI engine with a two-stage combustion system with a cylinder powered by gasoline and a prechamber powered by gas LPG. The results were compared to results of a conventional engine with a one-stage combustion process.

2. TEST ENGINE

The test engine was constructed on the basis of a four-stroke compression-ignition engine manufactured by "ANDORIA" Diesel Engine Manufacturers of Andrychow, which, after some constructional changes, was designed for the combustion of gaseous fuel as a spark-ignition engine due to a new fuel supply system and an ignition installation. The engine is a stationary, two-valve unit with a horizontal cylinder configuration. The engine block is made of cast iron and is integrated with the crankcase. The engine cooling system used is the evaporation of the water jacket.

The paper presents changes in the base engine design, which enabled the implementation of a two-stage combustion system within the sectional combustion chamber.

The main engine element that underwent modernisation was the head (Figure 1). The changes implemented allowed an additional combustion chamber (prechamber) to be installed in the previously existing head of the S320 ER engine by setting the compression ratio to 8.6 (Figure 2).

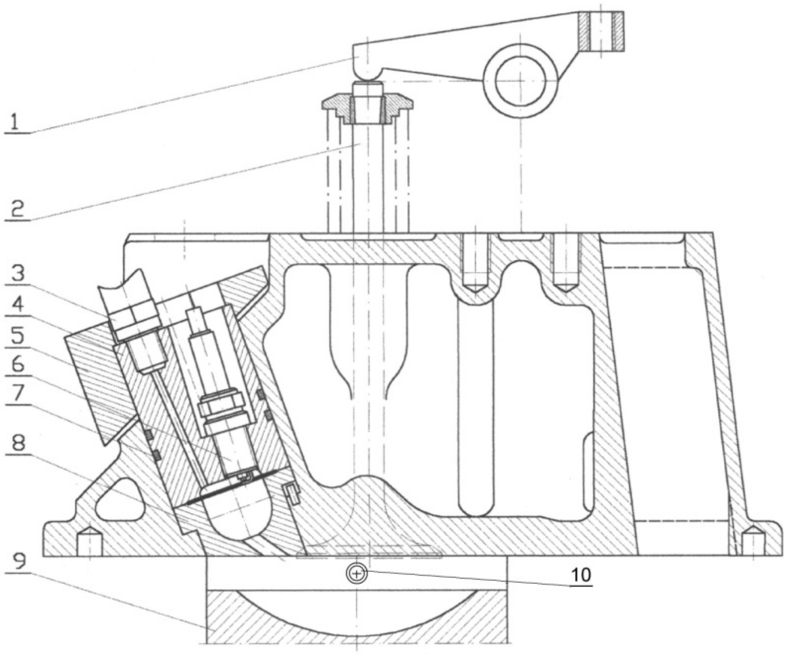

Fig. 1. Test engine head with prechamber
1 - valve rocker, 2 - inlet valve, 3 - flame suppressor, 4 - prechamber head, 5 - retaining cover, 6 - spark plug, 7 - sealing ring, 8 – prechamber body, 9 – piston, 10 – pressure sensor

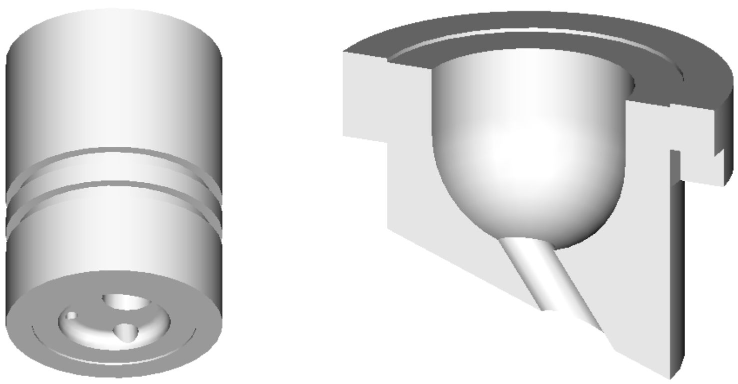

Fig. 2. Head and body of prechamber

The additional combustion chamber was made of an alloy with enhanced strength properties at high temperatures, called Nimonic 90. In the chamber, where the volume makes up 4.5% of the total combustion volume, an M14x1.25 sparking plug was mounted, and an M7x0.75 piezo-quartz sensor was installed. The prechamber was joined to the cylinder chamber by a canal of 6 mm in diameter.

The fuel mixture in the prechamber is enriched at the end of the compression stroke with gaseous fuel supplied through the additional fuel supply system, and the main elements are a set of non-return valves and a solenoid inlet valve. The water-cooled valves were placed in a special water jacket (Figure 3).

The main parameters of the test engine are shown in Table 1.

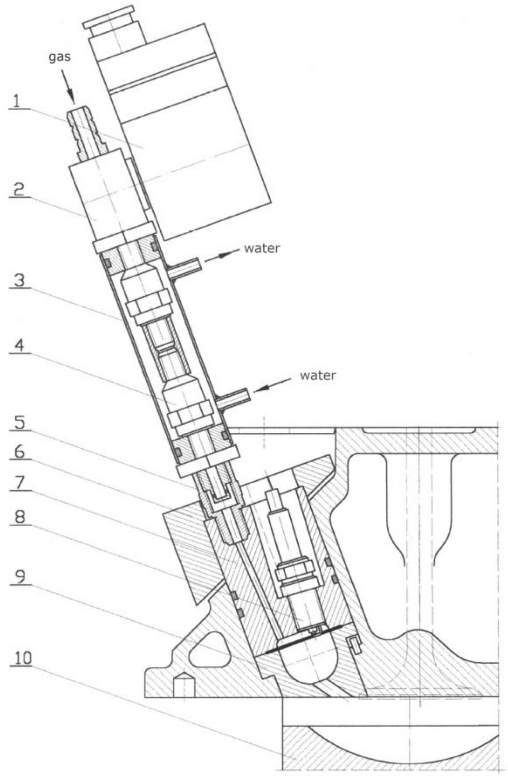

Fig. 3. Cooling system non-return valvesł
1 - solenoid coil, 2 - solenoid valve, 3 - water jacket, 4 - non-return valve, 5 - flame suppressor, 6 - retaining cover, 7 - prechamber head, 8 - spark plug, 9 - prechamber body, 10 - piston

Table 1. Main engine parameters

displacement volume	1810 cm^3
number of cylinders	1
cylinder configuration	horizontal
cylinder bore	120 mm
connecting-rod length	275 mm
piston stroke	160 mm
compression ratio	8.6
engine speed	1000 rpm

3. MEASUREMENT PROCEDURE

The tests were conducted at a constant rotational speed of n = 1000 rpm. The engine was brought to full loading after prior thermal stabilisation, that is, the cooling water was brought to boiling (the evaporation cooling system).

The tests included three main measurement series allowing for a different ratio of the thermal energy input with the fuel to the prechamber, Q_{in}, to the thermal energy input to the whole engine, Q_{tot}. The pressures occurring in the engine combustion chambers were recorded for Q_{in}/Q_{tot} = 2.5%, for Q_{in}/Q_{tot} = 5% and for Q_{in}/Q_{tot} = 8%, while the excess air factor was changed in the range from 1.4 to 2.0 and the ignition advance angle in the range from 6 deg to 18 deg before the TDC. The recording was made for 95 successive operation cycles every 1 deg with specialised software (Gruca, 2001). At the same time, other quantities necessary for the subsequent analysis of indication results (Cupiał, 2002) were measured, such as rotational speed, air consumption, liquid fuel consumption, gas fuel consumption, air temperature, combustion-gas temperature and ambient pressure and temperature. Using a combustion-gas analyser, variations in the emission of engine exhaust gas components, such as CO, HC and NO_x, were measured.

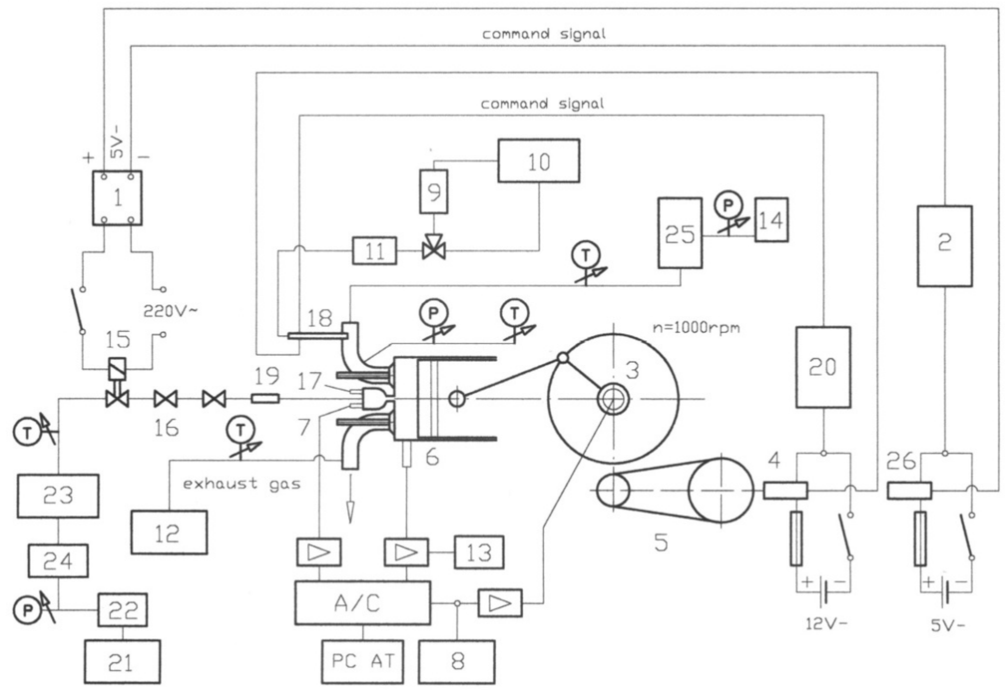

Fig. 4. Schematic diagram of the test standł

1 – electronic relay, 2 – pulse generating system, 3 – CA crank angle transmitter, 4 – magnetic induction sensor, 5 – belt transmission, 6 – main chamber pressure sensor, 7 – prechamber pressure sensor, 8 – counting module, 9 – flowmeter, 10 – liquid fuel tank, 11 – electric fuel pump, 12 – combustion-gas analyser, 13 – oscilloscope, 14 – measuring orifice, 15 – solenoid valve, 16 – non-return valve set, 17- sparking plug, 18 – electric injector, 19 – flame suppressor, 20 – electric injector control system, 21 – gas fuel tank, 22 – pressure regulator, 23 – pressure fluctuation damping reservoir, 24 – set of measurement rotameters, 25 – equalising tank, 26 – magnetic induction sensor

The study was conducted on a test stand (Figure 4) consisting of a combination of the following elements (Table 2).

Table 2. List of the elements of experimental setup

1. S320 ER modernised engine with prechamber for the two-stage combustion process
2. Combustion engine indication system
a. digital measurement system, which records and analyses fast-changing pressures in the engine combustion chambers • piezo-quartz pressure sensor, Kistler 6061 SN 298131 • charge amplifier, Kistler 5011 • piezo-quartz pressure sensor, PCB Piezotronics M112B10 SN 20761 • charge amplifier, PCB Piezotronics M462A

• a/d converter, eight-channel, 12-bit card AMBEX LC-020-812 • PC computer • digital CA crank angle transmitter, Kistler CAM 2611 • program for digital recording and analysis of the frequency signals LCTXR (Gruca, 2001)
b. system for measuring the liquid fuel supply in the main chamber of the engine • flowmeter of fuel with a volume of 40 cm^3 • timer with a scale interval 0.01 seconds
c. system for measuring the gas fuel supply in the prechamber • gas rotameters: TG 06.1 239 in the range of 2000 l/h and with a scale interval of 20 l/h, TG 03.1 195 in the range of 220 l/h and with a scale interval of 2 l/h • non-return valve • pressure fluctuation damping reservoir with a capacity 4.8 dm3 • analogue manometers: RPT94 229 in the range of 0.4 MPa, and with the scale interval 0.01 MPa and of the accuracy class 2.5; RPT96 238 in the range of 0.4 MPa, and with the scale interval 0.01 MPa and of the accuracy class 2.5 • digital thermometer ETT-2 of the range +200°C and of the scale interval 0.1°C • temperature sensor TP-204K-1b-100-1.5
d. measurement system of the engine speed based on the module count of pulses from crank angle CA transmitter with an accuracy of 1 deg
e. air consumption measurement system • equalising tank with a capacity of 115 dm3 • measuring orifice • micromanometer MPR-4 with the range of 200mmC2H5OH and the scale interval of 1 mmC2H5OH • digital thermometer ETT-2 with the range of +200°C and the scale interval of 0.1°C, • temperature sensor TP-204K-1b-100-1.5
3. Supply system for liquid fuel to the main chamber
a. liquid fuel tank
b. electric fuel pump
c. electric injector Mono-Motronic MA 1.7 Bosch
d. electric injector control system
e. fuel filter
4. Supply system for gas fuel to the prechamber
a. system controlling the electromagnetic valve inlet gas fuel • belt transmission to the transmitter • magnetic induction sensor Sels PCID-4 • pulse generating system • electronic relay Celduc SC942110
b. gas fuel tank
c. pressure regulator Metrix 2R1.5PN
d. analogue manometers: RPT94 229 with the range of 0.4 MPa, and with the scale interval 0.01 MPa and an accuracy class of 2.5; RPT96 238 with the range of 0.4 MPa, and with the scale interval 0.01 MPa and an accuracy class of 2.5
e. solenoid valve Danfoss EVI-C3
f. non-return valve set Danfoss NRV-G
g. flame suppressor

5.	Electronic ignition system that allows continuous adjustment of the ignition advance angle at work engine NGV Autogas Gastronic ESZ03
6.	Two-channel oscilloscope HAMEG HM2307
7.	Five gaseous exhaust gas analysers, Radiotechnika AI 9600 model, and software enabling the measurement of toxic exhaust components for the three types of engine fuel: gasoline, propane-butane and natural gas.

The selected parameters of the gas analyser are shown in Table 3.

Table 3. Selected parameters of the gas analyzer

Measured quantity	Range	Unit	Measuring accuracy
λ	0 ÷ 2	-	-
NO_x	0 ÷ 4000	ppm	± 32 ppm for the range 0÷1000 ppm
			± 60 ppm for the range 1001÷2000 ppm
			± 120 ppm for the range 2001÷4000 ppm
HC	0 ÷ 20000	ppm	± 12 ppm vol.
CO_2	0 ÷ 20	% vol.	± 0.5% vol.
CO	0 ÷ 10	% vol.	± 0.06% vol.

4. RESEARCH RESULTS

Examples of courses in the cylinder pressure and engine prechamber for the average value of excess air ratio λ = 2.0, Q_{in}/Q_{tot} = 2.5%, and the ignition advance angle 12 deg before the TDC as a function of the crank angle, are represented by the characteristics in Figure 5. Examples of courses in the cylinder pressure and engine prechamber for the average value of excess air ratio λ = 2.0, Q_{in}/Q_{tot} = 2.5%, and the ignition advance angle 12 deg before the TDC as function of the change in the displacement volume of the cylinder are represented by the characteristics in Figure 6.

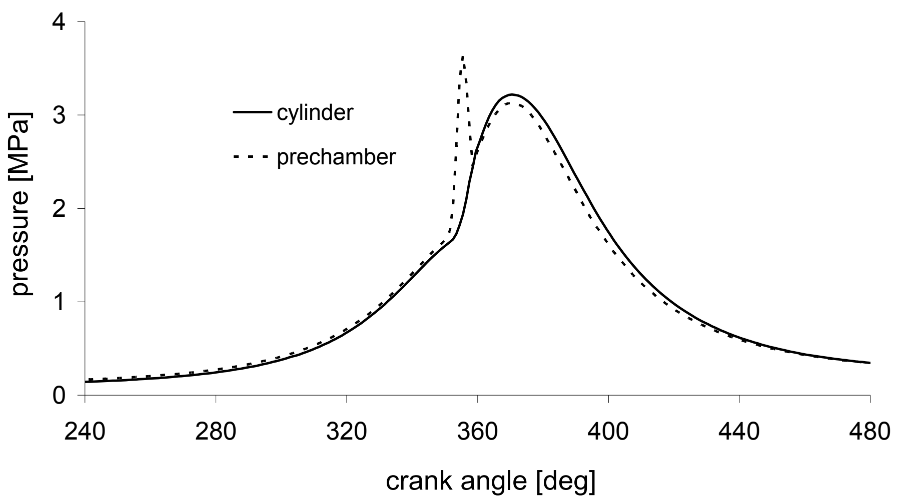

Fig. 5. Pressure in the cylinder and prechamber versus crank angle for λ = 2.0, Q_{in}/Q_{tot} = 2.5% and L_i = 0.55 MJ/m^3

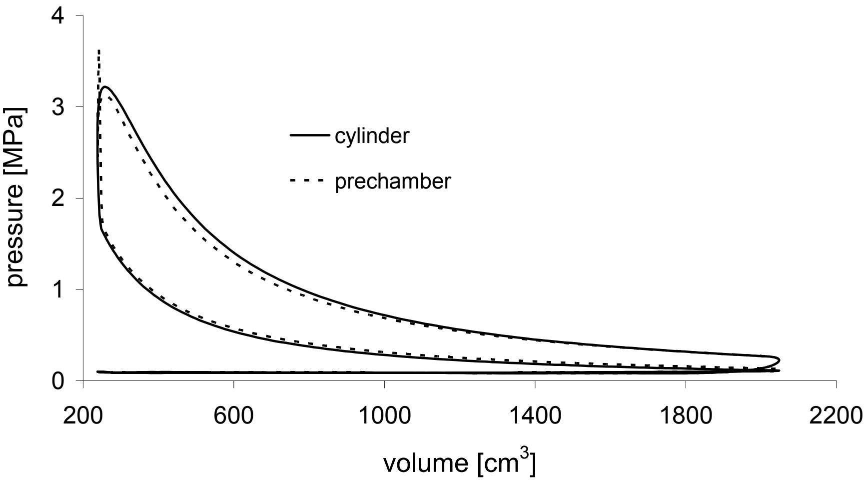

Fig. 6. Indicator diagram p-V of the cylinder and prechamber for λ = 2.0, Q_{in}/Q_{tot} = 2.5% and L_i = 0.55 MJ/m^3

Based on the characteristics of the ignition advance angle effect on indicated efficiency η_i and indicated work L_i, Figures 7, 8, and 9 show the optimum values of the ignition advance angle. The optimum value of the angle was that which occurred at the maximum efficiency and indicated work.

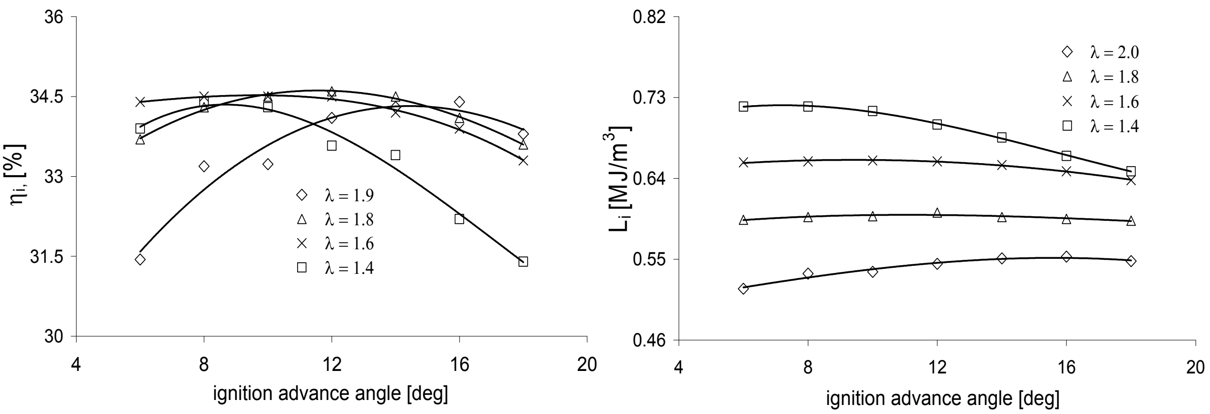

Fig. 7. Effect of ignition advance angle on the indicated efficiency and indicated work for Q_{in}/Q_{tot} = 2.5%

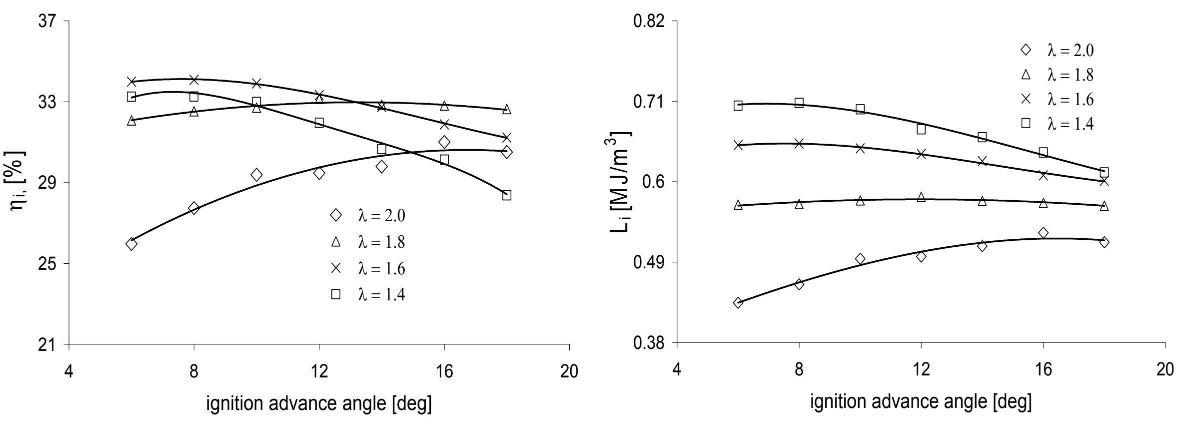

Fig. 8. Effect of ignition advance angle on the indicated efficiency and indicated work for Q_{in}/Q_{tot} = 5%

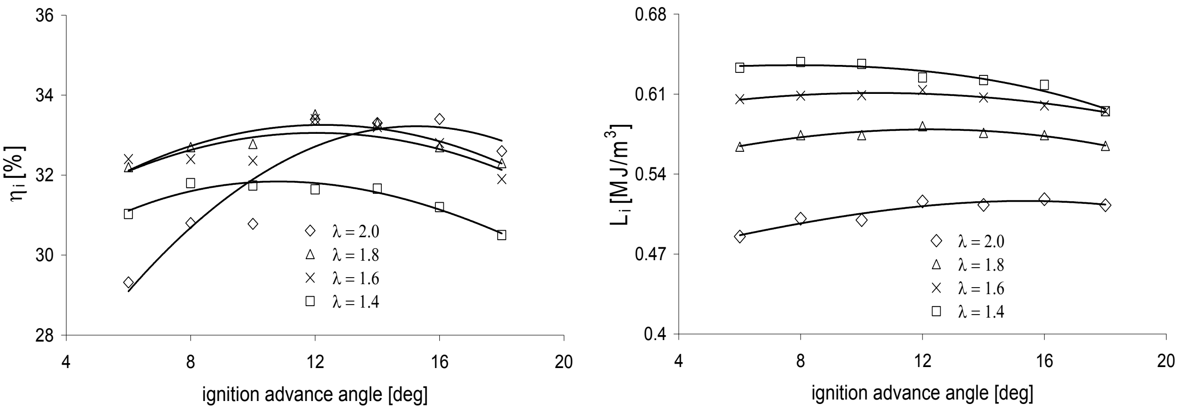

Fig. 9. Effect of ignition advance angle on the indicated efficiency and indicated work for Q_{in}/Q_{tot} = 8%

The ignition advance angle of the test engine with a two-stage combustion system depends on the load share of energy supplied to the prechamber Q_{in} and increases for all values of the Q_{in}/Q_{tot} ratio with increasing excess air factor fuel mixture (Figure 10).

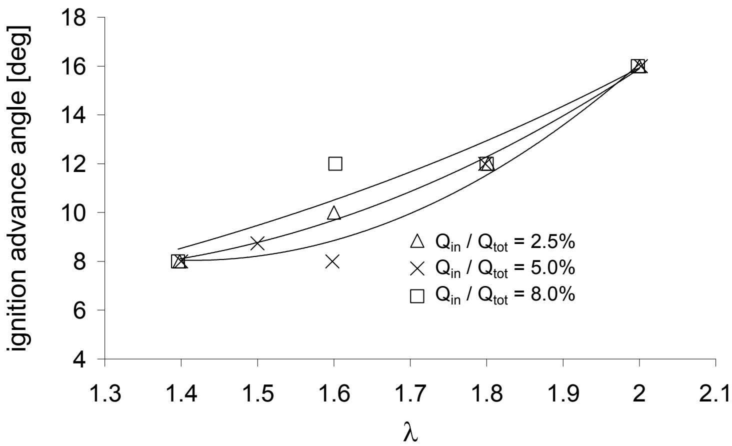

Fig. 10. The optimum values of the ignition advance angle of the engine with a two-stage combustion system

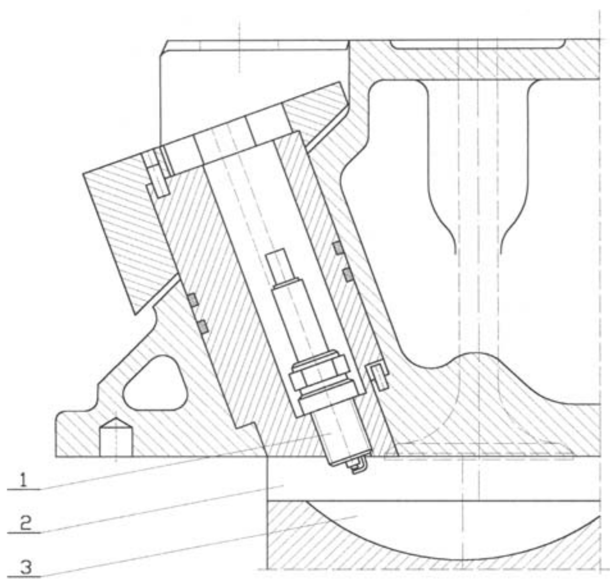

Fig. 11. Conventional test engine head;
1 – spark plug, 2 – combustion chamber in cylinder, 3 – combustion chamber in piston

The obtained results were compared with the results of measurements done on a conventional engine with a single-stage combustion system, whose cylinder bore was 120 mm with a compression ratio equal to 9 (Cupiał et al., 2002).

Quantities comparable to those of the conventional engine, such as indicated mean effective pressure, the non-repeatability factor of indicated work, and the amount of toxic exhaust components as a function of the excess air factor, were determined for the optimum values of the ignition advance angle.

The results of the combustion processes of a conventional engine powered by mixtures of gasoline with air in excess air ratios of 0.8, 0.9, 1.0, 1.3, 1.4 and 1.5, in which the combustion process proceeded steadily and there were no signs of misfire or knock.

The conventional test engine head and test engine head with prechamber are shown in Figures 11 and 12.

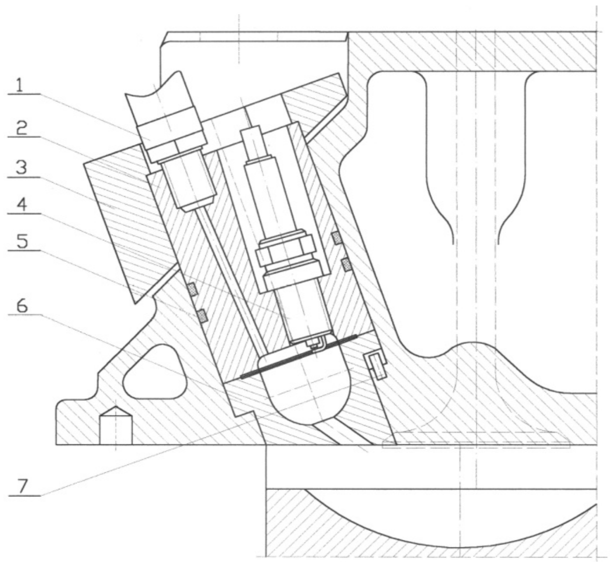

Fig. 12. Test engine head with prechamber;
1 – flame extinguisher, 2 – prechamber body, 3 – retaining cover, 4 – spark plug, 5 – sealing ring, 6 – prechamber, 7 – chamber dowel pin

One of the parameters determining the performance of the combustion engine is the indicated work.

$$L_i = \frac{\sum_{0}^{720} \frac{p_n + p_{n+1}}{2} (V_{n+1} - V_n)}{V_s} \tag{1}$$

where p_n, p_{n+1} are the instantaneous values of the pressure in the cylinder; V_n, V_{n+1} are the instantaneous values of the cylinder volume; V_s is the displacement volume.

The increase in excess air factor mixture, due to its depletion, leads either to an engine with a single- or two-stage combustion system or to a decline in the utility of performance. Figure 13 shows the change in the value of the indicated work for both systems as a function of the excess air factor. With the increase of the value of λ, the indicated work test engine decreases for both combustion systems. In the engine with a traditional combustion system, depending on the excess air factor, the value of L_i is included in the range of 0.96 to 0.8 MJ/m^3 (a decrease of 16.7%). In the engine with a sectional combustion chamber, the value of the indicated work depended on the Q_{in}/Q_{tot} and ranged from 0.72 to 0.52 MJ/m^3. For example, the share of Q_{in} was a 2.5% decline in indicated work, as analysed with a composition of the mixture from 0.72 to 0.55 MJ/m^3, which was equal to 23.6%.

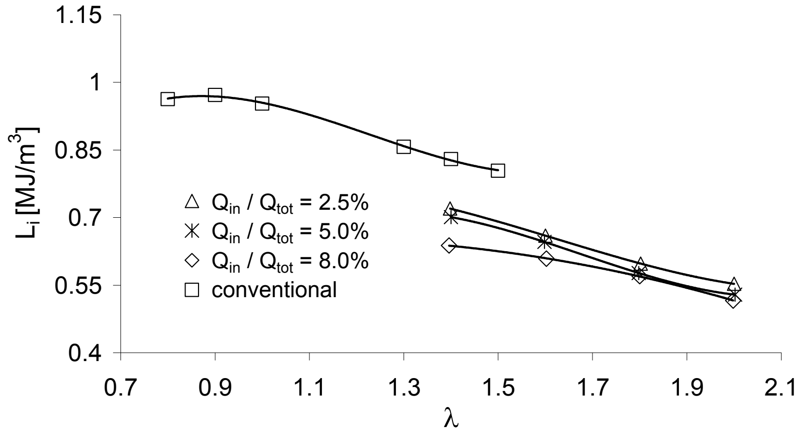

Fig. 13. Indicated work of the engine for the single - and two-stage combustion systems

The average value of the indicated efficiency of the test engine:

$$\eta_i = \frac{L_i V_s}{Q_{tot}} 100\% \tag{2}$$

$$\eta_i = \frac{L_i V_s}{Q_{cyl} + Q_{in}} 100\% \tag{3}$$

where Q_{tot} is the total heat supplied to the engine, Q_{cyl} is the heat supplied to the engine cylinder, and Q_{in} is the heat supplied to the prechamber.

The heat supplied to the engine cylinder:

$$Q_{cyl} = \frac{V_{cyl}\, \rho_{gas} W_{gas}}{0.5\, n\, t} \tag{4}$$

where V_{cyl} is the volume of gasoline delivered to the engine cylinder, ρ_{gas} is the density of gasoline, W_{gas} is the calorific value of gasoline, n in the engine speed, rpm, and t is the time consumption of gasoline delivered to the engine cylinder.

The heat supplied to the prechamber:

$$Q_{in} = \frac{V'_{in}\, \rho_{LPG} W_{LPG}}{0.5\, n} \tag{5}$$

where V'_{in} is the propane-butane jet delivered to the prechamber, ρ_{LPG} is the density of propane-butane, and W_{LPG} is the calorific value of propane-butane.

The increase in the indicated efficiency of the engine with a conventional combustion system, followed by an increase in the excess air factor fuel mixture to a value of λ = 1.3, reached a maximum value equal to 34.2%. Further blending of the mixture to a maximum value of λ = 1.5 resulted in a decrease in η_i for the engine and was associated with greater non-repeatability of engine cycles. In the case of the test engine with a sectional combustion chamber, the maximum indicated efficiency was dependent on the ratio of Q_{in}/Q_{tot} and was from 32% to 34.5%. For the 2.5% share of the energy load enriching the engine with a two-stage combustion system, the indicated efficiency was maximum and equal to 34.5%. The efficiency remained almost constant throughout the analysis of the excess air factor (Figure 14).

In conventional SI engines, the combustion process of the correct mixture takes place in a fairly narrow range of λ. Depletion of the fuel mixture to a level at which the value of the excess air ratio exceeds 1.5

causes irregular engine performance manifested by, inter alia, misfire and non-repeatability of engine cycles.

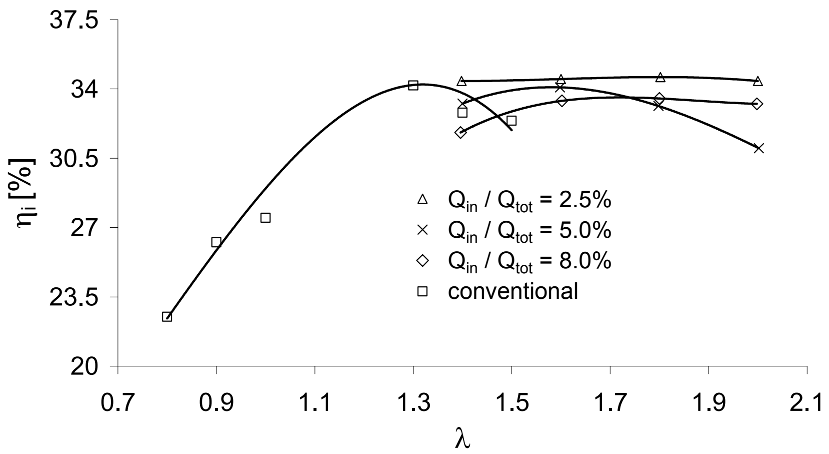

Fig. 14. Indicated efficiency of the engine for the single- and two-stage combustion systems

In the present study, the non-repeatability cycle phenomenon was defined by a factor of the non-repeatability factor of the indicated work engine COV_{Li}.

$$COV_{Li} = \frac{\sigma_{Lii}}{L_i} 100\% \tag{6}$$

where σ_{Lii} is the standard deviation of the indicated work and L_i is the mean value of the indicated work.

Standard deviation of the indicated work:

$$\sigma_{Lii} = \sqrt{\frac{1}{N-1} \sum \left(L_{ii} - L_i\right)^2} \tag{7}$$

where N is the number of measurements and L_{ii} is the value of indicated work in cycles.

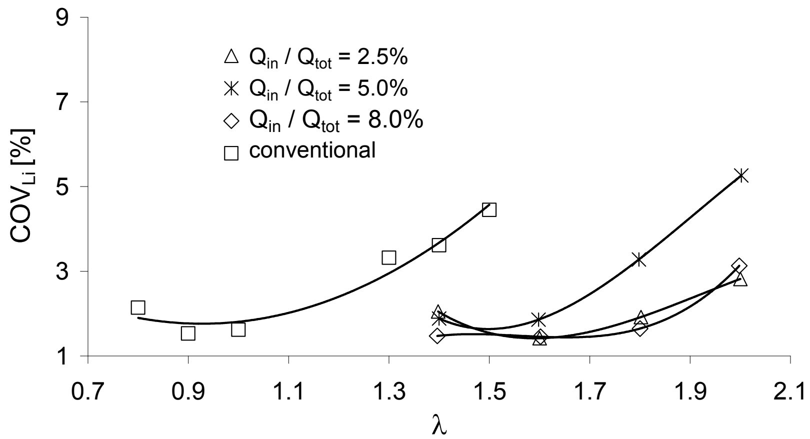

Fig. 15. Non-repeatability factor of the indicated work COV_{Li} for single- and two-stage combustion systems

Figure 15 shows a comparison of the non-repeatability factor of the indicated work COV_{Li} for the engine with a conventional combustion system and the engine with a prechamber. In the first case, depending on the value of the coefficient λ, COV_{Li} it ranged from 1.53% for λ = 0.9 to 4.45% for λ = 1.5, while in the second, it ranged from 1.45% for λ = 1.4 and Q_{in}/Q_{tot} = 8% to 5.26% for λ = 2.0 and Q_{in}/Q_{tot} = 5%. In the λ from 1.4 to 1.8 for the Q_{in}/Q_{tot} = 2.5% and 8% the coefficient COV_{Li} of the indicated work did not exceed 2% and was similar to the values achieved in the conventional engine with λ from 0.8 to 1.0. When burned in a two-stage system, of the poorest mixture, at $\lambda = 2.0$, the value COV_{Li} ranged from 2.82% for the Q_{in}/Q_{tot} = 2.5% to 5.26% for the Q_{in}/Q_{tot} = 5% and corresponded to the values achieved in the conventional engine at λ from 1.3 to 1.5.

Figures 16 to 18 show a summary of the registered emissions of toxic components in the exhaust of the engines with conventional and two-stage combustion systems throughout the analysis of the excess air factor. Using the gas analyser levels of NO_x, HC and CO were measured and recorded. The measured concentrations were related to the effective operation of the engine and expressed in g/kWh.

Effective work of the test engine:

$$L_e = \eta_m L_i \tag{8}$$

where η_m is the mechanical efficiency of the engine.

Mechanical efficiency of the test engine:

$$\eta_m = \frac{L_e}{L_i} 100\% = \frac{L_i - L_m}{L_i} 100\% \tag{9}$$

where L_m in the friction work (Heywood, 1988).

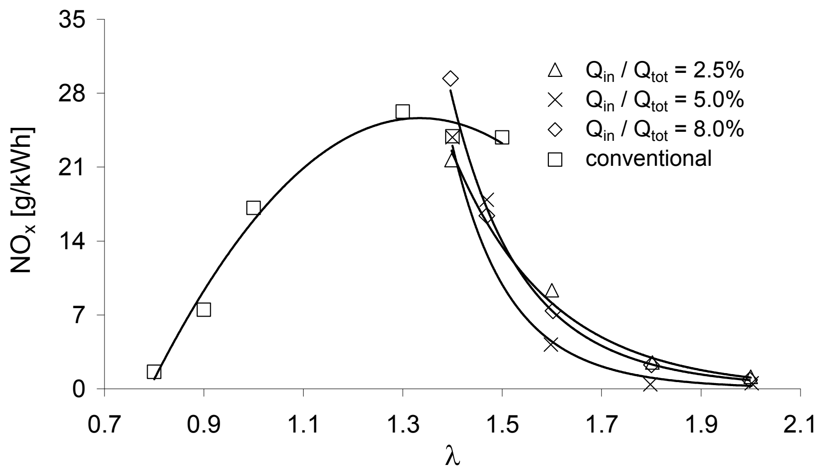

Fig. 16. Emissions of nitrogen oxides from the engine with single- and two-stage combustion systems

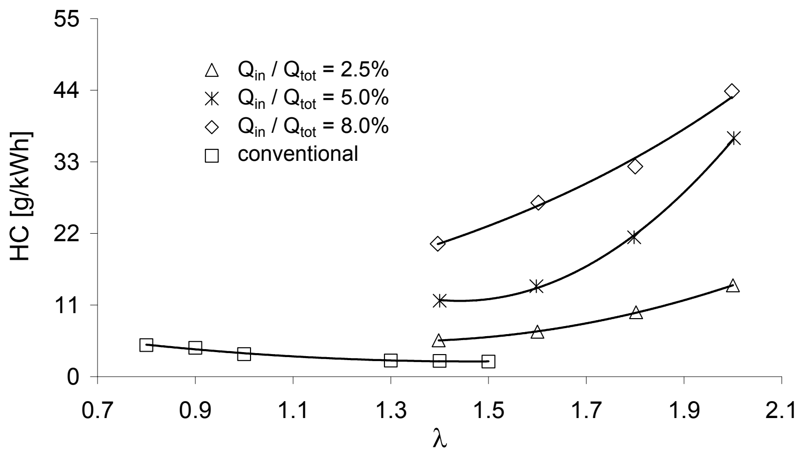

Fig. 17. Emissions of hydrocarbons from the engine with single- and two-stage combustion systems

For a conventional engine, an increase in the factor λ increased the content of nitrogen oxides in the exhaust gases, reaching the highest value of 26.26 g/kWh at $\lambda = 1.3$ (Figure 16). Increasing λ to a value of 1.5 decreased emissions of hydrocarbons, reaching a minimum equal to 2.32 g/kWh (Figure 17) and decreased emissions of carbon monoxide, for which the lowest value was 7.36 g/kWh (Figure 18). In the engine with a sectional combustion chamber, an increase in excess air factor decreased the content of nitrogen oxides in the exhaust of the engine and increased the amount of hydrocarbons and carbon monoxide. A decrease of the fuel energy of the prechamber compared to the total energy of the fuel supplied to the engine proved to be beneficial in the case of emissions of HC and CO. The test engine with the prechamber showed the lowest level of emissions of nitrogen oxides equal to 0.43 g/kWh at $\lambda = 1.8$, reached when working with a 5% share of load enrichment (Figure 16). With the increase of λ in the engine, the $Q_{in}/Q_{tot} = 8\%$ and reached the maximum concentration of HC and CO in the exhaust gas. For $\lambda = 2.0$, it was 43.86 g/kWh (Figure 17) and 31.15 g/kWh (Figure 18).

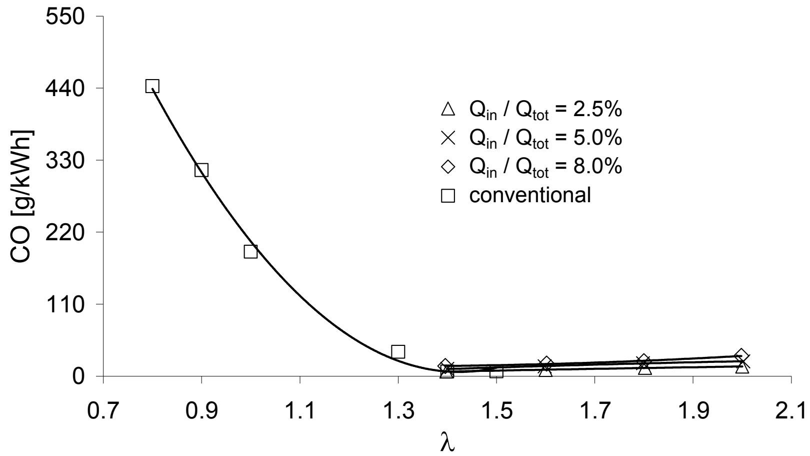

Fig. 18. Emissions of carbon monoxide from the engine with single - and two-stage combustion systems

The results of emissions of toxic components in the exhaust of the test SI engine with a prechamber performing a two-stage combustion process and the cylinder powered gasoline, for the $Q_{in}/Q_{tot} = 2.5\%$, expressed in g/kWh effective operation, were compared with the emissions standards for toxic exhaust components for engines fueled with liquid fuel, for HDVs (heavy duty vehicles) - EURO IV and EURO V (Table 4).

Table 4. Emission limit values of toxic exhaust components specified by the EURO IV and EURO V, for HDV motor vehicles powered by liquid fuel, and emission test engine with prechamber for $Q_{in}/Q_{tot} = 2.5\%$

		CO, g/kWh	HC, g/kWh	NO_x, g/kWh
EURO IV		4.0	0.55	3.5
EURO V		4.0	0.55	2.0
Test engine with two-stage combustion system	λ			
	1.4	7.21	5.59	21.67
	1.6	9.58	6.9	9.34
	1.8	12.94	9.88	2.5
	2.0	14.72	14.03	1.14

NO_x emissions for the experimental test SI engine, performing a two-stage combustion process in the sectional combustion chamber, and the cylinder powered by gasoline and working without catalyst, for the $Q_{in}/Q_{tot} = 2.5\%$ at $\lambda = 2.0$, did not exceed the limits contained in the NO_x emissions standards

EURO IV and EURO V for engines powered by liquid fuel for HDVs. The levels of HC and CO emissions in the whole range of air fuel factors for the test engine with a prechamber exceeded the limits contained in those standards.

However, the toxicity of engine exhaust measurements were carried out in conditions incompatible with the standards because they were made only at full load and not in the programming cycle of series load changes in the tests required under EURO IV and EURO V.

5. CONCLUSIONS

The test engine fuel mixture stratification method, with a two-stage combustion system in an engine with a prechamber, allowed to burn a lean mixture with an average excess air factor equal to 2.0 and thus led to lower emissions of nitrogen oxides in the exhaust of the engine. The test engine with a conventional, single-stage combustion process allowed to properly burn air-fuel mixtures with excess air factors not exceeding 1.5. If the value of λ is greater than 1.5, the non-repeatability factor COV_{Li} and engine efficiency decrease, which makes it virtually impossible for the engine to operate.

The energetic share of enriching fuel in the prechamber in an engine with a two-stage combustion system has influence on the level of toxic components in the exhaust emissions, the indicated efficiency, the indicated work and the stability of the engine work and repeatability of cycles. The best effects of the test engine were achieved for the lowest analysed energetic share of enriching fuel, which was 2.5%. The engine with a two-stage combustion process, working with $\lambda = 2.0$, the $Q_{in}/Q_{tot} = 2.5\%$, reduced the NO_x content in exhaust gases to a level of about 1.14 g/kWh. This value is significantly lower than the value obtained in the conventional engine, which worked at $\lambda = 1.3$, with a comparable non-repeatability of successive cycles (about 3%) and similar indicated efficiency (about 34%), was characterised by the emission of NO_x in the exhaust equal to 26.26 g/kWh. An increase in the excess air factor fuel mixture leads, in engines with either a single- or a two-stage combustion system, to a decline in engine performance, expressed as the value of the indicated work. The decline in L_i in the engine with a sectional combustion chamber with an excess air factor from 1.4 to 2.0 was on average 22.5%. However, the value of the indicated work can be increased by supercharging the engine and corrected by increasing the boost pressure. This solution is widely used among others in stationary gas engines, high-powered, lean mixtures and driving electrical generators.

This study showed that a two-stage combustion system in an engine with a sectional combustion chamber can be successfully used in industrial supercharged engines powered by lean mixtures because it achieves a small non-repeatability of cycles for $\lambda \geq 2$ and very low levels of NO_x in the exhaust gas. This study mainly concerns stationary gas engines, which are often supplied with fuel containing large amounts of non-flammable gases, greatly magnifying the non-repeatability of engine cycles.

SYMBOLS

CO	carbon monoxide
COV_{Li}	non-repeatability factor of indicated work, %
HC	hydrocarbon
L_e	effective work, MJ/m^3
L_i	indicated work, MJ/m^3
L_{ii}	value of indicated work in cycles, MJ/m^3
L_m	friction work, MJ/m^3
LPG	liquefied petroleum gas (propane-butane)
n	speed engine, rpm

NO_x	nitric oxide
p	pressure, MPa
Q_{cyl}	heat supplied to the engine cylinder, MJ
Q_{in}	heat supplied to the prechamber, MJ
Q_{tot}	total heat supplied to the engine, MJ
SI	spark ignition
T	temperature, K
t	time, min
TDC	top dead centre
V'_{in}	propane-butane jet delivered to the prechamber, m^3/min
V_{cyl}	volume of gasoline delivered to the engine cylinder, m^3
V_s	displacement volume, m^3
W_{gas}	calorific value of gasoline, MJ/kg
W_{LPG}	calorific value of propane-butane, MJ/kg

Greek symbols

η_I	indicated efficiency, %
η_m	mechanical efficiency, %
λ	excess air factor
ρ_{gas}	density of gasoline, kg/m^3
ρ_{LPG}	density of propane-butane, kg/m^3
σ_{Lii}	standard deviation of indicated work, MJ/m^3

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Received 07 December 2010
Received in revised form 27 September 2011
Accepted 06 January 2012