RESEARCH OF THE GAS ENGINE WITH TWO – STAGE COMBUSTION SYSTEM

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INTRODUCTION

The problem of atmospheric air pollution by the exhaust gas of piston engines, particularly in highly motorized countries, is presently one of the most important aspects of struggle for the protection of the natural environment of man. The necessity of limiting the exhaust gas toxic components and reducing fuel consumption has resulted in a change in the combustion engines design and development.

Among numerous harmful exhaust gas components, the emission of nitric oxides (NO_x) is the most difficult to limit. High temperature and excess of oxygen in the cylinder cause the formation of nitric oxide which transforms into nitric dioxide in the exhaust system and then in the atmosphere. The sum of these components is marked as NO_x. Nitric oxide creation speed increases as the temperature in the combustion chamber rises, especially above 1600 K. The burning of lead mixture leads to a decrease in the temperature of combustion process and is a method of the reduction of nitric oxide emission. It also increases engine efficiency. Sectional combustion chamber engines of heterogeneous mixture burning allow higher fuel depletion (approximately $\lambda = 2$) than it is possible in engines, which burn homogeneous mixtures (maximum $\lambda = 1.6$).

Heterogeneous mixture two-stage combustion system with sectional prechamber (rich mixture in small prechamber and very lean mixture in the other part of the combustion chamber) is mainly used in modern, stationary medium and high power gas engines with the cylinder diameter of above 200 mm. This system leads to an improvement in engine efficiency and causes high decrease in toxic components emission in exhaust gas, especially NO_x.

In recent years gas fuelling of car and stationary combustion engines is more commonly used. This is mostly caused by good characteristics of gaseous fuels in economical and ecological aspects. Gas fuelled combustion engines are characterized by lower toxic components content in exhaust gas in comparison to engines powered by liquid fuel. The interest in gaseous fuels is also connected with its lower price and greater natural resources. Hydrocarbon gaseous fuels like propane-butane (LPG) or natural gas (CNG) are used on a large scale for combustion engines fuelling.

Better knowledge of mixture formation and combustion processes in two-stage combustion system has a great influence on gas engine design.

The aim of the research was the analysis and better understanding of heterogeneous mixture formation and combustion processes in stationary gas engines of medium and high power.

TEST STAND DESCRIPTION

The test engine has been constructed on the basis of a four-stroke compression-ignition engine manufactured by “ANDORIA” Diesel Engine Manufacturers of Andrychów which, after some constructional changes, is designed for the combustion of gaseous fuel as a spark-ignition engine owing to a new fuel supply system and an ignition installation. The engine is a stationary, two-valve unit of a horizontal cylinder configuration. The engine block is made of cast iron and is integrated with the crankcase.

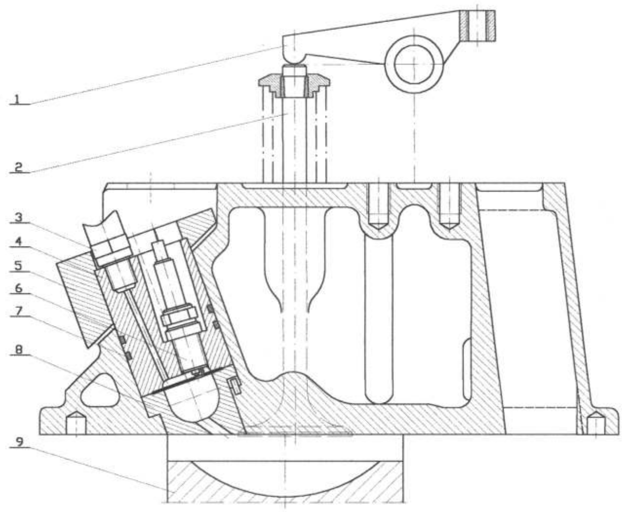

Fig. 1. Test engine head with prechamber: 1 – valve rocker, 2 – inlet valve, 3 – flame suppressor, 4 – prechamber body, 5 – retaining cover, 6 – spark plug, 7 – sealing ring, 8 – prechamber, 9 – piston

The main engine element that underwent modernization was the head. Introduced changes allowed an additional combustion chamber (prechamber) to be installed in the previously existing head of the S320 ER engine.

The change in the head shape called for making a new head casting. The additional combustion chamber was made of an alloy of enhanced strength properties at high temperatures, called Nimonic 90. In a chamber with a volume making up 4.5% of the total combustion volume, spark plug with M14×1.25 thread and piezo-quartz sensor with M7x0.75 thread were installed. 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 whose main elements are a set of non-return valves and a solenoid inlet valve. Positioned in a special water jacket, the non-return valves are water cooled during engine operation.

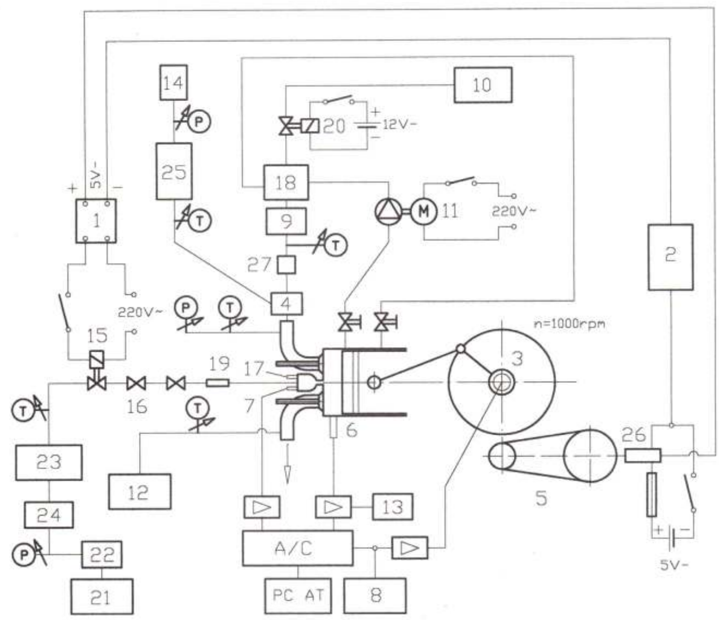

Fig. 2. Schematic diagram of the testing stand: 1 – electronic relay, 2 – pulse generating system, 3 – CA angle transmitter, 4 – gas mixer, 5 – belt transmission, 6 – main chamber pressure sensor, 7 – prechamber pressure sensor, 8 – counting module, 9 – gas flowmeter, 10 – main gas fuel tank, 11 – circulating-water pump, 12 – combustion-gas analyser, 13 – oscilloscope, 14 – measuring orifice, 15 – solenoid valve prechamber, 16 – non-return valve set, 17 – spark plug, 18 – reducer-evaporator, 19 – flame suppressor, 20 – gas solenoid valve, 21 – gas fuel tank prechamber, 22 – pressure regulator, 23 – pressure fluctuation damping reservoir, 24 – set of measurement rotameters, 25 – equalizing tank, 26 – magnetic induction sensor, 27 – metering valve

The process of enriching the fuel mixture was realized by propane-butane injection into the prechamber. This injection started in the time of compression stroke and lasted, depending on gas dose, for 30° to 75° CA and finished 45°CA before TDC. The initial, preliminary dose of additional fuel was obtained by setting on the gas regulator an adequately high supply pressure, which was between 0.2 and 0.6 MPa. Precise measure of the dose was realized by electromagnetic inlet valve time control system. The valve opening time at stable supply pressure measured the enriching dose of gaseous fuel. This system allowed fine control of the valve opening time under working engine. In the main fuel supply system the propane-butane mixer supply system was used. The gas was delivered from the gas cylinder of additional pressure of 0.8 MPa. The gas regulator was heated by water from the engine cooling system. The water circulation was provided by a suitable circulating pump. The gas mixer of radial slots was placed in the suction manifold of the engine. The mixture ratio regulation was realized by manually controlled dosage valve placed between the gas regulator and the gas mixer.

COURSE OF RESEARCH

The tests included three main measurement series allowing different ratio of the thermal energy delivered by the fuel to the prechamber, Q_{in}, to the thermal energy delivered to the whole engine, Q_{tot}. Pressures 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%, with excess air factor being changed between 1.4 and 2.0 and the ignition advance angle being changed between 6° and 22° CA (Crank Angle) before the TDC (Top Dead Centre). The recording was made for 97 successive operation cycles every 1° CA by using specialized software [4]. At the same time, other parameters necessary for the subsequent analysis of the indication results [3] such as: rotational speed, air consumption, gaseous fuel consumption in main chamber, gaseous fuel consumption in prechamber, air temperature, gaseous fuel mixture temperature, temperature of gaseous fuel in prechamber, combustion-gas temperature and ambient pressure and temperature were measured. Using an exhaust-gas analyzer, variations in the emission of engine exhaust gas components such as CO, CO_2, HC and NO_x, were measured.

The obtained results were compared to results of analogical measurements performed on SI engine of two-stage combustion system with prechamber powered by liquid fuel within the confines of [1] and [2].

RESEARCH RESULTS

On the basis of the determined dependencies of the effect of ignition advance angle on the efficiency η_i and indicated work W_i (Fig. 3, 4, 5), an optimal ignition advance angle of a gas engine as well as an engine with liquid fuelling was established at 12° CA before TDC. In order to compare the gas engine powered by propane-butane with the engine of the main chamber powered by gasoline, values of the indicated work, indicated efficiency, standard deviation of the indicated work and the amount of exhaust gas toxic components were determined for both engines for the same ignition advance angle.

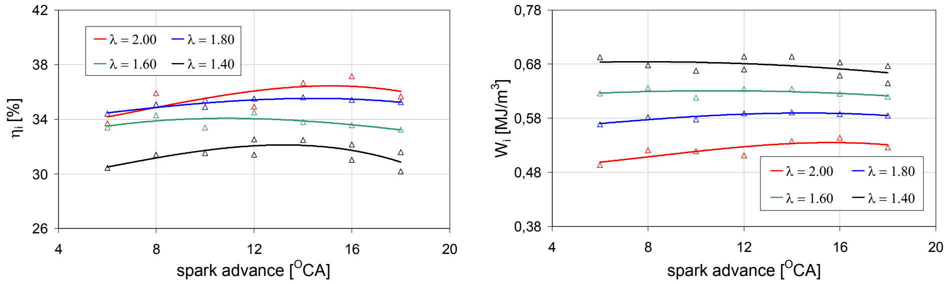

Fig. 3. Effect of ignition advance angle on indicated efficiency and indicated work of a gas engine for Q_{in}/Q_{tot} =2.5%

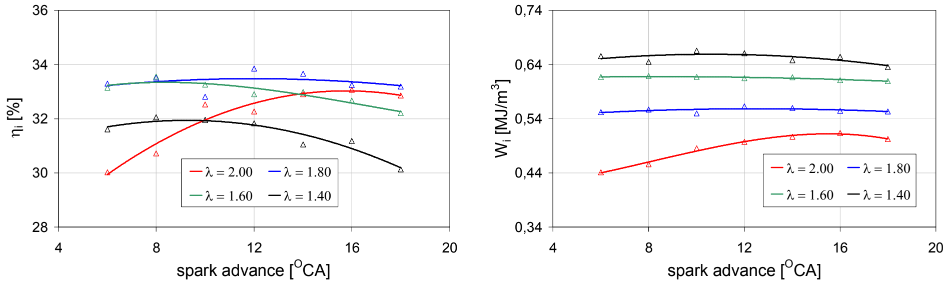

Fig. 4. Effect of ignition advance angle on indicated efficiency and indicated work of a gas engine for Q_{in}/Q_{tot} =5.0%

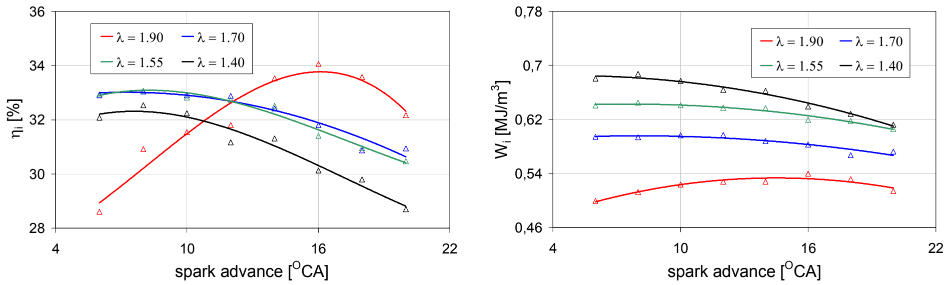

Fig. 5. Effect of ignition advance angle on indicated efficiency and indicated work of a gas engine for Q_{in}/Q_{tot} = 8.0%

The graphs based on the analysis of results obtained on the testing stand equipped with a gas engine indicate that similarly to engine of main chamber powered by gasoline, the variation of energetic share of fuel delivered to prechamber Q_{in} in total energy Q_{tot} delivered in fuel had a significant influence on the indicated work (Fig. 6) and efficiency (Fig. 7) of test the engine.

When the share of energy delivered in fuel to prechamber in total energy was the smallest and amounted to 2.5%, the indicated work was between 0.69 and 0.54 MJ/m^3 and it decreased with an increase of excess air factor. For Q_{in}/Q_{tot} = 5% at a similar decrease with an increase of λ the range of the indicated work value was between 0.65 and 0.5 MJ/m^3 and for Q_{in}/Q_{tot} = 8% between 0.66 and 0.53 MJ/m^3. The maximum values of the indicated efficiency for each of the three cases were achieved for different excess air factors. In case of Q_{in}/Q_{tot} = 2.5% the test engine indicated efficiency was between 32.5 and 35.5% and it reached its maximum at λ = 1.8. The greatest indicated efficiency of 32.5% for Q_{in}/Q_{tot} = 5% the test engine achieved at λ = 1.6, and for Q_{in}/Q_{tot} = 8% η_i = 33% at λ = 1.7.

The lowest indicated work covariation factor of 0.66% to 0.84% was obtained for each of the three considered shares of enriching fuel in prechamber at $\lambda = 1.6$ (Fig. 8).

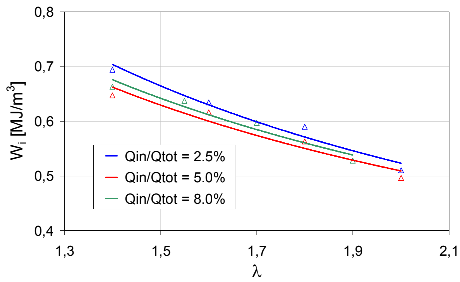

Fig. 6. Indicated work of 97 cycles of gas engine operation

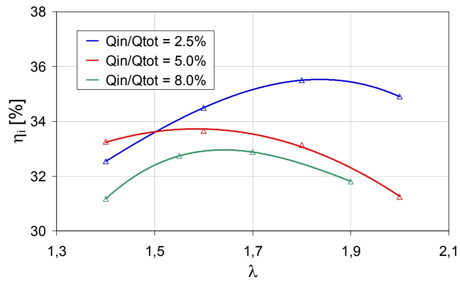

Fig. 7. Indicated efficiency of gas engine

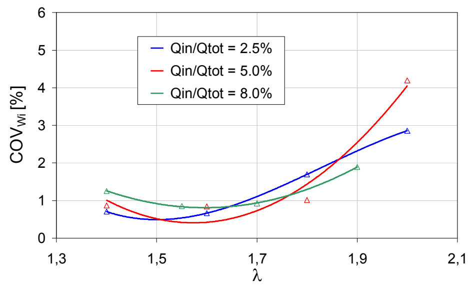

Fig. 8. COV_{Wi} factor indicated work of 97 cycles of gas engine operation

Within the range of the research, the toxicity of gas engine exhaust gas was measured and analysed. The analysis showed that similarly to gasoline powered main chamber engine the toxic components emission also depended on Q_{in}/Q_{tot} value.

The decrease of Q_{in}/Q_{tot} was favourable for the two-stage combustion system gas engine in the aspect of all the analysed exhaust gas components emission. At the lowest dose of enriching fuel the share of the burnt fuel in the first stage of combustion process in prechamber for high pressure and temperature conditions is the lowest. The content of NO_x (Fig. 9), HC (Fig. 10), CO_2 (Fig. 11), and CO (Fig. 12) in exhaust gas was the lowest for the case of $Q_{in}/Q_{tot} = 2.5\%$.

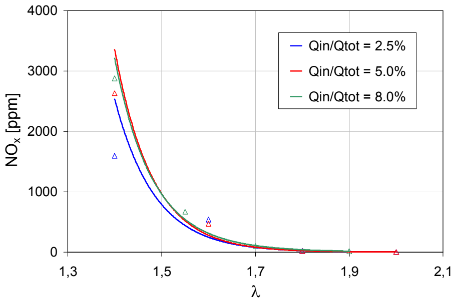

Fig. 9. Nitric oxide emission of gas engine

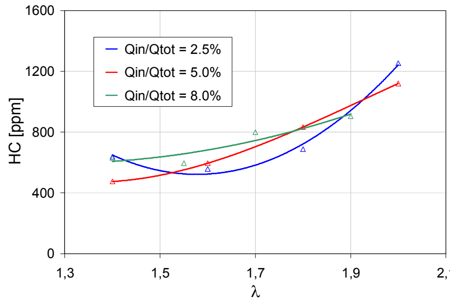

Fig. 10. Hydrocarbons emission of gas engine

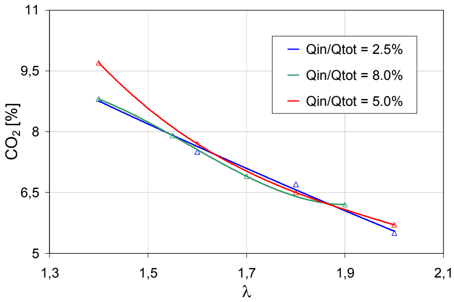

Fig. 11. Carbon dioxide emission of gas engine

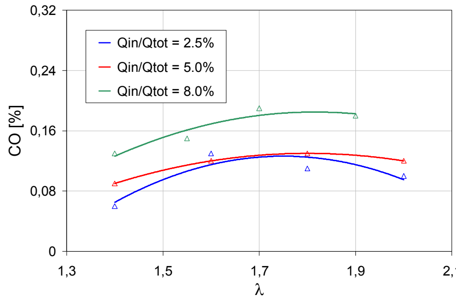

Fig. 12. Carbon monoxide emission of gas engine

THE INFLUENCE OF FUEL KIND ON SELECTED PARAMETERS OF THE SI TEST ENGINE WITH SECTIONAL COMBUSTION CHAMBER

Selected parameters of SI test engine with sectional combustion chamber determined for the cases of propane-butane and gasoline main chamber fuelling were compared during an analysis of fuel kind influence on those parameters in the test engine. The comparisons of indicated work and efficiency, standard deviation of indicated work and exhaust gas toxicity were made for the most favourable case of $Q_{in}/Q_{tot} = 2.5\%$.

Figures 13, 14, 15 show variations of indicated work, indicated efficiency and volumetric efficiency in function of excess air factor of gaseous and liquid fuel powered engines for the case of $Q_{in}/Q_{tot} = 2.5\%$. Indicated work in both engines decreased with an increase of excess air factor and for the gas engine it was between 0.69 and 0.51 MJ/m^3. For the liquid fuel powered engine it was between 0.7 to 0.55 MJ/m^3.

The maximum indicated efficiency value of 35.5% the gas fuel powered engine of $Q_{in}/Q_{tot} = 2.5\%$ reached at $\lambda = 1.8$ and the minimum value of 32.5% at $\lambda = 1.4$. In case of gasoline powered main chamber the indicated efficiency of the tested engine in the whole analysed mixture ratio range was almost at the same level of 34.5%.

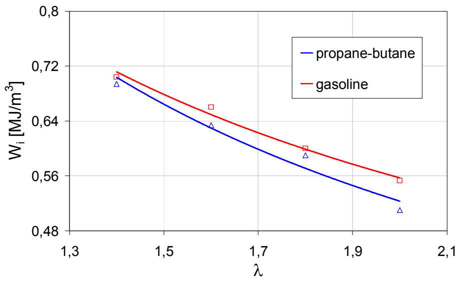

Fig. 13. Indicated work of an engine with gas and liquid supply system for Q_{in}/Q_{tot} = 2.5%

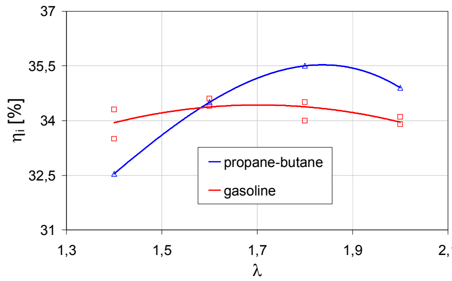

Fig. 14. Indicated efficiency of an engine with gas and liquid supply system for Q_{in}/Q_{tot} = 2.5%

The value of volumetric efficiency η_v determined on the basis of air consumption and fuel consumption measurement in both engines increased with an increase of excess air factor. In gas engine the value of volumetric efficiency η_v determined on the basis of air consumption measurement ranged from 0.76 to 0.82, however it ranged from 0.78 to 0.84 when determined on the basis of air consumption and gaseous fuel consumption. In liquid fuel powered engine the volumetric efficiency value was between 0.8 and 0.85.

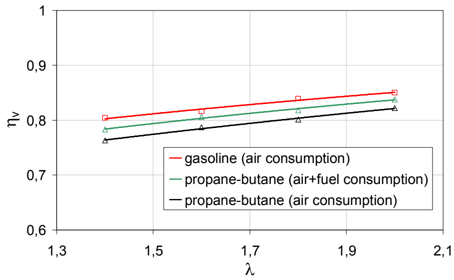

Fig. 15. Volumetric efficiency of an engine with gas and liquid supply system for Q_{in}/Q_{tot} = 2.5%

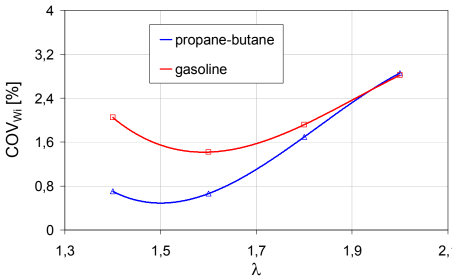

Fig. 16. COV_{Wi} factor indicated work of an engine with gas and liquid supply system for Q_{in}/Q_{tot} = 2.5%

Standard deviation of indicated work can be the measure of cycle non-repeatability of gas engine as well as liquid fuel powered engine.

Covariation factor of indicated work COV_{Wi} in function of excess air factor for propane-butane powered engine and gasoline powered main chamber engine in which Q_{in}/Q_{tot} = 2.5% was shown below. The minimal value of COV_{Wi} factor of a gas engine and a liquid fuel powered engine was obtained at λ = 1.6 and amounted to 0.66% in the first case and to 1.42% in the second case.

As it had been expected, propane-butane fuelling instead of gasoline fuelling had a significant influence on toxic components emission rate in exhaust gas of the test engine. The differences in emission of all the analysed toxic components like NO_x, HC, CO_2 and CO in function of excess air factor were similar in both supply systems.

The quantity of NO_x in exhaust gas increased with decrease of excess air factor and at λ = 1.4 it reached its maximum value of 2600 ppm for liquid fuel powered engine and 1593 ppm for gaseous fuel powered engine. The measure of NO_x concentration, taken with an exhaust-gas analyser of measuring error equal to 32 ppm at lean mixture of λ = 2.0 showed that in case of propane-butane powered engine the NO_x emission level was close to zero and in case of gasoline powered engine it was 94 ppm (Fig. 17).

With λ increase the HC content in exhaust gas increased up to 422 ppm in a gasoline powered engine and to 1254 ppm in a gas engine. The lowest value of HC emission equal to 558 ppm *Ws* reached by a propane-butane powered engine at λ = 1.6 and the lowest value for a gasoline powered engine was 245 ppm at λ = 1.4 (Fig. 18).

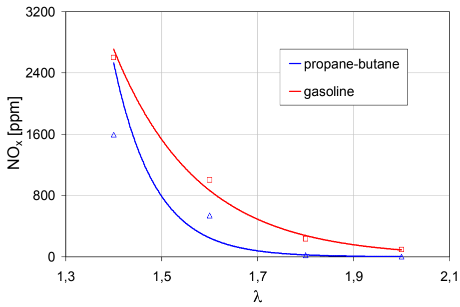

Fig. 17. Nitric oxide emission of an engine with gas and liquid supply system

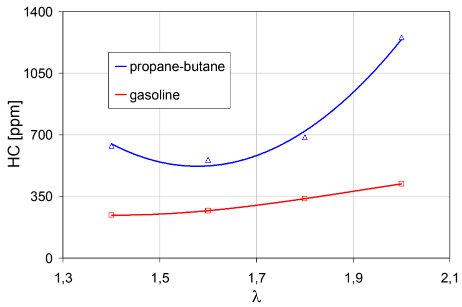

Fig. 18. Hydrocarbons emission of an engine with gas and liquid supply system

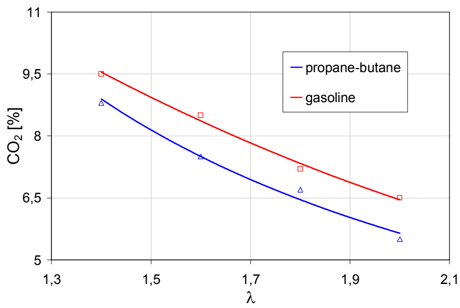

Fig. 19. Carbon dioxide emission of an engine with gas and liquid supply system

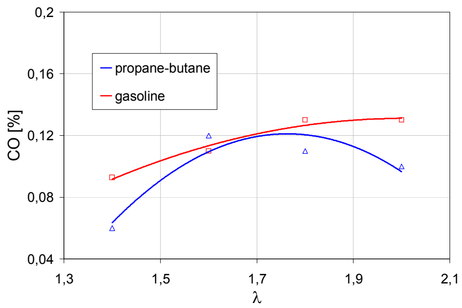

Fig. 20. Carbon monoxide emission of an engine with gas and liquid supply system

Carbon dioxide emission in both cases decreased with fuel depletion until the maximum value of excess air factor λ = 2.0 and for a propane-butane powered engine it was 5.5% whereas for a gasoline powered engine it was 6.5%. The greatest CO_2 concentration in exhaust gas was 8.8% for a gas engine and 9.5% for a liquid fuel powered engine (Fig. 19).

In case of Q_{in}/Q_{tot} = 2.5% the highest CO emission was 0.12% for gas engine at λ = 1.6 and 0.13% for gasoline powered engine at λ = 2.0 (Fig. 20).

CONCLUSIONS

The following conclusions can be formulated on the base of the comparison of the two-stage combustion system gas engine and the earlier research of a double-fuel version of the test engine [1, 2]:

1. The energetic share of enriching fuel in prechamber in a two-stage combustion system gas engine, as well as in a liquid fuel powered engine has an influence on the level of toxic components emission, the indicated efficiency and indicated work and also on the stability of engine's work and cycles repeatability. The best effects in both versions of the test engine were achieved for the lowest of the analysed energetic share of enriching fuel, which was 2.5%.

2. In a gas engine in the case of 2.5% share of enriching fuel in prechamber, the indicated work decreased with an increase of excess air factor and was between 0.69 and 0.54 MJ/m^3. In a liquid powered engine the indicated work decreased to 1.5% for the richest mixture of $\lambda = 1.4$ and to 8% for the leanest mixture of $\lambda = 2.0$. Lower values of the obtained indicated work can result from lower heat value of gas-air mixture in comparison with the mixture based on liquid fuel and from a worse cylinder filling by mixer based gas supply system than gasoline injection to suction manifold based supply system. The volumetric efficiency of gas engine determined on the basis of and air and gaseous fuel consumption measurement was approximately 2% lower than the volumetric efficiency in a liquid fuel powered engine. Lack of mixture cooling in inlet manifold as a result of fuel evaporation in liquid fuel powered engine has an influence on volumetric efficiency decrease of a gas engine. The volume of gaseous fuel is significantly higher than liquid fuel and considerably decreases the amount of air sucked into the cylinder.

3. The highest efficiency (35.5%) of a gas engine was achieved at $Q_{in}/Q_{tot} = 2.5\%$ and $\lambda = 1.8$. It was 1% higher than the highest value achieved in a liquid fuel powered engine. The indicated efficiency of a gas engine in the excess air factor range of 1.4 to 1.6 was lower in comparison with a liquid fuel powered engine. In a propane-butane engine the indicated efficiency at λ between 1.6 and 2.0 increased and was higher than in a gasoline engine.

4. A very good and minimum gas engine indicated work covariation factor nitric value of 0.66% for $Q_{in}/Q_{tot} = 2.5\%$ was achieved at $\lambda = 1.6$. The decrease of COV_{Wi} factor occurred in the whole analysed range of mixture composition. The greatest COV_{Wi} factor decrease from 2.05% to 0.7% was at $\lambda = 1.4$.

5. The maximum NO_x value of 1593 ppm in exhaust gas of propane-butane powered engine was measured for $Q_{in}/Q_{tot} = 2.5\%$ at $\lambda = 1.4$. The measure of NO_x concentration, which was done using an exhaust-gas analyser of the measuring error equal to 32 ppm in case of lean mixture of $\lambda = 2.0$ showed that oxide emission level in exhaust gas was close to zero. The nitric oxide emission decrease in comparison with liquid fuel powered engine was 90 ppm at $\lambda = 2.0$ and 1000 ppm at $\lambda = 1.4$.

6. The maximum carbon oxide emission value of a gas engine was 0.12% and it was similar to maximum carbon oxide emission value of gasoline engine which was 0.13%.

7. The lowest HC emission value of 558 ppm for Q_{in}/Q_{tot} = 2.5% was achieved by the gas engine at λ = 1.6 and the highest value of 1254 ppm at λ = 2.0. Gaseous fuel combustion in the main chamber of the test engine, in comparison with gasoline combustion, contributed to an increase of hydrocarbon emission in exhaust gas. For example HC concentration in exhaust gas of the engine at λ = 1.4 increased from 245 ppm to 638 ppm.

8. 1.8 is the optimum value of excess air factor in two-stage combustion system gas engine for Q_{in}/Q_{tot} = 2.5%. For this λ value sectional chamber gas engine is characterized by low cycle non-repeatability (COV_{Wi}<2%), the highest indicated efficiency of η_i = 35.5% and low NO_x emission. Oxidizing catalyst must be applied to reduce the CO and HC concentration in exhaust gas.

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SUMMARY

The article presents the results of laboratory research of SI engine with sectional combustion chamber where both main chamber and prechamber are powered by gaseous fuel. The obtained results are compared to an earlier analysis of two-stage combustion system engine with the main chamber powered by gasoline.