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Study on selected parameters of engine with the active combustion chamber

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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 CR_{max} *and the indicator* γ_{fm} *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.*

Key words: *combustion engine, active combustion chamber, variable compression ratio*

1. Introduction

Each internal combustion engine has its characteristic design parameter called the geometric compression ratio which is the ratio of the maximum volume of the combustion chamber at the bottom dead center (BDC) piston position and the minimum volume of the combustion chamber at the top dead center (TDC) piston position. In conventional engines such design parameter is constant, which is not optimal from the point of view of the heat release in the combustion process and emissions, especially under altering conditions of load and speed during city driving [26]. The most common combustion parameters responsible for the loss of efficiency in an SI engine are knock, load fluctuations, misfiring, cycle by cycle fluctuations, etc. [48].

The improvement of the ecological and energetic performances of the modern piston engines is possible using, among other tools, variable compression ratio playing as one of the engine operation regulators [51]. Variable compression ratio (VCR) is more often applied in compressed ignition (CI) engines [15, 18, 27, 30, 36, 37], to name a few, and less in spark ignition (SI) engines supplied with various fuels, to which group belonged the engine investigated in the presented study.

In most known configurations of VCR engines, inter alia presented further in the article, the CR is changed in a predetermined way by additional elements, often based on an existing algorithm, and therefore with some delay in response to the measured value of pressure in the combustion chamber. To the authors' knowledge, the engine with active combustion chamber (ACC) analyzed in the present study is the first solution in which a variable compression ratio is implemented in the conditions of a vibrating system consisting of an auxiliary piston and a spring cooperating with it. This enables a much faster adaptation of the combustion chamber volume to the current value of the pressure inside it.

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 NO_x emission enhanced with the increase of CR.

Qasemian et al. [40] studied effects of varied CR of biofueled SI engines on their thermal balance and waste heat recovery potential. They found that the heat loss to brake power ratio (Q_{ht}/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_{ex}/Q_{ht}) for all studied fuels. There existed a direct relationship between the ethanol in fuel content and Q_{ex}/Q_{ht} ratio. As the percentage of ethanol in fuel enhanced, the Q_{ex}/Q_{ht} 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 S.I. 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_{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 NO_x, 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 thermodynamically 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 octane 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 monohead (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.

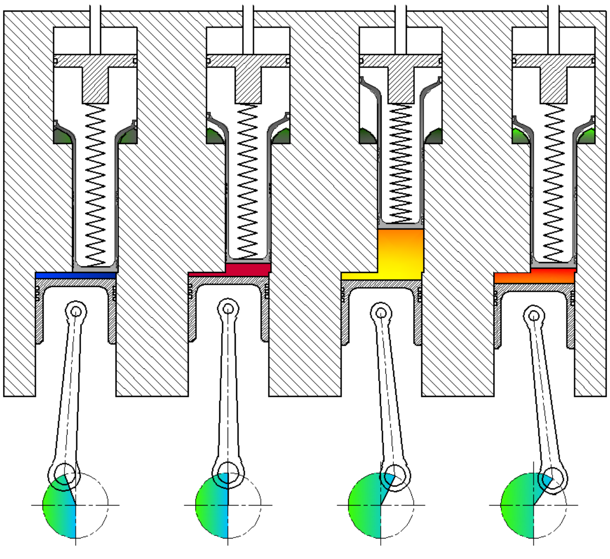

Fig. 1. The principle of variable expansion in the ACC engine

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 the compression stroke, changed 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.

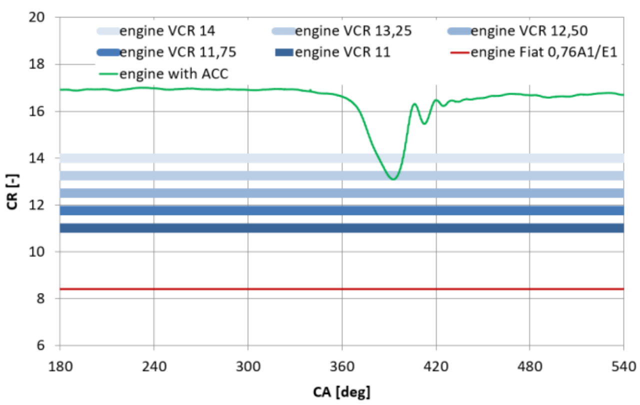

Fig. 2. CR changes in an engine with ACC within a portion of a single working cycle against the background of constant CR values in old and new SI engines and fixed CR values actuated by the control system in an engine with a VCR system

It can be seen from Fig. 2 that the concept of compression ratio regarding the ACC engine is strict, therefore the concept of the maximum compression ratio CR_{max} has been introduced. The maximum compression ratio is the compression ratio the ACC engine being achieved with the lowest position of the auxiliary piston in the ACC system (Fig. 1), expressed by equation (1):

$$CR_{max} = \frac{V_s + V_{C_{min}} + V_{ACC_{min}}}{V_{C_{min}} + V_{ACC_{min}}} \qquad (1)$$

where: V_s – displacement volume, V_{Cmin} – the smallest possible volume of the combustion chamber, V_{ACCmin} – volume of the combustion chamber at blocking pressure.

During the bench tests, to obtain the lowest position of the additional piston in the ACC system, the blocking pressure p1 (i.e. the pressure at which the ACC system did not operate) was used, the value of which was experimentally determined to be equal to 14 bar, but it is not a constant value, but rather a design value. Referring to Fig. 1, the maximum compression ratio is to be understood as the compression ratio corresponding to the relative position of the auxiliary piston in the position in Fig. 1a and the position of the main piston in the position of Fig. 1b. To maintain the safe operation of the ACC engine, the movement of the additional piston was limited by the elastic support. As a result, further increasing the blocking pressure would cause a further, but only up to hundredths of a millimeter, asymptotic movement of the auxiliary piston towards the main piston, resulting in only a negligible increase in the maximum compression ratio. Therefore, the use of the wording VCR used in the understanding of the previous designs of engines with a VCR system in relation to the ACC engine must be considered inadequate. The displacement of the additional piston in the ACC system during the compression stroke is not constant not only under the conditions of a fixed working cycle, but even within a segment of such a cycle characterized by slight changes in pressure in the cylinder, because it has an oscillating nature with an amplitude that automatically adapts to the current pressure conditions in the chamber combustion.

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 γ_{fm}. When introducing the indicator γ_{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 γ_{fm} was defined by the ratio of the additional area in the combustion chamber expressed in cm^2 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 γ_{fm} was described by equation (2).

$$\gamma_{fm} = \frac{\pi d_1^2}{4m} \sim 0.7854 \frac{d_1^2}{m} \qquad (2)$$

where: γ_{fm} – the indicator determining the correct operation of the ACC engine [cm^2/kg], d_1 – diameter of the additional piston in the combustion chamber [cm], m – mass of the additional piston [kg].

The diameter d_1 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 γ_{fm} while ensuring good pneumatic support properties.

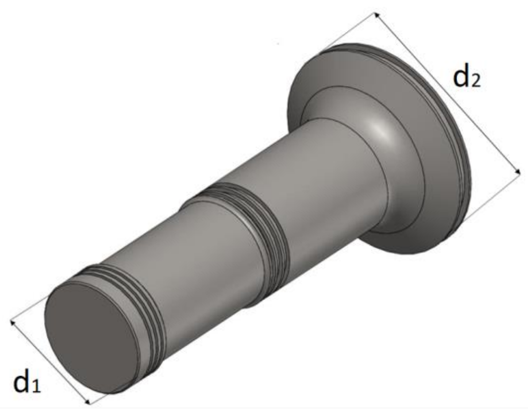

Fig. 3. Additional piston ACC system with its characteristic diameters d_1 and d_2

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 γ_{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 γ_{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 γ_{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 γ_{fm} = 88 cm^2/kg has been exceeded, thus the first positive results, for a CR_{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 γ_{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].

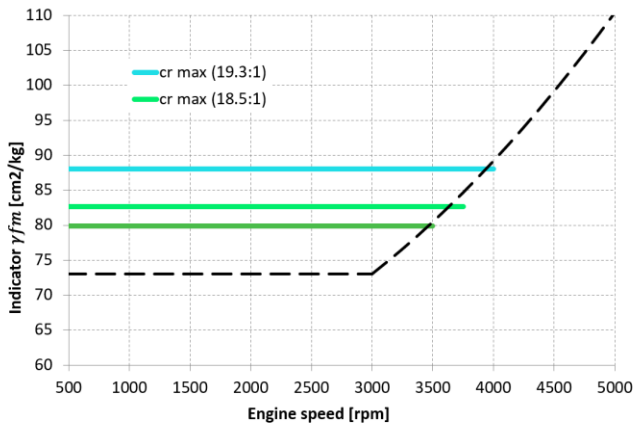

Fig. 4. The effect of the indicator γ_{fm} on the achieved maximum ACC engine speed

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 CR_{max} values can also be obtained. Unfortunately, the value of the main design parameter should not be associated directly with CR_{max} value, because there is always such a value of CR_{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 γ_{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 CR_{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 CR_{max} = 18.5:1 and CR_{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_{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_{max} = 18.5:1, while the blue color was for the engine with CR_{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.

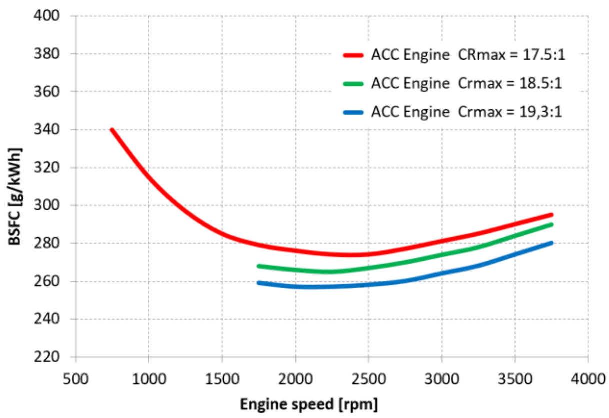

Fig. 5. The specific fuel consumption (BSFC) as a function of rotational speed for three CR_{max} values in the ACC engine

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 exceeded 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.

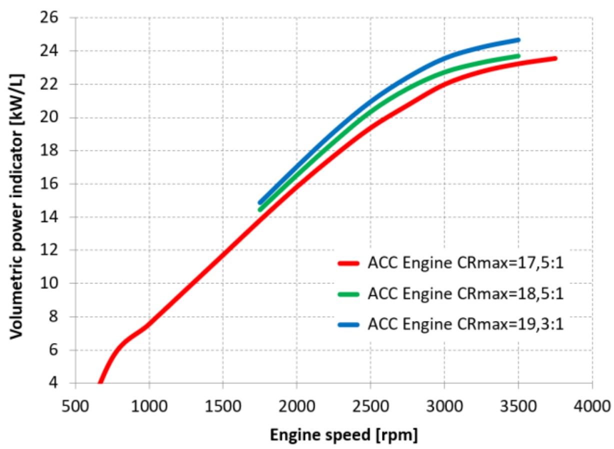

Fig. 6. Power for the maximum torque of the ACC engine with a displacement of one liter obtained for three CR_{max} values

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 exceeded 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.

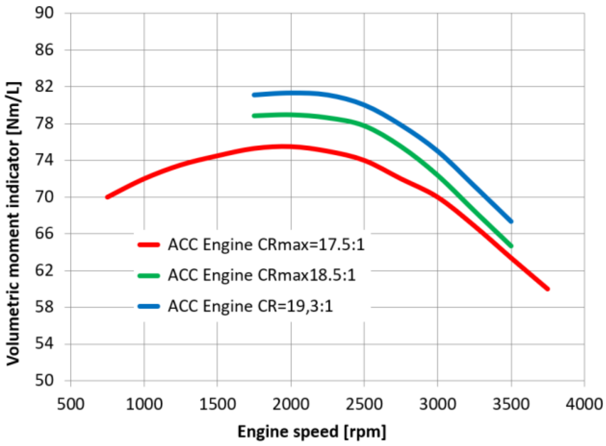

Fig. 7. Full load torque for the ACC engine with a displacement of one liter obtained at different maximum compression ratios

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].

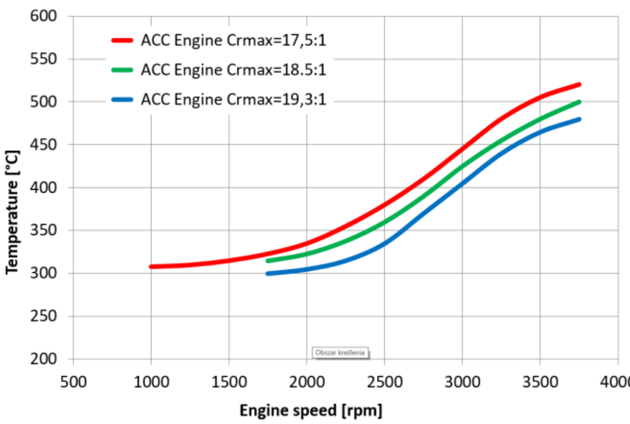

Fig. 8. Exhaust gas temperature measured in the ACC engine as a function of engine speed for maximum engine torque and various compression ratios

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_{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 γ_{fm} 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.

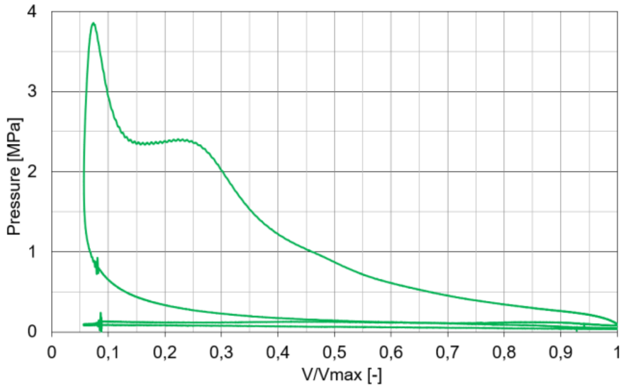

Fig. 9. Indicated pressure in the ACC engine under CR_{max} = 17.5: 1 at engine speed of 1000 rpm

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.

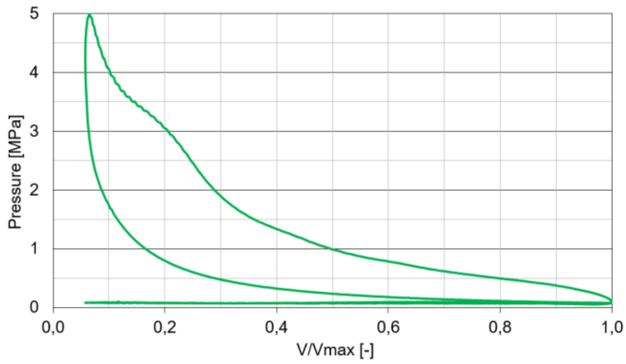

Fig. 10. Indicated pressure in the ACC engine under CR_{max} = 17.5: 1 at engine speed of 1900 rpm

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.

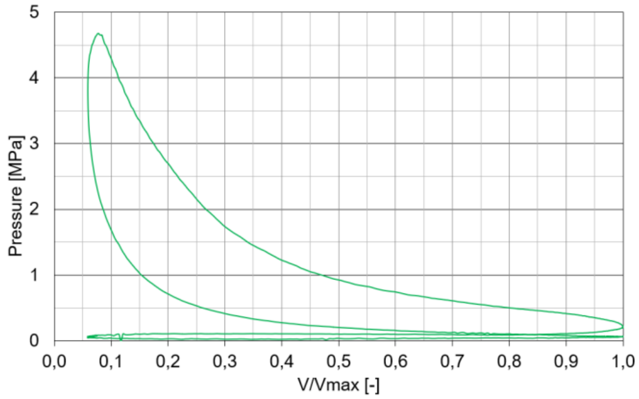

Fig. 11. Indicated pressure in the ACC engine under CR_{max} = 17.5: 1 at engine speed of 3750 rpm

Figure 12 shows the influence of the ACC system on the smooth running of the engine – pressure amplitude. It shows the indicated pressure graph (green) and the corresponding phase shift of the additional piston (APPS) in the ACC system in relation to the engine piston (deep pink). The diagrams clearly correlate with each other: the higher amplitude of the indicated pressure always corresponds to the higher amplitude of the lift of the additional piston in the ACC system. Of course, the degree of self-regulation carried out in a single cycle by the ACC system depends primarily on several parameters, which include: the main design parameter, engine speed, and pressures p_1 acting on area $(\pi d_1^2)/4$ of the upper surface of additional piston and p_2 acting on area $(\pi d_1^2 - d_2^2)/4$ of its bottom surface being outside the area affected by gas pressure in the engine cylinder [16].

The ACC system, operating in accordance with the diagram in Fig. 1, increases the volume of the combustion chamber mainly on the expansion stroke. The magnitude of this change depends on the pressure amplitude in the cylinder. Thus, higher pressure is accompanied by a greater change in volume caused by the ACC system, and the final consequence is a reduction in the very amplitude of the pressure in the cylinder during the combustion process. At this point it should be noted that this specific amplitude 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%.

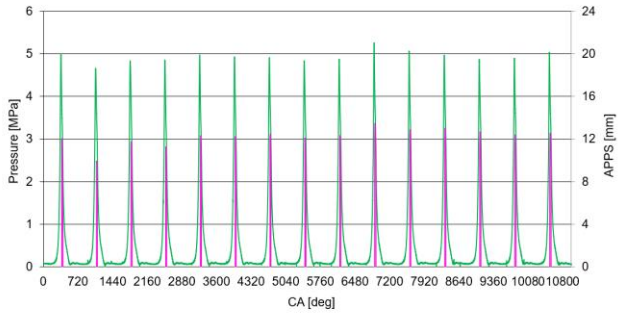

Fig. 12. Influence of the ACC system on the uniformity of indicated pressure amplitudes

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 igniton
TDC top dead center
VCR variable compression ratio
VVA variable valve actuation
VVT variable valve timing

Bibliography

[1] Abdel-Rahman AA, Osman MM. Experimental investigation of varying the compression ratio of SI engine under different ethanol–gasoline fuel blends. Int J Energy Res. 1997;21(1): 31-40. https://doi.org/10.1002/(SICI)1099-114X(199701)21:1%3C31::AID-ER235%3E3.0.CO;2-5

[2] Aina T, Folayan CO, Pam GY. Influence of compression ratio on the performance characteristics of SI engine. Adv Appl Sci Res. 2012;3(4):1915-1922.

[3] Al-Baghdadi S, Maher AR. Effect of compression ratio, equivalence ratio and engine speed on the performance and emission characteristics of SI engine using hydrogen as a fuel. Renew Energy. 2004;29(15):2245-2260. https://doi.org/10.1016/j.renene.2004.04.002

[4] Aldikas AC. Heat transfer characteristics of a spark-ignition engine. J Heat Transfer. 1980;102(2):189-193. https://doi.org/10.1115/1.3244258

[5] Babagiray M, Kocakulak T, Ardebili SMS, Calam A, Solmaz H. Optimization of operating conditions in a homogeneous charge compression ignition engine with variable compression ratio. Int J Environ Sci Te. 2022;9:1-22. https://doi.org/10.1007/s13762-022-04499-9

[6] Bas O, Akar MA, Seri H, Özcanli M. Performance of an SI engine with different spark plugs, VCR and H2 addition. Sigma J Eng Nat Sci. 2020;38(4):1951-1961. https://sigma.yildiz.edu.tr/article/149

[7] Basu D, Phulli S, Kotebavi V. Performance analysis of a VCR SI engine using petrol alcohol blends. 2014 Power and Energy Systems: Towards Sustainable Energy. 2014:1-5. https://doi.org/10.1109/PESTSE.2014.6805264

[8] Beroff J. Internal combustion engine with variable capacity and compression ratio. European Patent EP20010400802. 2005.

[9] Bonatesta F, Waters B, Shayler PJ. Burn angles and form factors for wiebe function fits to mass fraction burned curves of a spark ignition engine with variable valve timing. Int J Engine Res. 2010;11(2):177-186. https://doi.org/10.1243/14680874JER05009

[10] Brevick J. Design and development of a pressure reactive piston (PRP) to achieve variable compression ratio. Department of Energy Contract FC02-99EE50576, Final Technical Report 2000:44-49.

[11] Brevick JE. Variable compression ratio piston. US Patent 5755192A. 1998.

[12] Brzeski P, Pavlovskaia E, Kapitaniak T, Perlikowski P. The application of inerter in turned mass absorber. Int J Nonlinear Mech. 2015;70:20-29. https://doi.org/10.1016/j.ijnonlinmec.2014.10.013

[13] Caris DF, Nelson EE. A new look at high compression engines. SAE Technical Paper 590015. 1959. https://doi.org/10.4271/590015

[14] Clarke JR, Tabaczynsky RJ. Internal combustion engine with adjustable compression ratio and knock control. US Patent 6135086A. 2000.

[15] Dasore A, Rajak U, Panchal M, Reddy VN, Verma TN, Chaurasiya PK. Prediction of overall characteristics of a dual fuel CI engine working on low-density ethanol and diesel blends at varying compression ratios. Arab J Sci Eng. 2022; 2:1-8. https://doi.org/10.1007/s13369-022-06625-8

[16] Dabrowski A, Glogowski M, Kubiak P. Improving the efficiency of four-stroke engine with use of the pneumatic energy accumulator-simulations and examination. Int J Automot Techn. 2016;17:581-590. https://doi.org/10.1007/s12239-016-0058-1

[17] Ferrey P, Miehe Y, Constensou C, Collee V. Potential of a variable compression ratio gasoline SI Engine with very high expansion ratio and variable valve actuation. SAE Int J Engines. 2014;7(1):468-487. https://doi.org/10.4271/2014-01-1201

[18] Ganji PR, Vysyaraju RKR, Surapaneni SR, Kumar BK. Enhancement of combustion characteristics of VCR diesel engine by optimizing engine parameters. SN Appl Sci. 2021;3(8). https://doi.org/10.1007/s42452-021-04739-6

[19] Ghojel JI. Review of the development and applications of the Wiebe function: A tribute to the contribution of Ivan Wiebe to engine research. Int J Engine Res. 2010;11(4):297-312. http://dx.doi.org/10.1243/14680874JER06510

[20] Glogowski M, Kubiak P, Šarić Ž, Barta D. New theoretical cycle for active combustion chamber engine. The Archives of Automotive Engineering. 2019;83(1):23-42. https://doi.org/10.14669/AM.VOL83.ART2

[21] Glogowski M, Kubiak P, Szufa S, Piersa P, Krukowski M. The use of the Fourier series to analyze the shaping of thermodynamic processes in heat engines. Energies. 2021; 14(8):2316. https://doi.org/10.3390/en14082316

[22] Gooijer LHD. A reciprocating piston mechanism. European Patent EP2620614 A1. 2013.

[23] Hammermueller B, Orlowsky K, Hanciogullari D, Pischinger S. Experimental analysis of the system behavior of a two-stage variable compression ratio (VCR) system in fired SI-engine operation. Int J Engine Res. 2022;23(10):1619-1633. https://doi.org/10.1177/14680874211032382

[24] Heywood JB. Internal combustion engine fundamentals, 2nd ed. New York: McGraw-Hill; 2018.

[25] Hiyoshi R, Aoyama S, Takemura S, Ushijima K, Sugiyama T. A study of a multiple-link variable compression ratio system for improving engine performance. SAE Technical Paper 2006-01-0616. 2006. https://doi.org/10.4271/2006-01-0616

[26] Joshi MP. Variable compression ratio (VCR) engine – a review of future power plant for automobile. Int J Mech Eng Res Develop. 2012;2(1):9-16.

[27] Kumar C, Rana KB, Tripathi B. Effect of diesel-methanol-nitromethane blends combustion on VCR stationary CI engine performance and exhaust emissions. Envir Sci Pollut R. 2019; 26(7):6517-6531. https://doi.org/10.1007/s11356-018-04058-1

[28] Lin J, Yang S. A predictive study of a new VCR engine with high expansion ratio and high-efficiency potential under heavy load conditions. Energies. 2020;13:1655. https://doi.org/10.3390/en13071655

[29] López JJ, García A, Monsalve-Serrano J, Cogo V, Wittek K. Potential of a two-stage variable compression ratio downsized spark ignition engine for passenger cars under different driving conditions. Energ Convers Manage. 2020;203:112251. https://doi.org/10.1016/j.enconman.2019.112251

[30] Manimaran R, Mohanraj T, Venkatesan M. ANN modeling for forecasting of VCR engine performance and emission parameters fueled with green diesel extracted from waste biomass resources. Envir Sci Pollut R. 2022;29(34):51183-51210. https://doi.org/10.1007/s11356-022-19500-8

[32] Meirovitch L. Fundamentals of Vibrations; McGraw-Hill. New York, USA, 2001.

[33] Mendler C. High-efficiency VCR engine with variable valve actuation and new super-charging technology. Final Report of the NETL Contract Number DE-EE0005981. 2018. https://doi.org/10.2172/1545742

[34] Meyer M, Frank VDS. Internal combustion reciprocating piston engine. European Patent EP1811151 A2. 2007.

[35] Moteki K, Aoyama S, Ushijima K. A study of a variable compression ratio system with a multi-link mechanism. SAE Technical Paper 2003-01-0921. 2003. https://doi.org/10.4271/2003-01-0921

[36] Moteki K, Fujimoto H, Aoyama S. Variable compression ratio mechanism of reciprocating internal combustion engine. US Patent 6505582 B2 2003.

[37] Murugapoopathi S, Ramachandran T, Rajaganapathy C. Theoretical performance on energy and exergy analysis of methyl esters of rubber seed oil fueled on supercharged VCR engine. Environ Dev Sustain. 2022;8:1-20. https://doi.org/10.1007/s10668-022-02642-7

[38] Nayyar A, Sharma D, Soni S, Mathur A. Experimental investigation of performance and emissions of a VCR diesel engine fueled with n-butanol diesel blends under varying engine parameters. Environ Dev Sustain. 2017;24(25):20315-20329. https://doi.org/10.1007/s11356-017-9599-8

[39] Nuthan PBS, Pandey JK, Kumar GN. Impact of changing compression ratio on engine characteristics of an SI engine

fueled with equivolume blend of methanol and gasoline. Energy. 2020;191:116605.
https://doi.org/10.1016/j.energy.2019.116605

[40] Ozcan V, Yamin JAA. Performance and emission characteristics of LPG powered four stroke SI engine under variable stroke length and compression ratio. Energ Convers Manage. 2008;49:1193-1201.
https://doi.org/10.1016/j.enconman.2007.09.004

[41] Qasemian A, Haghparast SJ, Azarikhah P, Babaie M. Effects of compression ratio of bio-fueled SI Engines on the thermal balance and waste heat recovery potential. Sustainability. 2021;13(11):5921. https://doi.org/10.3390/su13115921

[42] Rakopoulos DC, Rakopoulos CD, Giakoumis EG, Kosmadakis GM. Numerical and experimental study by quasi-dimensional modeling of combustion and emissions in variable compression ratio high-speed spark-ignition engine. J Energ Eng. 2021;147(5):1-22.
https://doi.org/10.1061/(ASCE)EY.1943-7897.0000780

[43] Rapan G, Rapan I. Variable compression ratio engine. Romanian Patent RO115662B. 2000.

[44] Riley MB. Internal Combustion Engine with Variable Combustion Chambers and Increased Expansion Cycle, US Patent 5341771 A. 1994.

[45] Rinu T, Sreesankaran M, Jeevan J, Dileep MP, Manjunath P. Experimental evaluation of the effect of compression ratio on performance and emission of SI engine fueled with gasoline and n-butanol blend at different loads. Perspectives in Science. 2016;8:743-746.
https://doi.org/10.1016/j.pisc.2016.06.076

[46] Roberts M. Benefits and challenges of variable compression ratio (VCR). SAE Technical Paper 2003-01-0398. 2003.
https://doi.org/10.4271/2003-01-0398

[47] Roy A, Mishra C, Jain S, Solanki, N. A new design to achieve variable compression ratio in a spark ignition engine. Innovative Design, Analysis and Development Practices in Aerospace and Automotive Engineering (I-DAD 2018) 2018;2: 388-396. https://doi.org/10.1007/978-981-13-2718-6_37

[48] Shaik A, Moorthi NSV, Rudramoorthy R. Variable compression ratio engine: a future power plant for automobiles – an overview. P I Mech Eng D-J Aut. 2007;221(D9):1159-1168.
https://doi.org/10.1243/09544070JAUTO573

[49] Shetty S, Jose A, Shrinivasa RBR. Comparison of performance and combustion analysis of a VCR SI engine using audio transducers. Mater Today-Proc. 2021;46(7):2798-2806.
https://doi.org/10.1016/j.matpr.2021.02.603

[50] Rao SG, Paul PJ, Mukunda HS. Biomass derived producer gas as a reciprocating engine fuel – an experimental analysis. Biomass Bioenerg. 2001;21:52-72.
https://doi.org/10.1016/S0961-9534(01)00014-9

[51] Takahashi N, Aoyama S, Moteki K, Hiyoshi R. A study concerning the noise and vibration characteristics of an engine with multiple-link variable compression ratio mechanism. SAE Technical Paper 2005-01-1134. 2005.
https://doi.org/10.4271/2005-01-1134

[52] Takalkar M, Khan I. Study of variable compression ratio engine (VCR) and different innovations to achieve VCR. International Journal for Research in Applied Science and Engineering Technology. 2017;5(11):1473-1478.
https://doi.org/10.22214/ijraset.2017.11213

[53] Tomar SK, Mishra R, Bisht S, Kumar S, Saxena G. Design of variable length connecting rod for VCR engine. International Journal of Engineering Research and Applications (IJERA) 2014, Special Issue: International Conference on Emerging Trends in Mechanical and Electrical Engineering (ICETMEE) 13th-14th March 2014, 280-285.

[54] Topgül T, Yücesu H, Çinar C, Koca A. The effects of ethanol-unleaded gasoline blends and ignition timing on engine performance and exhaust emissions. Renew Energ. 2006; 31(15):2534-2542.
https://doi.org/10.1016/j.renene.2006.01.004

[55] Wittek K, Geiger F, Andert J, Martins M, Cogo V, Lanzanova T. Experimental investigation of a variable compression ratio system applied to a gasoline passenger car engine. Energ Convers Manage. 2019;183:753-763.
https://doi.org/10.1016/j.enconman.2019.01.037

[56] Wong VW, Stewart M, Lundholm G, Hoglund A. Increased power density via variable compression/displacement and turbocharging using the al-var-cycle engine. SAE Technical Paper 1998-981027. 1998. https://doi.org/10.4271/981027

[57] Wos P, Kuszewski, H, Ustrzycki A. Exhaust emission features of variable compression ratio (VCR) diesel engine. Journal of KONES Powertrain and Transport. 2010;17(3):3-6.

[58] Yücesu SH, Topgül T, Çinar C, Okur, M. Effect of ethanol–gasoline blends on engine performance and exhaust emissions in different compression ratios. Appl Therm Eng. 2006;26(17-18):2272-2278.
https://doi.org/10.1016/j.applthermaleng.2006.03.006

[59] Yuh M, Tohru G. The effect of higher compression ratio in two-stroke engines. Yamaha Motor Co Ltd. SAE 2008-931512:355-362.

[60] Zhan YL, Wan BY, Wang XZ, Hu YH. A simulation model for the main engine of the modern container ship. Machine Learning and Cybernetics. Proc. IEEE 2004 Int. Conf. 2004; 5:2996-3002. https://doi.org/10.1109/ICMLC.2004.1378546

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