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CONTROL STRATEGY COMBUSTION OF SYSTEM WITH SEMI-OPEN COMBUSTION CHAMBER OF SPARK IGNITION ENGINES

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Abstract

The attempts to increase the engine combustion efficiency of the spark ignition (SI) engines have led to development of the new combustion system with semi-open combustion chamber. This system is similar to flame jet ignition systems, which were applied in many production internal combustion engines. The similar pulsed jet combustion system was elaborated by Professor A. K. Oppenheim. In the system developed at Aircraft Engine Department of Warsaw University of Technology (AED) the standard combustion chamber of SI engine was divided on prechamber and main combustion chamber using partition. The ignition in prechamber is more reliable and repeatable but total burning time in the prechamber and main combustion chamber is shorter than in standard combustion chamber. The principal problem of efficient operation of this system and obtaining of the required performance is securing the relevant control strategy. The performances of this system are dependent from: the rate of this prechamber volume to total combustion chamber volume, the orifice diameter in a partition, the ignition place and the ignition advance angle (ignition timing). Among these parameters the ignition timing only may be varied in the continuous manner during the engine operations without of the engine disassembling. Therefore constant values separate parameters (prechamber volume, orifice diameter, ignition place) should be selected in comprehensive, time consuming researches for different engine operation conditions (engine speed, load and environment conditions) and subsequently the map of ignition advance angles should be determined at state remaining separate parameters. The values of the ignition advance angle should be selected and programmed in electronic control unit, which will be control the ignition advance angle at different engine operating conditions. The dependences between engine operating parameters and the ignition timing are not linear and therefore mechanical control system is not effective. Some research results concerning the best engine operating parameters has been presented in this paper.

***Keywords**: combustion engines, spark ignition engines, combustion processes, new combustion system, new strategy of combustion control*

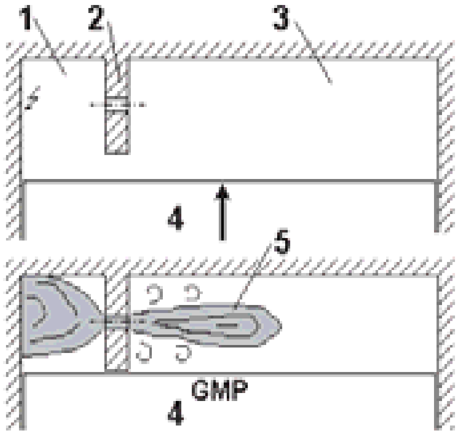
1. Introduction

Regardless of the operation combustion system mechanism, applied in spark ignition engines, there is always the problem of the system operation control at different: speeds, loads and ambient conditions. To all parameters of the engine, it is necessary to fit the amount of fuel and air and ensure that supplied fuel will be burned with the highest possible efficiency. The highest thermal efficiency can be obtained during operation on the lean charge, but lean operation causes difficulties with a charge ignition and exhaust aftertreatment. The most efficient exhaust aftertreatment can be achieved when the engine is running on the stoichiometric charge. But when working on the lean charge a bad performance of the engine may be obtained too [1, 2, 3 4 5 6, 7 8]. Obtaining comparable performances to stoichiometric engines, in the case of engines operating on the lean charge requires the supercharging what complicates the engine design. Because of the difficulty in the exhaust aftertreatment of engines, running on the lean charge, the rivalry between the GDI (gasoline direct injection) and PFI (port fuel injection) engines remains open mainly. PFI engines work on stoichiometric charge, so it can allow applying the high-quality aftertreatment exhaust systems TWC (three way catalytic system) of a high durability. In the case of GDI engines which run on both the lean charge and stoichiometric charge, and it is necessary to use storage catalysts which require periodic regeneration. At the rich mixture time of regeneration is provided, which causes a fuel consumption increase. As a result, fuel consumption when working on a lean mixture and stoichiometric mixture differs very little, while there is a large difference in system complications; the exhaust aftertreatment and combustion process control system. So we can, most likely, say that the rivalry between GDI and PFI can be resolved in favour of GDI engines, if the combustion process organization allows the obtaining of the engine emission output which will be compatible with the legislation requirements. The introduction of the catalytic combustion is one of the possible methods which would be enabling this. However, the combustion chamber surface elements contacting with burning mixture is too small to effectively afterburn and reduce exhaust gas components. It is need to ensure the relevant catalytic bed temperature, which, obtaining however is very difficult because the temperature in the combustion chamber of the piston engine, is permanently varied with engine speed and load, but during start up the temperature is very low. Another approach associated with the lean mixture engine operation is difficult of charge ignition. The reliable ignition can be obtained if the excess air coefficient is about 1.06, while using special high energy ignition system to 1.18 one. Therefore, it was introduced the method of stratified charge operation, in which the ignition initiation occurs in the similar to the stoichiometric mixture and another lean (from the stoichiometric) mixture layer ignite from travelling flame front. They are used also the prechambers with stoichiometric mixture, in which discharged the flame – jet ignition is applied to ignite a lean mixture in main combustion chamber. One of the possible methods of lean mixture combustion is to apply the HCCI (homogeneous charge compression ignition) system, in which the ignition occurs as a charge compression result similarity in diesel engine. This requires, however, increase of the initial temperature of the charge at the inlet to the combustion chamber, so during the compression stroke the ignition temperature has been reached [8, 9, 10, 11, 12, 13].

The proposed combustion system, developed in the Aircraft Engines Department (AED) of the Warsaw University of Technology is much simpler than the other known systems e.g.: GDI, PFI, HCCI. In this connection, that the effects of an individual systems use of are similar, but in the case of the ignition approach and exhaust aftertreatment system of homogeneous, stoichiometric charge is simpler, it was decided to use in this research the AED system. This allows the use of existing measuring methods of the fuel and air amounts. The most important issue, in the first stage of the work, it was to determine the combustion mechanism. Already in the first visualization studies, carried out with the use of the rapid compression machine (RCM), it was found that this mechanism differs substantially in relation to the operation in the standard combustion chambers of spark ignition (SI) engines. Therefore it was needed to propose the fixed parameters of the new combustion system such as: the rate of the prechamber volume to the total volume of the combustion chamber (sum of volume prechamber and main combustion chamber), the hole diameter in the partition, the ignition places and the most favourable values of the ignition advance angle for all conditions of engine operation. The cycle maximum pressure, combustion time, combustion speed, compression work and expansion work as a criterion of evaluation were adopted. It was found out that the best output results were obtained when the total combustion time in prechamber and main combustion chamber the shortest was. It was found that by using the new combustion system the burning time can be shorten, relative to the burning time in the standard combustion chamber of the engine. This requires, however, a proper selection of the control engine parameters. Two combustion chambers of different dimensions (the chamber volume, the hole diameter in the partition), were used in the tests with the use of RCM [14, 15, 16, 17, 18, 19, 20]. The test results identify the combustion mechanism in this system and show what is their impact on the specific control parameters. The combustion process courses versus time in the combustion chamber, applying different parameters of combustion system, has been recorded on the photographic film, using high speed photographic methods. Simultaneously the high speed pressure measurements were made, allowing the registration correlation of the pressure and a course of combustion. Analysis of the results shows that they are not linear in nature. Therefore, the engine control should be performed using the map of values of the ignition advance angle on the determined characteristics in the bench engine test. Engine control unit should choose the most advantageous an ignition advance angle in terms of engine speed and load.

2. New combustion system operation

Therefore, that this original design was based on the deducted considerations so the functioning rules require clarification and properly determination how it will be work during the engine operation, because course of experimental test is not always consistent with the foresight or assumptions.



at ignition the prechamber and the main combustion chamber are filled up with homogeneous, stoichiometric mixture and they are open

if piston is nearly TDC, the prechamber with burned gas is separated from the main combustion chamber and the burned gases begin to outflow from the prechamber to main combustion chamber

Fig. 1. Operating principle of the new combustion system: 1. Prechamber, 2. partition with orifice, 3. main combustion chamber, 4. piston, 5. stream of burned gases and radicals

Figure 1 shows the operation principle of the new combustion system by using a simple diagram of the combustion chamber of the SI engine. The combustion chamber was placed in the cylinder head. It was divided in the prechamber and the main chamber with the partition. Prechamber has volume several times less than the main chamber (the prechambers from two to eight times smaller than the main combustion chamber volume used in the study). In the partition separating the prechamber from the main combustion chamber the hole was made (it can be also a few smaller holes) through the burned mixture out flowed from prechamber to main combustion chamber. Mixture ignition occurs in the prechamber, using an electric spark igniter. Prechamber and main combustion chamber is supplied by the same engine intake system. Since the ignition initiation takes place in the prechamber, the pressure in the prechamber is growing faster than in the main combustion chamber. When the pressure difference between prechamber and main combustion chamber reaches the relevant value, the burning mixture outflows from prechamber to the main combustion chamber through a hole in the partition, igniting the successive layer of the mixture in the main combustion chamber. In view, however, that the prechamber is clearly separated when the piston is in the TDC only, the ignition in prechamber should be execute at the time enough advanced, to obtain the relevant pressure difference between the prechamber and main combustion chamber, allowing an outflow of burning mixture, when the piston is near the TDC. If this pressure difference is reached sooner or too late, the outflow may occur rather through the gap between the piston crown and the partition than through a hole in the partition, because the gap is several times larger than the cross-section area of the hole in the partition. This will cause the total change of the system operation rules. The shortest combustion time, including the prechamber and main combustion chamber, is obtained, as shown by the study, when the entire outflow of burning stream mixture goes through a hole in the partition. In addition, stream energy must be large enough to reach the opposite wall of the combustion chamber, which depends on the volume of the prechamber, the hole diameter in the partition, the ignition place and the ignition advance angle.

3. Experimental studies

3.1. Test stand and test apparatus

Tests were conducted using a rapid compression machine (RCM), specially made of for this purpose, to explore the combustion mechanism. Application of the RCM allows convenient use of the visualization techniques to combustion process together with the simultaneous measurement of high speed courses of pressure. Fig. 2 shows the schema of the RCM, and apparatus used in pressure measuring, and control devices.

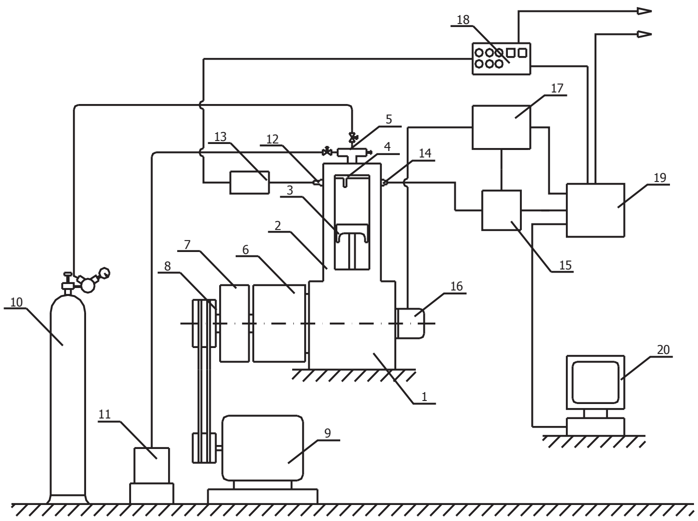


Fig. 2. Experimental set up schema. 1. Crank mechanism, 2. combustion chamber, 3. piston, 4. insert model combustion chamber, 5. refuelling and emptying system, 6. electromagnetic clutch, 7. flywheel, 8. external belt transmission, 9. electric motor, 10. pressurized bottle, 11. vacuum pump, 12. spark plug, 13. ignition apparatus, 14. piezoelectric transducer, 15. amplifier, 16. crank encoder, 17. indiskope 427, 18. ECU of optical system, 19. measurement card, 20. PC

Figure 3 shows a diagram of the optical system capable of taking pictures of the combustion process. The drum photographic camera and the films with very high sensitivity were used to registration of combustion process. High speed pressure measurements were performed using piezoquartz sensors and Kistler charge amplifier and AVL apparatus for research results acquisition. More information about the RCM and test equipment can be found in the literature [2, 4].

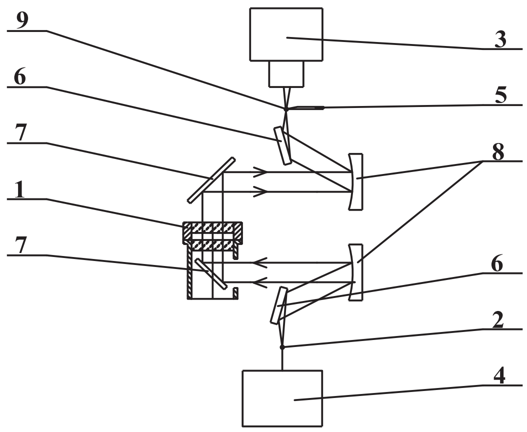


Fig. 3. Optical system, used in these experimental diagram: 1 - combustion chamber, 2, 9 - focus, 3 – drum camera, 4 – light source, 5 – optical knife, 6, 7 – plane mirrors, 8 – hemispherical mirror

3.2. Experimental procedures

Research on process control in the new combustion system, will be presented on the example of the two most characteristic configuration of the combustion chamber, because lack of space prevents representation of many other test configuration.

Variant A: prechamber with a volume of 20%, the hole diameter in the partition- 3 mm, the place ignition on the wall.

Variant B: prechamber with a volume of 28%, the hole diameter in the partition – 5 mm, the ignition place on the wall.

During the research each of the variants the of ignition advance angle has been changed and at each that ignition angle the course of combustion in the combustion chamber was photographed and the high speed pressure courses were registered. The pressure courses were recorded in one rotation of the crankshaft, from the beginning compression stroke to the end of the expansion stroke. The photographed combustion courses from the time a spark jump at the spark plug, by max. 8.4 ms what only allows the film length in the drum camera. Not always so it can be register the full combustion process, from the ignition to the end of the expansion stroke, but this range was the most important, because it allows determination of the stream range with different test parameters. A certain difficulty in interpreting the research results is that the pressure course were rerecorded in function of the crankshaft rotation angle but the combustion course was recorded versus the burning time from ignition. On the basis of the chart of pressure test results were determined: the maximum pressure value in the cycle, the growth pressure speed, the average pressure in the compression stroke, the average pressure in the expansion stroke, the useful work and the combustion efficiency. On the basis of the combustion photographs the flame front displacement and the combustion nature were determined. Comparing the results of pressure measurements and the combustion photo the selection of the most advantageous angle of ignition advance, for a given system configuration, from the performance point of view was accomplished.

The test results showed, that the combustion system design optimization requires selecting the fixed parameters of the system, which include: the prechamber volume, the hole diameter in the partition, the location of the ignition place, the shape combustion chamber and determination for each of the working conditions of the combustion system, the most favourable values of the ignition advance angle. The ignition advance angle is the only parameter which can be changed continuously during operation of the engine. Therefore it must be programmed in the engine control unit. Selecting the combustion system fixed parameters must be taken into account above all the ability to generate the stream in prechamber capable to travel with high speed the main combustion chamber. If the prechamber volume is too small, the stream energy outflowed through the hole is too small, to travel full the main combustion chamber. If the prechamber has too much volume, the burning time in the prechamber is too long and we do not achieve the desired effect. With regard to the diameter of the hole (or holes) in the partition, with too small a diameter the drag force are too large, reducing the stream range. With large hole diameter the outflow occurs at less pressure difference, which also affects in the stream range. Hence the choose problem is selection one or a few holes in the partition? What concerns the place of ignition in the prechamber, from placing the electrodes of the spark-plug depend on the combustion nature it has be either turbulent or laminar. This affects on the combustion time in combustion chamber, so the total combustion time in prechamber and main combustion chamber. The test appeared that the best results were obtained with the spark ignition on the wall, as the combustion is turbulent.

Of course, the combustion system constant parameters mutually influence on one to another and have not one simple prescription on the choice of the combustion system optimal parameters. This requires much tedious and time-consuming experimental and theoretical research, especially with regard wide application in transportation engines, which operate in variable loads and speeds conditions.

3.3. Test results-variant A

The pressure courses in the combustion chamber as a function of the crankshaft rotation angle (CA) obtained with different values of the ignition advance angle, from 20^0 to 45^0 CA BTDC the Fig. 4 show.

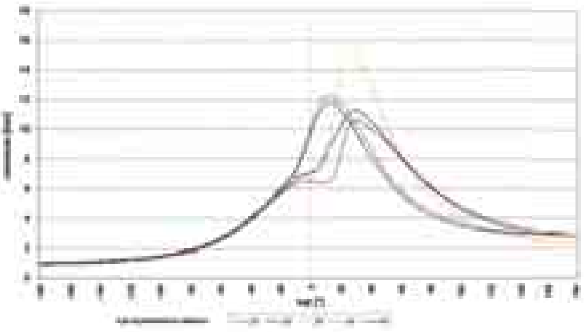


Fig. 4. The waveforms of pressure in the combustion chamber versus crankshaft rotation angle for different values of ignition advance angle

It is clear that the pressure course at the advance ignition angle range from 20 to 35^0 CA BTDC in compression stroke is similar to the course mixture compression without burning. In the case of the ignition advance angle of 40^0 and 45^0 CA BTDC is seen already clearly the influence of the too early ignition and compression of burning mixture.

Figure 5 shows the course of combustion, when the ignition advance angle was 20^0 CA BTDC. The lowest value of maximum pressure in this variant was obtained in this test. The pressure charts shows, that the pressure is initially increasing similarly as during compression without combustion and do not see of the changes even immediately after ignition. After passing of the position of TDC the pressure decreases slightly, because the increase in pressure, caused by the charge burning do not compensate the pressure drop as a result of the expansion. When the crankshaft turn by around 16^0 CA ATDC only, the pressure begins to grow, and after the turn 31^0 CA reaches the maximum pressure value 10.7 bars.

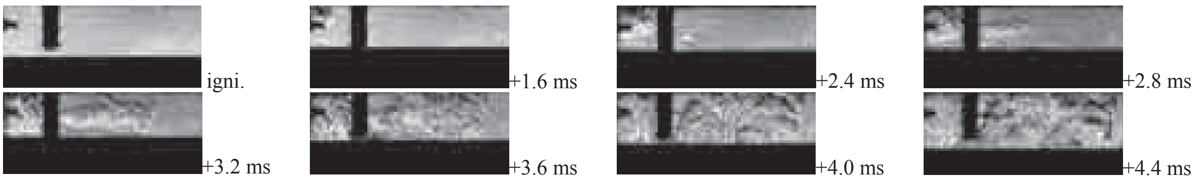


Fig. 5. The combustion development in the new combustion system: ignition advance angle 20^0 CA BTDC

Illustration depicting the combustion history shows that the stream outflows by hole from the prechamber to the main combustion chamber, follows at the time about 2.6 ms from ignition, when the piston is near the TDC. Initially, the stream quickly move through the main combustion chamber, but after turning 15^0 CAATDC, there has follow another stream outflow trough gap between the partition and the piston crown, which clearly braked the stream speed, which outflowed through the hole in the partition. It can see clearly in the stream displacement chart shown in Fig. 7.

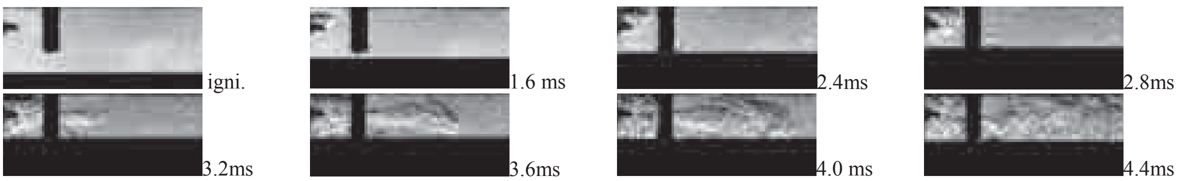


Fig. 6. The combustion development in the new combustion system : ignition advance angle 350 CA BTDC

Figure 6 shows the combustion course when the advance ignition angle was 35^0 CABTDC. The pressure chart shows that after ignition the pressure slowly increased, in relation to the compression pressure without combustion, and after passing about 10^0 CA ATDC started to grow rapidly, reaching a maximum of 15.3 bar, after the crankshaft turn around 27^0 CA ATDC. On the photo of the burning, you can see that the stream outflow from the prechamber to the main combustion chamber through hole occurred approximately 3ms from the moment of ignition just before reaching the piston TDC. Because of, the opening of the gap between the partition and piston crown occurred when the stream outflow through the hole in the partition reached already the opposite wall, so this has not already influence on the speed of movement stream through the main combustion chamber. So in effect, the shortest burning time, the maximum cycle pressure, and also the greater useful work and the highest combustion efficiency were obtained.

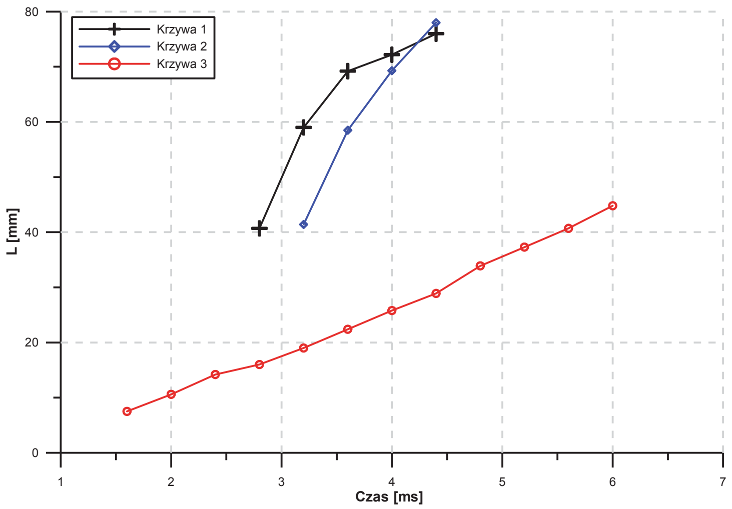


Fig. 7. The comparison of the stream front displacement for ignition advance angle 200 and 350 CA BTDC

Figure 7 compares the course of the stream front displacement, which outflows from the prechamber to the main combustion chamber as a function of time, for the ignition advance angle 20^0 CA BTDC and 35^0 CA BTDC, in relation to the new combustion system, and for comparison the standard combustion chamber of the engine. In the figure you can see that total burning time in the prechamber and main combustion chamber is much shorter than using standard combustion chamber. After combustion of the charge in the prechamber the substantial increase in the speed of combustion is observed; this is compliant with the system operation concept. The greater speed of combustion is obtained when the ignition advance angle was 35^0 CA BTDC, what reflected in higher performances, what it can be seen in Tab. 1. This is the effect of good parameter selections of the combustion system.

3.4. Test results: variant B

Figure 8 shows the pressure courses of in the combustion chamber, as a function of the angle of the crankshaft rotation, with different values of the ignition advance angle, from 15^0 to 45^0 CA BTDC. On the pressure graphs, you can see that for the values of the ignition advance angle : 15^0, 20^0 30^0 CA BTDC the ignition occurred too late and therefore after the piston passing by TDC position pressure growth due to combustion was not able to compensate the pressure drop caused by the charge expansion. In the cause of the largest ignition advance angle e.g. 45^0 CA BTDC it can be seen a clear increase in pressure at the compression side, which is reflected in the increased compression work. Fig. 9 shows of combustion course as a function of burning time expressed in milliseconds, the ignition advance angle 20^0 CA BTDC. On the pressure chart, it can be seen that after the ignition in prechamber the pressure growth was minimal and the pressure up to the crankshaft turn by around 35^0 CA BTDC, was similar as in the case of compression without combustion. Then the pressure increased from 4.7 bar to 5.9 bar, but and so the pressure was lower than in TDC (6 bar). On the combustion photos, you can see that only after the crankshaft turn by over 35^0 CA ATDC the burning mixture started to outflow from the prechamber to main combustion chamber.

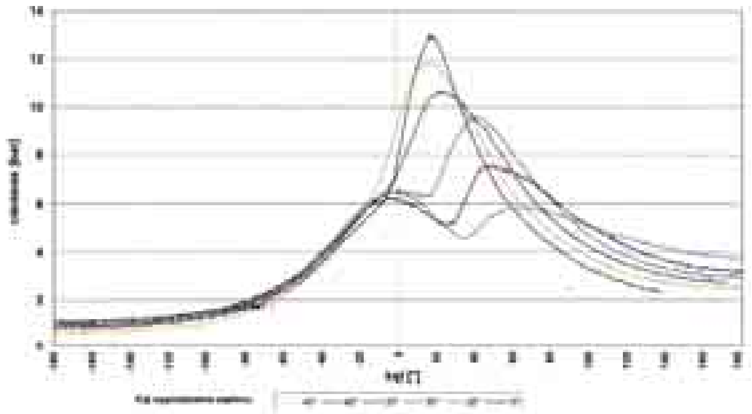


Fig. 8. The waveforms of pressure in the combustion chamber versus crankshaft rotation angle for different values of ignition advance angle

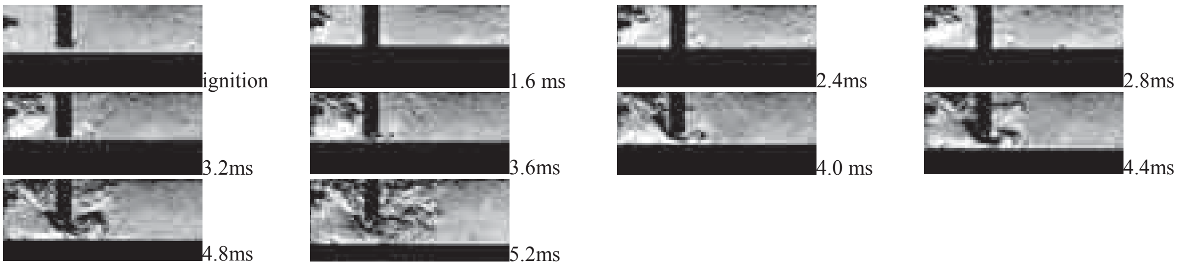


Fig. 9. The combustion development in the new combustion system : ignition advance angle 20^0 CA BTDC

However, because the gap between the partition and the piston crown has a much greater cross section than the hole in the partition, the bigger quantity of the charge from prechamber flows by the gap, and only small mass output flows out through the hole in partition. The outflowed charge from the prechamber is swirled on the partition edge, at the side main combustion chamber and as result the ignition in main combustion chamber follows slowly from the stream, which swirl direction causes braking of the stream displacement. Hence a little shorter combustion time caused by the faster charge burning in the prechamber only.

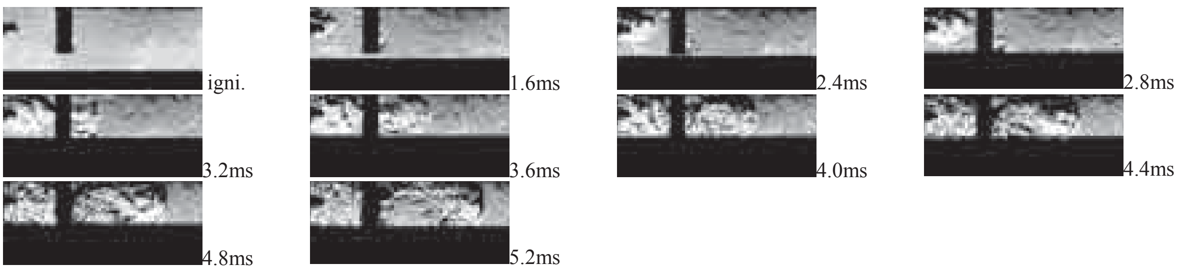


Fig.10. The combustion development in the new combustion system : ignition advance angle 350 CA BTDC

Figure 10 shows the course of combustion for the ignition advance angle 35^0 CA BTDC. On the pressure chart, it can be seen that the pressure from the ignition time increases permanently, and when the crankshaft passes the position of TDC for approximately 18^0 CA ATDC reaches maximum value, 13 bar. On the burning photos, it can be seen that the stream start outflows from the prechamber to the main combustion chamber at the time of approximately 2.9 ms since the ignition, just before reaching of the piston TDC position. However the stream energy, was probably too small to quickly travel through the main combustion chamber. This is probably due to the large hole diameter in the partition. Therefore, when after about 10^0 CA ATDC the gap between the partition and piston crown has opened up, the mixture part began to outflow through the gap between partition and piston crown. A distinctive stream swirling occurred on the edge of the partition, and the swirl direction is opposite to the direction of the stream displacement, which outflowed through the hole in the partition. As a result, it can be seen in the displacement stream chart, shown in Fig. 11 that the stream after start of outflow by the hole, its speed diminishes during flow throughout the main combustion chamber. On the chart can be see also the stream front displacement with ignition advance angle 20^0 CA BTDC. For comparison in the Fig. 11 you can see also the displacement flame front in the case of a standard combustion chamber.

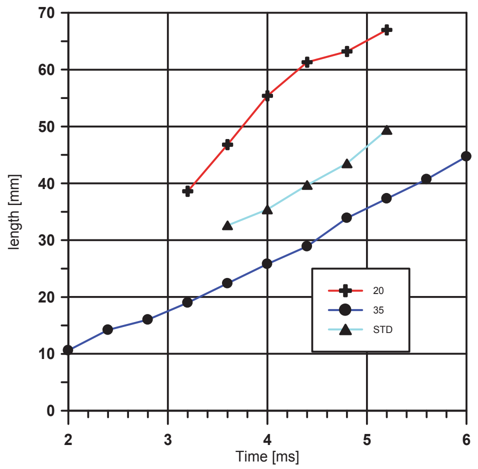


Fig. 11. Displacement flame front at the new combustion system and standard combustion system

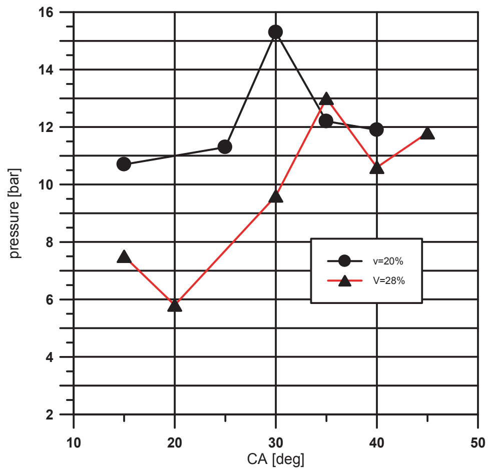


Fig. 12. The comparison of maximum cycle pressure as a function of ignition advance angle

Tab. 1. The research results of combustion system – variant A

	φ	p_{max}	φ p_{max}	Δ	η_{sp}
1	20	10.7	30.9	590.5	16.9
2	30	11.3	30.0	670.3	18.3
3	35	15.3	26.8	704.9	18.9
4	40	12.3	13.9	532.3	16.7
5	45	11.8	13.0	513.6	16.0

Tab. 2. The research results of combustion system – variant B

	φ	p_{max}	φ $_{pmax}$	Δ	η_{sp}
	^{0}CA	bar	^{0}CA ATDC	bar*^{0}CA	%
1	15	7.5	48.4	474.4	15.3
2	20	5.9	63.0	431.9	14.7
3	30	9.6	42.3	575.2	16.5
4	35	13.0	17.7	603.8	17.0
5	40	10.6	24.2	542.8	15.6
6	45	11.9	14.5	622.5	16.8

φ [^{0}CA BTDC] – *the ignition advance angle [crank angle before top dead centre],*
p_{max} [bar] – *maximum cycle pressure,*
φ $_{pmax}$ [^{0}CA ATDC] – *angle for maximum pressure [crank angle after top dead centre],*
Δ [bar^{0}CA]* – *the increment of turning pulse,*
η_{sp} [%] – *combustion efficiency coefficient.*

Table 2 compares the combustion system performance for the configuration variant (B).

Figure 12 compares the maximum pressure in the cycle as a function of the of ignition advance angle for both system variants. There is a clear maximum pressure and the highest system performance, for the system configuration (A), if we compared with the performance of configuration (B). The chart layout in the case of configuration B, shows that it is the possibility of existence of the several pressure maximum, which can to make difficult the choice of the most advantageous system configuration.

4. Summary

The researches to choose the control strategy in the new combustion system are presented in the paper. These researches were preceded by endeavour aimed at explaining the combustion mechanism in the new combustion system. This mechanism has a very strong impact on the control strategy. Until recently, the combustion systems applied in standard piston engines, the only such control parameters ware the ignition advance angle and the injection time (to ensure the stoichiometric mixture). Currently the systems are more complex and it is also controlled the compression ratio, the ignition timing, the valve timing, the supercharging ratio, and therefore in the new combustion system is required the choice of fixed system parameters and adjust for these, in all operating conditions the optimum values of the ignition advance angle.

The results presented in the paper concern on only one point of control, for given the engine speed and load, but the system used in the production engine would require determining the entire map of the characteristics. This shows how the huge work they need and as the costly are the researches to introduce the solution in mass production.

The engine control equipment in dependence on engine speed and load, should have a preselected of ignition advance angle map for each, the engine anticipated operation conditions, thereby improving the performance of the engine, by intensify of combustion process.

It is also possible to use other control parameters of the system in the new combustion system, such as: the compression ratio and the valves timing, which could lead to additional positive effects in the system operation.

Control in the new combustion system is similar, in terms of its complication, to the standard combustion systems of the SI engines and is certainly easier than the control systems of GDI engines.

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