Study on selected parameters of engine with the active combustion chamber

The present study was focused on the combustion engine with a variable compression ratio (VCR), namely the four-stroke air-cooled engine with the active combustion chamber (ACC). An indicated pressure, torque, power, and specific fuel consumption of that engine were investigated experimentally as a goal of the present study. Experiments were conducted using two versions of an engine. Two parameters particularly influencing the ACC engine performance including the maximum compression ratio CRmax and the indicator γp, determining the correct operation of the ACC system, were described. It was found that the ACC engine allowed avoiding detonation combustion without changing the amount and composition of the combustible mixture, and even without delaying the ignition advance angle. In addition, the possible range of control of the combustion process allowed the ACC engine to operate with different types of hydrocarbon fuels, for example, in the form of petrol with various alcohol admixtures. The very intense flow of the combustible mixture inside the cylinder of the ACC engine allowed describing the combustion in the ACC engine with zero-dimensional mathematical models with the dual Vibe function providing the proper characterization of the heat release process. The use of very high maximum compression ratios allows the ACC engine to operate to a certain extent as a Homogeneous Charge Compression Ignition (HCCI) engine with high lambda coefficients.

The study was aimed to determine an indicated pressure, torque, power, and fuel consumption of such an engine.

2. Present state of the development of VCR and ACC engines

2.1. The research on VCR engines

There are some studies on various VCR systems utilized in various engines. López et al. [29] reported that the use of two-stage VCR systems allowed the extraction of high thermal efficiency with high CR at lower loads and extended knock-free high load operation with low CR. However, slow CR transitions led to fuel consumption penalties, pointing to the need for optimizing the control strategies of the VCR system. After such an optimization fuel consumption was reduced up to 4% compared to the fixed CR.

Wittek et al. [54] applied a two-stage VCR system to a mass production engine. They reported that switching to low CR took last less than 1 s. The fuel consumption can be lowered at part load in a wide area of the map. The investigated VCR engine exhibited a slightly higher friction than the base engine.

Using a series of eight-cylinder engines supplied with gasoline, Caris and Nelson [13] investigated the effect of CR on volumetric efficiency, mean effective pressure and indicated thermal efficiency. The best performance of the engine was obtained at an intermediate CR.

Ferrey et al. [17] noticed that coupling the VCR engine with the variable valve actuation (VVA) technologies was beneficial for the combustion process therein. There exists thermodynamic benefit from the enhancement of the geometric CR (CR > 18:1) in combination with VVA compared to VVT strategies, thanks to the use of Atkinson/ Miller cycles. For the high compression ratio combustion
chamber geometries used with early or late intake valve closing strategies, an indicated efficiency enhanced up to 12 or 13% between compression ratio 10:1 and 18–20:1 at low loads (BMEP < 8 bar).

Nuthan et al. [38] conducted studies in wide open throttle condition (WOT) for speed range from 1200 rpm to 1800 rpm on a single-cylinder four-stroke VCR SI engine supplied with an equal-volume blend of methanol/gasoline fuel. The 14° BTDC ignition timing was maintained for all three different CRs (8, 9 and 10). Enhancing the latter from CR8 to CR10 for the methanol/gasoline blend increased combustion efficiency by enhancing the peak pressure and net heat release value by 27.5% and 30%, respectively at a speed of 1600 rpm. They reported that at CR10 at all engine speeds there was a 25% increase in BTE, and BSFC reduction by 19%, lowering CO and HC emissions by 30–40%. The NOx emission enhanced with the increase of CR.

Qasemian et al. [40] studied effects of varied CR of biofuel SI engines on their thermal balance and waste heat recovery potential. They found that the heat loss to brake power ratio ($Q_h/\dot{W}_b$) was enhanced with the increase in the compression ratio. In addition, an enhancement of the compression ratio lowered the exhaust power to heat loss ratio ($Q_e/Q_h$) for all studied fuels. There existed a direct relationship between the ethanol in fuel content and $Q_e/Q_h$ ratio. As the percentage of ethanol in fuel enhanced, the $Q_e/Q_h$ ratio rose. Thus, the more the ethanol in the fuel and the less the compression ratio, the more the potential for the waste heat recovery of the IC engine.

Abdel and Osman [1] investigated the influence of the variable CR on the engine performance with various ethanol-gasoline fuel blends. They found that, for each fuel blend, there is an optimum CR giving the maximum indicated power.

Ozcan and Yamin [39] elaborated a computer simulation model of a variable-stroke-length LPG (liquefied petroleum gas) supplied, single-cylinder, four-stroke, SI engine. Numerical analysis was carried out at various engine speeds and corresponding power and torque.

Yuh and Tohru [58] studied the effect of higher CRs in two-stroke engines. They found a decrease in fuel consumption for each unit rise in CR in the range of 6.6–13.6. The maximum CR was limited due to knock and enhanced in thermal load.

Some researchers were also focused on emission analysis for a VCR engine [3, 24].

Studies on two-stroke and four-stroke engines showed the increased performance of the engine with VCR [2, 21].

The gain in efficiency beyond a certain CR can be strongly limited due to other influencing factors such as heat loss and friction [45].

There exists the upper limit of CR for the combination of fuel and SI engine beyond which there is fall in efficiency [49].

Roy et al. [46] noticed that compression ratio strongly affected the performance of an SI engine at different load conditions. Therefore, the VCR engine can be an effective solution to the problems encountered while operating an engine at varying load conditions. VCR engines allow more flexibility to the user, improvement in brake power, brake thermal efficiency, and torque while weakening emissions and specific fuel consumption.

Bas et al. [6] studied brake power and brake-specific fuel consumption of a variable compression ratio spark-ignited engine equipped with different spark plugs (conventional, iridium and platinum). They investigated the effects of hydrogen enrichment and VCR on engine operation at various speeds. They obtained higher performance values with iridium and platinum spark plugs compared with the conventional spark plug at all hydrogen fractions, engine speeds and compression ratios. The platinum spark plug enhanced the engine performance more than the iridium one.

Hammermueller et al. [23] found that especially for turbocharged SI engines, the variable geometrical compression ratio is beneficial in terms of part load efficiency and maximum power output. Using a conrod-based two-stage VCR system with an eccentric piston pin bearing, they analyzed the influencing parameters resulting from the fired engine operation, such as dynamic pressure stimulation in the crankshaft oil supply or the acceleration of the internal oil columns. The tests were conducted on a three-cylinder full engine in the entire engine map, for the speed range up to 6500 rpm, loads up to BMEP equal to 21 bar both in high and low compression ratio. They found, that only so called “hydraulic pretension” of the support chamber provided stiff force transmission in the low compression ratio stage.

Basu et al. [7] studied the performance of a VCR SI four-stroke engine with electrical loading and air cooling supplied with the blends of butanol and ethanol with proportions 10, 20, 30 and 40% in petrol. The brake thermal efficiency and brake specific fuel consumption (BSFC) were compared for the above-mentioned blends for different compression ratios and loads. They found that the brake thermal efficiency enhanced and the BSFC weakened with an increase of blend amount in petrol and no knocking occurred during the engine operation. The engine ran smoothly up to 40% blends in each case. The maximum efficiency was obtained for a 40% blend.

Using the homogeneous charge compression ignition engine supplied with gasoline fuel, Babagiray et al. [5] studied the effect of compression ratio, engine speed inlet air temperature, lambda value, and research octane number value on indicated mean effective pressure, indicated thermal efficiency, maximum pressure rise rate, specific fuel consumption, cyclic differences, and emissions including unburned hydrocarbons, carbon monoxide, and nitrogen oxide. The engine operation condition characterized by the compression ratio of 12, intake air temperature of 333 K, lambda value of 1.8, engine speed of 935 rpm, and RON40 was found as the optimal one for the HCCI engine. At this point, 5.08 of indicated mean effective pressure, 35% of indicated thermal efficiency, 243.28 g/kWh of specific fuel consumption, 4.43 bar/CA of maximum pressure rise rate, and 3% of COV$_{\text{IMEP}}$ occurred. The optimum values of engine emissions were of 355,586 ppm for UHC, of 3% for CO, and very small value ppm for NOx, respectively.

Depending on the performance necessity of a vehicle its VCR engine operates at compression ratios being varied via changing the combustion chamber volume. It is thermody-
namically beneficial, as the engine map shows [52]. At low power levels, such an engine operates at a higher CR to capture low fuel consumption while at high power levels the engine operates at low CR to limit knocking. The optimum CR is affected by inlet air temperature, engine coolant temperature, exhaust gas temperature, engine knock, fuel type, the octave rating of fuel, and other parameters. Also, the operating temperature is maintained at optimum allowing obtaining high combustion efficiency.

Rakopoulos et al. [41] elaborated a comprehensive, quasi-dimensional, two-zone combustion model allowing the prediction of the combustion characteristics, performance, nitric oxide (NO), and carbon monoxide (CO) emissions of high-speed spark-ignition engine. The model is validated on an experimental, Ricardo E6, mono-cylinder, high-speed SI engine operational over a wide range of compression ratios and (fuel-air) equivalence ratios (EQR). The engine was supplied with gasoline and operated under various CR and EQR values at WOT position.

2.2. The VCR strategy realization

The VCR strategy can be realized through [47, 56]:
- articulated monohed (Saab)
- piston of variable deck height (Daimler-Benz)
- eccentrics on crankshaft bearings (FEV)
- multilink rod-crank mechanisms (Nissan)
- secondary moving piston or valve in cylinder head (Ford)
- gear-based crank mechanisms (MCE-5)
- precisely shifted cylinder block with cylinder head assembly (Self-Developed Design).

Some realizations of VCR strategy include moveable cylinder block [14, 55], use of eccentric bearings for crankshaft and connecting rod and use of adjustable lever rod between crankshaft and connecting rod [8, 11, 22, 25, 33-35, 42, 43, 45, 50].

Roy et al. [46] elaborated the VCR engine with a cylinder head which was equipped with an actuator-driven movable ram for varying the clearance volume, and thus the compression ratio.

Wos et al. [56] elaborated and test engine with variable combustion chambers volume realized by the shifting of the cylinder block with head assembly perpendicularly to the crankshaft axis, causing the variation of the engine CR from 19 to 9.

López et al. [29] elaborated a two-stage VCR system applied to a downsized SI engine. The VCR mechanism comprised an eccentric element in the small end of the connecting rod, rotated to enhance/lower the effective connecting rod length, achieving the CRs of 12.11:1 and 9.56:1.

Wittek et al. [54] developed a VCR system comprising a connecting rod with eccentrically piston pin suspension and hydraulic moment support. Stiffness of support mechanism was affected by engine speed and load. System was robust against variations of pressure, temperature, and aeration in the oil feed stream. Dissipation effects occurring in hydraulic support pistons were negligible.

Brevick [10] developed a "Pressure Reactive Piston (PRP)" technology with modification of the piston geometry. The mechanism had a secondary piston in the secondary cylinder, which reciprocated inside the combustion chamber. The spark plug was installed in the secondary piston, which lowered the space requirement for exclusive mounting of the secondary piston.

Lin and Yang [28] proposed a dual shaft control variable compression ratio (DSC-VCR) engine based on a gear-driven eccentric sleeve. This DSC-VCR allowed for double larger gears to share the load, and for the engine operation with a larger eccentric size and a narrower adjustment range than for the other similar mechanisms. This mechanism allowed the engine operation with a larger overexpansion ratio (OER, the ratio of expansion stroke and compression stroke) under all conditions to increase engine efficiency.

The VCR engine comprising two control shafts rotating 180 degrees in the same direction to adjust the CR ratio was described in [32]. Timing gears were utilized for the rotation of the control shafts with two sets of journals. The first set of journals was concentric with the timing gears and mounted in conventional bushings in the crankcase. The second set of journals was eccentrically located on the control shaft. The eccentricity provided a change of cylinder head height relative to the crankshaft. The eccentric journals were mounted in eccentric bushings. The eccentric bushings were mounted in the cylinder jug.

The interesting realization of VCR in SI engine was achieved using the concept of the active combustion chamber (ACC) [20], presented in the next subchapter.

2.3. The ACC engine

The ACC engine is an engine in which it is possible to change not only the compression ratio, but also the method / degree of expansion, which in a way influenced the introduction of the name of the engine as a unit with an active combustion chamber. It is the inherent ability of the ACC engine to change the expansion ratio, which is practically not used in the currently used engines. The favorable course of the combustion process, and indirectly of charge exchange, characterized, inter alia, by the absence of detonation combustion, obtained by controlling the compression ratio, is mainly visible at very high maximum compression ratios above 20:1 [20, 21]. However, the existing limitations as to the possibility of increasing the overall efficiency of the combustion process in the implementation of the VCR strategy prompted the authors to consider the use of the additional degree of freedom to control the combustion process and charge exchange, which is provided by the variable expansion ratio to increase the said efficiency.

Figure 1 shows schematically the principle of operation of variable expansion in the ACC engine.

The diagram in Fig. 1 shows the essence of the ACC engine operation using the four mutual positions of the crank-piston system and the movable piston in the ACC system. The first item is when ignition occurs. The second is TDC, the characteristic position of the crank-piston system with a slight displacement of the additional piston from the ACC system. The third position is the maximum displacement of the auxiliary piston from the ACC system and a slight displacement of the crank/piston system. The last fourth is the completion of essential ACC operation (reaching the first minimum) of the engine cycle.
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Figure 2 showed changes in the compression ratio in an engine with the ACC system within a fragment of one working cycle. The changes in the compression ratio were presented against the compression ratio values of the old and new SI engines, and the selected compression ratio values determined by the control system, achievable in an engine with a VCR system. The width of the blue lines corresponds to the observed fluctuations in the CR value.

In the exemplary ACC engine, the compression ratio, which the ACC automatically varied from a maximum of 17.5:1 to 16.7:1 over time, could be increased by only 0.8 per cycle without causing detonation in the ACC engine cylinder. In the expansion stroke of the engine, the changed CR value, due to the ACC system operation, even reached the value of 13.2:1.

3. The indicator determining the correct operation of the ACC engine

For the ACC engine, one indicator was distinguished, which was called the main design parameter of the ACC system, denoted by the symbol $\gamma_{fm}$. When introducing the indicator $\gamma_{fm}$ to the analysis, which is responsible for the dynamic properties of the ACC system, it was assumed that it should not be associated with the pressure unit.

The indicator $\gamma_{fm}$ was defined by the ratio of the additional area in the combustion chamber expressed in cm² to the mass expressed in kilograms of the complete additional piston, i.e., also considering the masses of sealing rings and O-rings connected to it. The indicator $\gamma_{fm}$ was described by equation (2).

$$\gamma_{fm} = \frac{\pi d_i^2}{4m} - 0.7854 \frac{d_i^2}{m}$$

where: $\gamma_{fm}$ – the indicator determining the correct operation of the ACC engine [cm²/kg], $d_i$ – diameter of the additional piston in the combustion chamber [cm], $m$ – mass of the additional piston [kg].

The diameter $d_i$ was shown in Fig. 3, while the other diameter $d_2$ also shown therein was selected in such a way as
to obtain the highest possible value of the indicator $\gamma_{\text{fm}}$ while ensuring good pneumatic support properties.

![Diagram](image)

Fig. 3. Additional piston ACC system with its characteristic diameters $d_1$ and $d_2$.

It should be added, it is desirable that its value does not deviate from the number one hundred taken as the reference value, since then the values of the phase shift after TDC corresponding to obtaining the maximum displacement of the ACC system piston with respect to the main piston were close to the minimum. During the iterative numerical and experimental design of an additional ACC system with an additional piston made of aluminium alloys, a favourable value of the indicator $\gamma_{\text{fm}}$ was obtained, equal to 88 cm$^2$/kg. In the case of using magnesium alloys for this purpose, the value of 100 cm$^2$/kg was successfully exceeded even without obtaining the optimal strength properties of the piston [16].

During several years of experimental research, it was also noticed that for the proper engine functioning, the value of the indicator $\gamma_{\text{fm}}$ should not be less than 75 cm$^2$/kg. Otherwise, the reaction of the system will be so delayed that the losses may outweigh the benefits, or, in the extreme case, the engine cycle will be distorted (broken), which has been observed many times during the tests.

This note does not apply to slow speed engines, i.e., not exceeding 1200 rpm, where lower values may apply due to the available cycle times, especially in large cylinders above 1000 cm$^3$. In such cylinders, the concept of multiplying the ACC system could also be possible or the use of more favourable proportions in the system itself, due to the amount of free space available in the cylinder head to be used. However, this requires further research.

At the initial stage of the research, in one of the tested ACC engines, solutions were applied in which the value of the indicator $\gamma_{\text{fm}}$ was less than 70 cm$^2$/kg, or even less than 66 cm$^2$/kg, but the effects were completely destructive additionally combined with uncontrolled detonations before and after the TDC point. At the current stage of construction development, the indicator threshold $\gamma_{\text{fm}} = 88$ cm$^2$/kg has been exceeded, thus the first positive results, for a $\text{CR}_{\text{max}}$ value above 19:1, were obtained.

The graph in Fig. 4 shows the values of the previously and currently used values of the indicator $\gamma_{\text{fm}}$ and the range of their effective use is marked with a black dashed line. It should be emphasized that the black dashed line is an imaginary line, especially in the horizontal course of the value, which is primarily the result of the research observation, while its curvilinear part is a parabola reflecting uniformly accelerated motion with the assumed initial speed of an additional piston in the ACC system, calculated in a simplified mathematical model [16].

![Graph](image)

Fig. 4. The effect of the indicator $\gamma_{\text{fm}}$ on the achieved maximum ACC engine speed

It can be seen from Fig. 4 that the increase in value corresponds to an increase in maximum engine speed ACC, but for that increased $\text{CR}_{\text{max}}$ values can also be obtained. Unfortunately, the value of the main design parameter should not be associated directly with $\text{CR}_{\text{max}}$ value, because there is always such a value of $\text{CR}_{\text{max}}$ beyond which the phenomenon of detonation combustion occurs, caused by exceeding the auto-ignition temperature during compression. The reaction time of the additional piston can be too long to prevent such combustion even for the indicator $\gamma_{\text{fm}}$ values above 100 cm$^2$/kg.

4. Materials and methods

The experimental tests were carried out on test stands (engine dynamometers) in which two different types of brakes were used, loading the tested ACC engines. The first was a water brake that enabled smooth load regulation and the use of wide range of loads and rotational speeds of the tested engine. This type of brake was used to measure an original air-cooled Fiat 126p twin cylinder SI engine (652 cc, 18 kW/4500 rpm, 42 Nm/3200 rpm) with a modified cylinder head incorporating an ACC system allowing a $\text{CR}_{\text{max}} = 17.5:1$. The second brake, on the other hand, was in the form of a generator (power generator) with known characteristics, connected with resistors of specific resistance and a voltmeter, enabling the load power to be determined.

Another brake was used to load the air-cooled CI engine driving the original generator set PROTON OASIS-3. This air-cooled single-cylinder direct injection (DI) diesel engine (463 cc, 6.2 kW/6.8 kW at 3000 rpm) was modified to be SI one by introducing a cylinder head with the ACC system enabling the achievement of $\text{CR}_{\text{max}} = 18.5:1$ and $\text{CR}_{\text{max}} = 19.3:1$ in a stepwise manner. The use of the existing generator set to load the tested engine caused a certain inconvenience, consisting in the fact that its operating range (rotational speed) was electronically limited from the bottom, which meant that the engine could only be loaded in the speed range from 1750 rpm to 3750 rpm and only in this
range it was possible to test ACC engines on the stand. On both stands power, torque, and fuel consumption of the investigated combustion engines were determined. For the case of Fiat 126p engine it was also measured indicated pressure and temperature of exhaust gases.

The simulation tests of the ACC engine operation were performed with the use of proprietary software implemented in the C++ environment. Detailed simulation model was described in [16]. The mathematical model was based on the standard physical zero-dimensional [4, 19, 24] engine description with an additional mechanical oscillator [12, 31, 59]. The simulation model comprised over 200 parameters to reflect engine geometry variables, physical gas constants and their mixtures occurring in the ACC engine. The description of the model presented in this article was limited to the field of Vibe’s [19] function application, modified by Bonatesta et al. [9]. The modified Vibe’s function for the ACC motor correctly defines the generated heat for up to 80% of the engine load [16].

5. Results and discussion

5.1. Rated parameters and engine exhaust gas temperature

The values of specific fuel consumption (efficiency) achieved during the tests for several selected engine speeds and the rated load are shown in Fig. 5. The results are selected from more favourable measurements and approximated to the assumed engine speeds. The line width was associated with an error of measurement of ±4%. Single points where exceptionally good values of specific fuel consumption were obtained were omitted due to the impossibility of repeating the parameters of the mixture composition due to the carburettor supply system.

The specific fuel consumption values are shown in three colors in Fig. 5. Dark green color indicates the results achieved in the air-cooled four-stroke ACC engine with the maximum compression ratio \( CR_{\text{max}} = 17.5:1 \) and the largest range of engine speeds from 750 to 3750 rpm. In the case of this graph, it is worth noting the minimum speed of the ACC engine.

The light green color was used for the engine with the maximum compression ratio \( CR_{\text{max}} = 18.5:1 \), while the blue color was for the engine with \( CR_{\text{max}} = 19.3:1 \), i.e., with the highest compression ratio used during the tests. Such a high value of the compression ratio was achieved by rebuilding the diesel engine from a power generator with a CR = 20.5:1. As mentioned earlier, the rotational speed was limited by the electronic system cooperating with the generator (power generator) to a value of 3650 rpm. Therefore, during the tests, the speed value of 3500 rpm was adopted as a safe value. Using the base 1.5L four cylinder DVVT engine supplied with gasoline and the same engine with dual shaft control variable compression ratio (DSC-VCR) Lin et al. [28] conducted comparative studies for the 3 cases: the maximum torque case with 1750 rpm (case 1), the maximum power case with 5500 rpm (case 2), and the high efficiency case with 2300 rpm and a 75% load (case 3). For the consecutive case numbers, the BSFC was equal to 260, 330 and 220 Nm, respectively. The DSC-VCR decreased BSFC by about 6.19%, 8.07%, and 8.67% in comparison to the base engine for each case. Under case 1 the BSFC was close or lower compared to the cases of ACC engines analyzed in the present study.

Another parameter which is important from the point of view of the ACC engine as a power generating device is its power to displacement ratio (power related to one liter of cubic capacity), as shown in Fig. 6. Obtained values of such a ratio did not exceed value of 25 kW/L. In this case, the ACC engine in the previous two-valve versions without the variable valve timing system and turbo boost should be compared with naturally aspirated engines up to approximately 3750 rpm. When this value is exceeded, the torque starts to drop significantly, making the engine less efficient due to the falling in cylinder filling.

For better illustration of the ACC engine ratings, Fig. 7 showed the full loaded engine torque to displacement ratio (engine torque related to one liter of displacement) in function of engine speed. Obtained values of such ratio did not exceed value of 82 Nm/L.

For the comparison during mentioned studies using base 1.5 L four-cylinder DVVT engine supplied with gasoline and the same engine with dual shaft control variable compression ratio (DSC-VCR) [28] the engine torque was equal to 210, 190 and 155 Nm, for the case 1, case 2, and case 3,
respectively. These values related to the consecutive values of the loaded engine torque to displacement ratio including 140, 126 and 103 Nm/L, respectively. For the case 1 such values were 1.4-1.9-fold higher compared to the cases of ACC engines analyzed in the present study. The DSC-VCR increased the torque by about 6.57%, 4.81%, and 6.19% in comparison to the base engine for each case.

5.2. Indicator charts

The selected four indicator plots shown in Fig. 9 to Fig. 12 include the results of research work on the ACC engine with CR$_{\text{max}}$ = 17.5:1. The choice was made due to two parameters: the load of at least 80% and the rotational speed, because the mutual relation of these parameters has a decisive influence on the course of the pressure indicated in the engine cylinder.

The diagram in Fig. 9 is an example of the indicator pressure course with the greatest influence of the ACC system, which is mainly due to the low engine speed and good dynamic properties of the ACC system, because the value of the indicator $\gamma_{\text{fmax}}$ used in the measurements was of about 80 cm$^2$/kg. The indicated pressure course was characterized by flattening responsible for energy recuperation by the ACC system.

Another parameter for assessing the operating ACC engine was the exhaust gas temperature measured close to the exhaust valve, i.e., at a distance not exceeding 100 mm from the valve surface in the cylinder, shown in Fig. 8. It was noticed that the maximum exhaust temperature measured at this point did not exceed 530°C at full engine load, while the temperature measured in the base engines was greater than 150°C to 350°C at the same load and rotational speed. For the comparison during studies on the movable cylinder block VCR and multi-fuel 660 cc engine with a maximum power of 3.7 kW supplied with gasoline, the exhaust gas temperature values vary in range 720–820°C for the CR = 7:1, in range 700–800°C for the CR = 8.5:1, and in range 660–770°C for the CR = 10:1, respectively. The highest values of the exhaust gas temperature were observed for the lowest brake power values for all CR values studied [44].

This flattening gradually faded away as the speed increased, as shown in the subsequent graphs of Fig. 10 and Fig. 11. For an engine speed above 4000 rpm, the recuperation effect disappeared due to the decreasing displacement of the additional piston in the ACC system and increasing delays in its return.

In Figure 11, the influence of the ACC system at a speed of 3750 rpm became barely visible. This rotational speed can be considered a limiting one, because after exceeding it, only the registered movement of the additional piston indicated that the ACC system was active.
equalization is an essential feature of the ACC engine and takes place whenever there is a reaction of the ACC system. The best results are achieved at low engine speeds and in such conditions the differences for the same dose of the air-fuel mixture are reduced to 1%.

6. Summary

The ACC engine allows avoiding detonation combustion without changing the amount and composition of the combustible mixture, and even without delaying the ignition advance angle. It allowed also to obtain lower fuel consumption and temperature of exhaust gas compared to the case of the combustion engines with fixed CR values. With enhancement of CR values for ACC engine it was observed a decrease of specific fuel consumption and temperature, end increase of volumetric torque and volumetric power indicators in whole range studied of engine speeds. In addition, the possible range of control of the combustion process allowed the ACC engine to operate with different types of hydrocarbon fuels, for example, in the form of petrol with various alcohol admixtures, as described in [53, 57]. The very intense flow of the combustible mixture inside the cylinder of the ACC engine allowed describing the combustion in the ACC engine with zero-dimensional mathematical models with the dual Vibe function providing the proper characterization of the heat release process [20]. The use of very high maximum compression ratios allows the ACC engine to operate to a certain extent as an HCCI engine with high lambda coefficients.

Nomenclature

| ACC | active combustion chamber |
| BDC | bottom dead center |
| CI | compression ignition |
| CR | compression ratio |
| HCCI | homogeneous charge compression ignition |
| APPS | additional piston phase shift |
| SI | spark ignition |
| TDC | top dead center |
| VCR | variable compression ratio |
| VVA | variable valve actuation |
| VVT | variable valve timing |

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