HIGH COMPRESSION SPARK IGNITION ENGINE WITH VARIABLE COMPRESSION RATIO USING ACTIVE COMBUSTION CHAMBER

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Abstract

In this article Authors present the continuation of the calculations for theoretical ACC engine cycle, considering additionally "VCR function" – changeable compression level. For this purpose the self-acting volume change, realized by ACC system, was used. The ACC system was adjusted appropriately to control the compression level. The analysis is based on three cases, representing delayed, premature and optimal reaction of ACC system. Reactions are presented in form of plots with indicated pressure in the combustion chamber.

As the result of the conducted analysis and interpretation of obtained graphs, the calculation approach of compression ratio for ACC presented in previous article is being challenged. For the optimal reaction of ACC system, the theoretical operation schematics are devised and presented in the key points of the work. Based on the schematics, the values of theoretical efficiency were calculated for different cycles of theoretical ACC engine, in which regulation of compression ratio takes place.

Moreover, the presented analysis includes graphs with optimal courses of indicated pressure for significantly different work parameters of ACC engine, showing its regulation possibilities. Also the time scaled graphs (with millisecond as basic time unit) are presented to show the possibilities of dynamic ACC systems, which are comparable with the combustion time (from 3 to 0,5 ms). In this paper the general discussion is started about the compression ratio in more complex kinematic systems including ACC.

Keywords: internal combustion engine, theoretical cycle, variable compression ratio, additional volume, active combustion chamber

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Nomenclature

$\alpha = \frac{V_5}{V_3}$ – loading ratio

$\beta = \frac{p_3}{p_2} = \frac{T_3}{T_2} = \varepsilon^{(1-\kappa)} \frac{T_3}{T_1}$ – pressure ratio

$\gamma = \frac{Q_{Otto}}{Q_{max}}$ – Otto power ratio

$\varepsilon = \frac{V_1}{V_2}$ – compression ratio

$\kappa = \frac{c_p}{c_v}$ – specific heat ratio

$\eta_1 = 1 - \frac{1}{\varepsilon^{(\kappa-1)}}$ – Otto cycle efficiency

η_2 – proposed cycle efficiency

1. Introduction

Up till now, in described theoretical cycles of internal combustion engines [19,23,28] there was no connection between the change of compression ratio and the way of energy delivering [1,27]. Reason being that there was no technical solution to achieve it [4,11,20], but the situation has changed when the ACC system was introduced [14]. During the first trials in 2005, the engine was showing some issues when it comes to tuning the ignition, which was calculated using the Otto cycle [22]. This inconsistency was confirmed experimentally once lasers of higher frequency (2 and 20 kHz) were introduced into the setup. These high frequency lasers were used to precisely determine the position of additional piston in ACC. The pressure course in the cumulative volume shown in the measurements, substantially differs from pressure course in a typical spark ignition engine. Immediately after publishing the results, a discussion concerning calculation of compression ratio and efficiency of ACC engine was started. To elaborate this issue, Authors are presenting an in-depth process of calculating the theoretical cycle for the ACC engine with VCR function [13]. Authors would also like to initiate a discussion between researchers working with internal combustion engines.

ACC engine with a high compression ratio (30:1), as any other spark ignition engine, needs an adjustable compression ratio. This will prevent it from detonative combustion (Fig. 2) having negative influence on the engine [12,21,26]. The work of four-stroke internal combustion engine, equipped with a compression ratio adjustment is possible due to many technical solutions [16,24]. These usually differ in the mean and place of regulation [3,6]. Nevertheless, one can distinguish several basic methods for changing the displacement or combustion chamber volume, which in other words is the regulation of compression ratio. The list below encloses the typical methods for compression ratio adjustment:

- Crankshaft displacement
- Engine block and head displacement
- Head displacement
- Additional, independent element in head displacement
- Additional, independent element in piston displacement
- Other not having a standard crank-piston system

Therefore, the variable compression ratio [17,18,25] should be applied to additionally increase the efficiency of engine at partial loads. ACC engine is built in such manner, which enables the compression variability and it can be realized together with a change of energy supply in the cycle.

All the features of ACC engine enable regulation possibilities unattainable for a conventional internal combustion engine. Although, before these are described in-depth, a short principle of operation will be presented based on scheme in Fig 1.

The principle of operation for the engine with active combustion chamber, hereinafter referred to as ACC system, is as follows: when the pressure in the engine cylinder overcomes the initial strain in the pneumatic spring (3), the additional piston (2)[5,7,10], which is a barrier in the active combustion chamber ascends, at the same time storing the energy in the spring (3). When the piston (2) reaches the maximum position, accumulated energy in the spring (3) attains its highest value. In this moment all forces acting on the piston (3), including inertia forces, are in equilibrium. From this moment on–the descent of the piston, recovery of energy accumulated in the spring (3) takes place. This energy can be then used to sustain the pressure and volume in combustion chamber, e.g. on the constant level. The control of the piston's (2) lower docking position is provided by a one-way pneumatic spring (5) and a one-way damper (6). The adaptation abilities of the ACC system are due to the change of the pneumatic spring (3) rigidity: by the change of the spring volume caused by movable partition (4) or by the change of the spring's supply pressure.

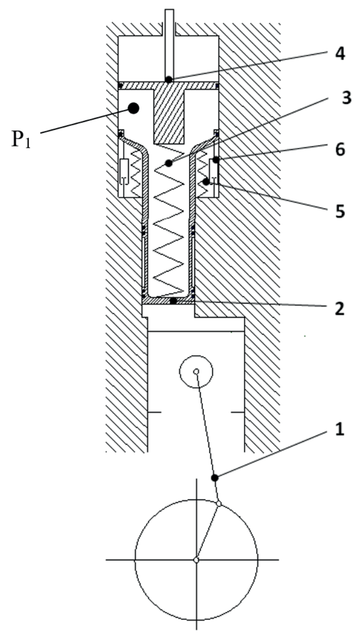

Fig. 1. Schematic diagram of ACC engine operation

VCR function for the ACC engine is a simple concept, since it only requires a faster reaction from ACC system, what is presented on Fig. 6, Fig. 7, Fig. 8 and Fig. 9. The adjustment of compression ratio is its immanent feature. This is beneficial for the ACC system, since the higher initial velocity of the additional piston in this system in the initial phase of combustion, decreases its reaction time during combustion. Direct consequence of a more energetic reaction is the possibility of further increase of the compression ratio up to the values which are not attainable in most of the engines with VCR system. Authors conclude that the maximum attainable compression ratio in the ACC engine with VCR is equal to 30:1 and therefore, in the calculations for theoretical cycle this value is used. This is also the reason why it is crucial to determine experimentally the real compression ratio for the ACC engines. In the research, for two stroke ACC engine, the value of 22:1 was achieved and it was not causing any technical problems. Whereas in the four stroke, whose results are presented in this article, this value was limited to 17,5:1 because of construction restrictions.

2. ACC engine efficiency

Similarly as in previous paper [15], before starting the theoretical analysis, Authors want to demonstrate the real graphs (Fig. 2, Fig. 3 and Fig. 4) with open indicated pressure loop and the movement of additional piston in the ACC system and the closed graphs of indicated pressure (Fig. 6, Fig. 7 and Fig. 8). These are scaled down by volume of the crank-piston system and cumulative volume, which is the sum of volumes of the crank-piston system and ACC system. Additionally, the graph on Fig. 5 is shown with the course of theoretical efficiency in the Otto cycle as a function of compression ratio. This is a standard approach to specify the VCR engine efficiency, however is not correct for the ACC engine. The presented graphs demonstrate the validity of conducted theoretical analysis further in the article. Special emphasis is put on description of compression ratio. The order of graphs is chosen in sequence to demonstrate the purposefulness of using an ACC system. Fig. 2 presents the delayed reaction of the system, what results in characteristic oscillations in the course of indicated pressure, clearly indicating the detonative combustion. Oscillations are present in spite of considerable displacement of the additional piston equal to 13 mm. If the compression ratio will be calculated for this system in traditional, typical way it would be equal to 10,4:1, what will be indicating, that the detonative combustion should not take place, especially with partial loading of ~70%. Of course in this place the discussion about the shape of the combustion chamber and other aspects can be started, but a question appears: how to explain the graph in Fig. 3, which presents a premature reaction of ACC system. The indicated efficiency if far to significant for such low compression ratio, calculated at 6,1:1.

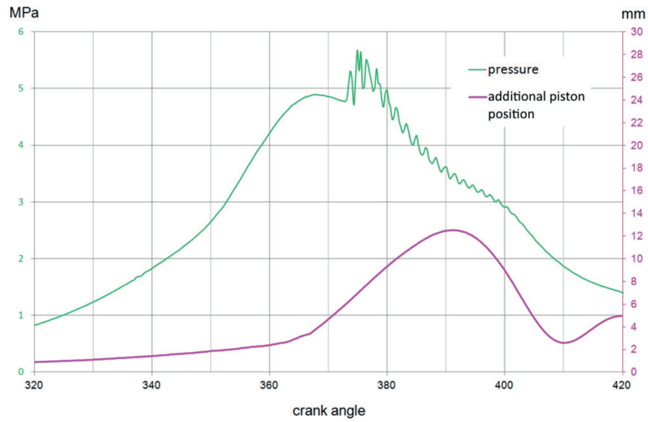

Fig. 2. Indicated pressure and additional piston movement for delayed reaction of ACC system

Moreover, the assessment of the efficiency is hindered by the fact that the maximum value of pressure is observed in the top dead center. The additional piston displacement was obtained in an indirect way by measuring the pressure p_1 in the pneumatic spring and despite the occurrence of slight oscillations due to wave phenomena, the measurement is valid. To prove the approach is proper, a static pressure calibration was conducted, in which a value of 43,5 mm at the same pressure level was measured.

The effect of advancing response of the ACC system, which is the change of the compression ratio, can be obtained by means of two approaches: change of the ignition advance angle or change of the ACC system parameters. The case from Fig. 3 was achieved by applying both of these approaches at the same time, namely changing the ignition advance angle from the optimal value equal to 20°ca at 2650 rpm, to 38°ca and changing the value of pressure p_1 from the value of 11 bar to 9 bar. It should be emphasized that all standard engine adjustment were not acting properly for the ACC engine and that was why it was mostly optimized during the trials. Amongst the most important factors is the ignition advance, which for a standard spark ignition engine seem unusual. However, due to its special build, the ACC is less sensitive to changing the advance angle. This is one of the reasons why on the graph characteristic oscillations – effects of the detonative combustion – are not visible, and which for sure will be present in the typical spark ignition engine. Also we can see typical negative loop on the closed graph (Fig. 7) for this values of ignition advance.

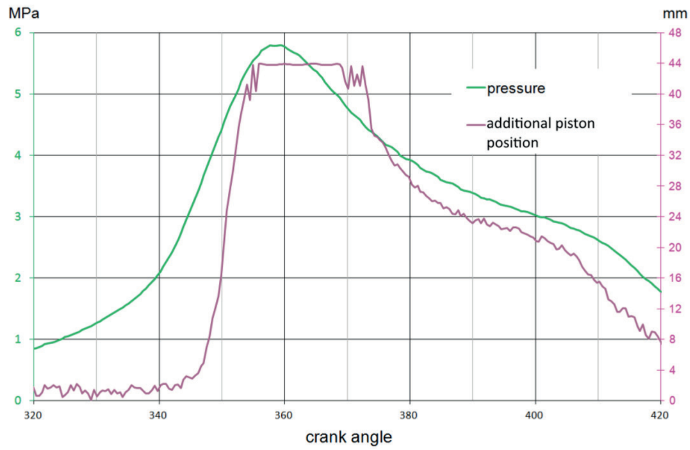

Fig. 3. Indicated pressure and displacement of the additional piston for an premature ACC system reaction

Naturally, at this angle, measurements have shown a slight drop of torque (~6%). This was a result of maximal pressure concentration, but the drop is significantly lower, than could be expected from compression ratio change.

Fig. 4 presents the case in which the reaction of the ACC system is optimal – in this measurement the biggest torque was observed using all possible adjustment parameters. Moreover, no oscillation due to detonative combustion is present.

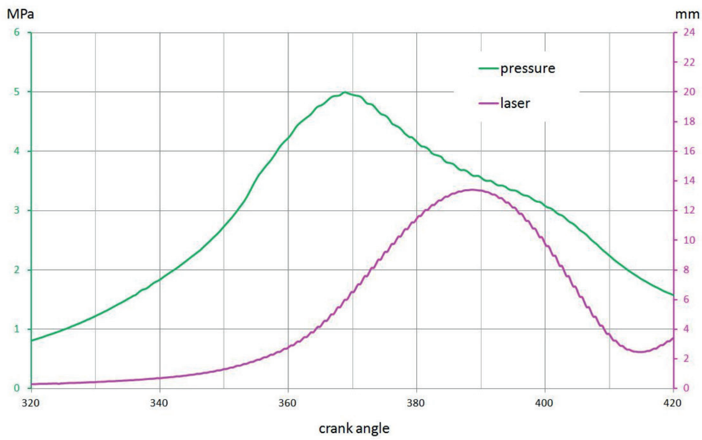

Fig. 4. Optimal pressure course and displacement of additional piston in ACC engine with VCR function

Fig. 4 is an example of optimal adjustment, so optimal choice of parameters is as follows: loading (~60%), pressure and volume in the ACC system and ignition overtaking angle (~23°ca), which results on only 3 mm difference between minimal position corresponding to the compression ratio of 17,5:1 in top dead center and the position from the graph. Such small difference at this loading was enough to control the combustion process, what resulted in high indicated efficiency.

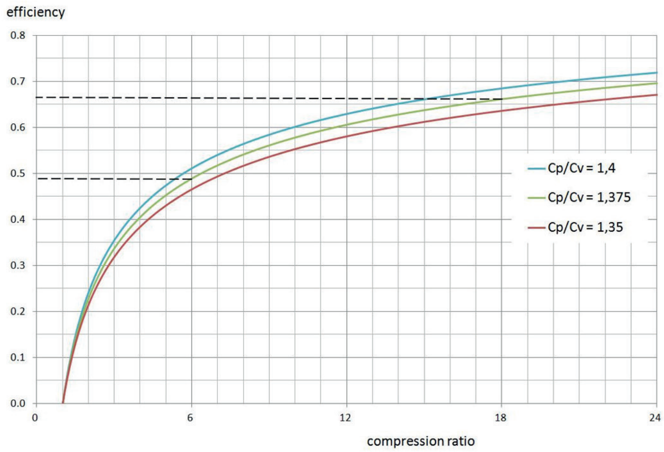

Fig. 5. Otto cycle efficiency as a function of compression ratio

At this stage, using compression ratio graph produced in the standard approach, and graph from Fig. 5, theoretical values of efficiency were found. The ranges, for the values c_p/c_v=1,4 ÷1,35 are presented in Table 1.

Table 1. The change of theoretical efficiencies calculated by means of standard approach

Figure	Compression Ratio	Otto Efficiency (%)	Difference $\eta_{\varepsilon max} - \eta_{\varepsilon 1\text{-}3}$ (%)	Comparison $\eta_{\varepsilon 1\text{-}3} / \eta_{\varepsilon max}$ (%)
Fig. 2	10,1 : 1	60,31 - 55,49	7,83 – 7.79	88,52 – 87,69
Fig. 3	6,1 : 1	51,49 - 46,90	16,69 – 16,38	75.52 - 74,11
Fig. 4	14,3 : 1	65,50 - 60,59	2,69 – 2,68	96,07 – 95,75

As demonstrated in Table 1, the biggest difference between maximum efficiency for theoretical Otto cycle and compression ratio 17,5:1 is equal to ~16,5%. This is for the change of compression ratio calculated in standard approach, so its apparent value drops to 6,1:1. While the decrease of efficiency, calculated as quotient of efficiencies, is equal to ~25%.

Efficiency of ACC engine calculated during the investigations for graph in Fig. 3, was equal to ~32%, which is not achievable experimentally, especially for well cooled engines with compression ratio of 6,1:1. If the fact that the additional piston from ACC system reaches maximal position after top dead center will be included into considerations, adjusted value of compression ratio calculated in typical way will decrease even more, reaching minimal value of 5,7:1. Differences presented in the table would increase further and would not be possible to validate.

Table 1 clearly shows that compression ratio calculated in standard approach is not competible with trial's results. While complementing the analysis with empirical values, a conclusion can be made that directly calculated compression ratio cannot be correct.

This raises a question of what is the proper compression ratio in ACC engine when power and compression strokes are overlapping.

Determining the compression ratio by the means of proposed approach, can be done by reading the values directly from the graphs, which in this case are pressure volume diagrams scaled by the volume of crank-piston system and cumulative volume for both systems. For this purpose, the smallest cumulative volume was used, which is reached by ACC engine before top dead center. Values are read from graphs in Fig. 6, Fig. 7 and Fig. 8.

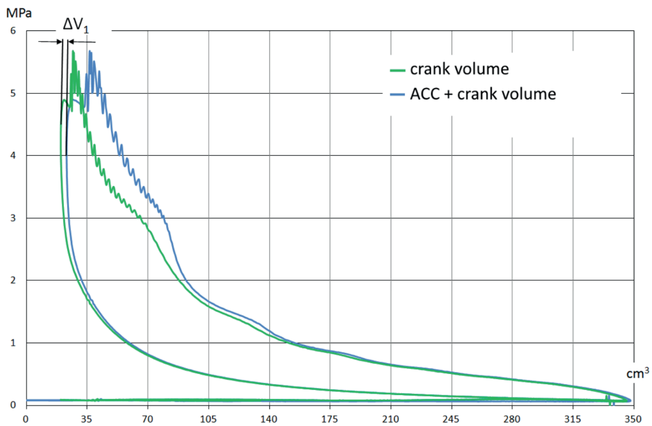

Fig 6. Pressure volume diagram of the crank-piston system volume and cumulative volume (delayed reaction of ACC system, 70% loading)

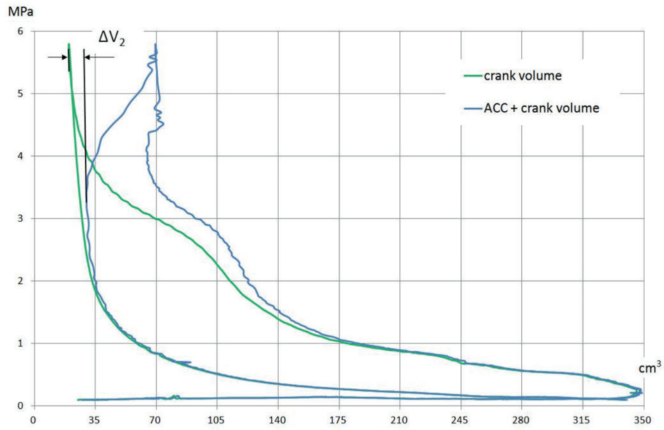

Fig. 7. Pressure volume diagram of the crank-piston system volume and cumulative volume (premature reaction of ACC system, 80% loading)

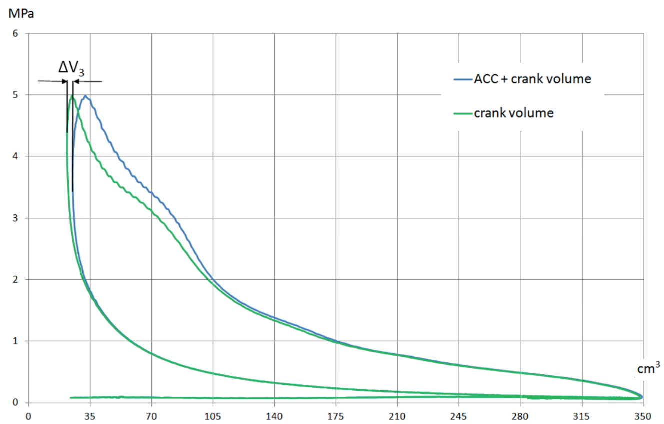

Fig. 8. Pressure volume diagram of the crank-piston system volume and cumulative volume (optimal reaction of ACC system)

For the graph presented in Fig. 7. the value is equal to ~0,074, what corresponds to the compression ratio of 13,5:1. This result is twice the difference between compression ratio calculated in standard and proposed approach. Nevertheless, the indicated efficiency advocates for the choice of proposed approach.

The replacement compression ratio can be presented by following equation:

$$\varepsilon_p = \frac{V_{min} + \Delta V_{1-3}}{V_{max}} = \varepsilon_{max} - \Delta\varepsilon$$

Table 2. Change of theoretical efficiency calculated in approximated way

Number of Figure	Compression ratio	Otto efficiency (%)	Difference $\eta_{\varepsilon max}$ - $\eta_{\varepsilon 1\text{-}3}$ (%)
Fig. 6	14,8 : 1	65,97 – 60,06	2,21 - 2,20
Fig. 7	11,8 : 1	62,74 – 57,85	5,431 - 5,43
Fig. 8	14,0 : 1	65,20 – 60,29	2,97 - 2,83

Therefore the equivalent compression ratio for any theoretical cycle realized by naturally aspirated ACC spark ignition engine, can be formulated as follows:

Equivalent compression ratio for the ACC engine is expressed by the ratio between the lowest cumulative volume of crank-piston system and ACC system reached before top dead center in compression stroke, and the highest cumulative volume of crank-piston system and ACC system, reached in bottom dead center for compression stroke.

3. ACC engine with VCR

For the presented definition three assumptions were made, to eliminate adverse cases:

1. Ignition takes place during the compression stroke before the top dead center point for the crank-piston system.
2. ACC system piston does the work "outside", what means it is in the power stroke (expansion) for ACC system and is moving up in the first displacement (as shown in Fig. 1).
3. Ignition takes place before reaching the smallest cumulative volume in compression stroke.

The first two assumptions raise no concerns, whereas the third one is considered safe, based on Authors' extensive experience with ACC engine. The purpose of it is to eliminate the cases in which the smallest volume appears before the ignition and is a result of improper adjustment and inertia of ACC system [2,8,9]. Including such value into calculations will result in incorrect estimation of compression ratio. Authors are aware that the definition proposed is not completely precise, but certain discrepancies are acceptable.

For proposed compression ratio schemes presented on Fig. 9, Fig. 10, Fig. 11 and Fig. 12 are prepared. These are simplified assuming that regulation of compression ratio is done continuously from the beginning until reaching the top dead center. This situation corresponds to horizontal pressure line on Fig. 11, reaching 360°ca, while in a real case it occurs at low rotation speeds and significant volumes in ACC system.

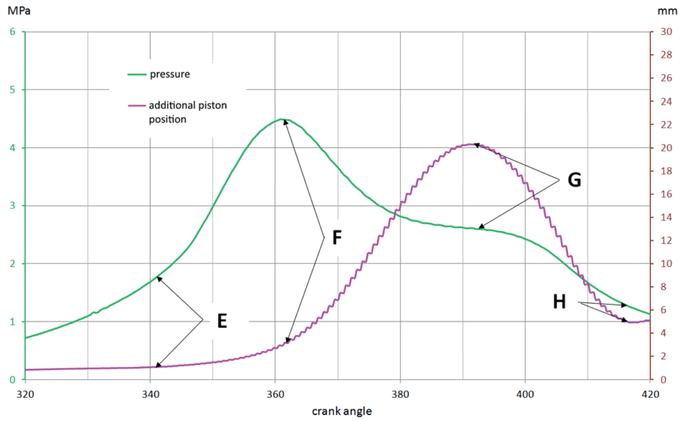

Fig 9. Four (E, F, G, H) characteristic points for real ACC engine subjected to partial load

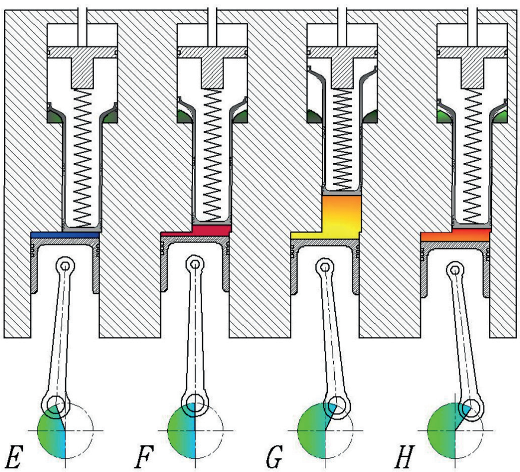

Fig. 10. Four (E, F, G, H) characteristic points of ACC engine

Fig. 9 and 10 present the principle of operation of ACC system with VCR function in real ACC engine with spark ignition, in its the most characteristic working points (E, F, G, H). Typical for real ACC system is the position of G point, which is the point of ignition, located between 20°ca and 30°ca after top dead center point. Point H is located between 40°ca and 65°ca after top dead center point, what is visible also on graph from Fig. 2. Within this limits also locations of points C and D were the most obtained during the measurements of the piston position (2) .

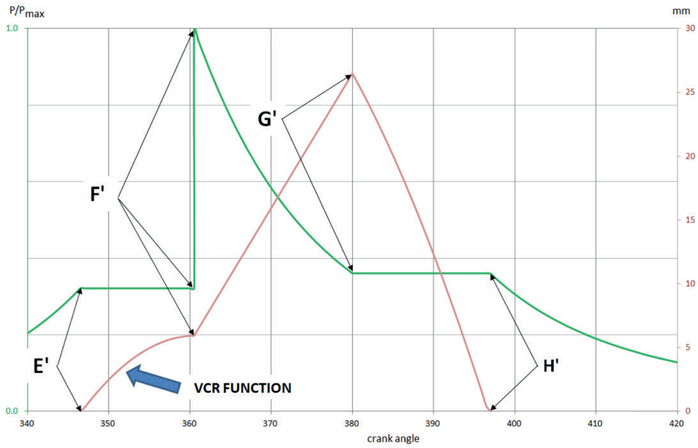

Fig. 11. Four characteristic points of work for theoretical cycle of ACC engine

The ignition delay was omitted, but compression ratio adjustment was taken into consideration, namely the advance displacement of additional piston in ACC. Fig. from 9 to 11 can be interpreted that a significant impact of ACC system takes place between points E and H for both theoretical and real engines. To be more precise, it takes place between 340°ca and 410°ca for the cycle of the real ACC engine.

For this reason, theoretical cycle of ACC engine cannot be used as Otto cycle to analyze following cycles of VCR engines, in which only compression ratio changes. A different concept is required.

The concept of work for the new cycle of ACC engine with VCR function can be presented as cycle where energy is absorbed by ACC system between points E and G, what is decreasing the compression ratio. The energy is always recuperated between points G and F. Fig.12 presents two superimposed theoretical cycles.

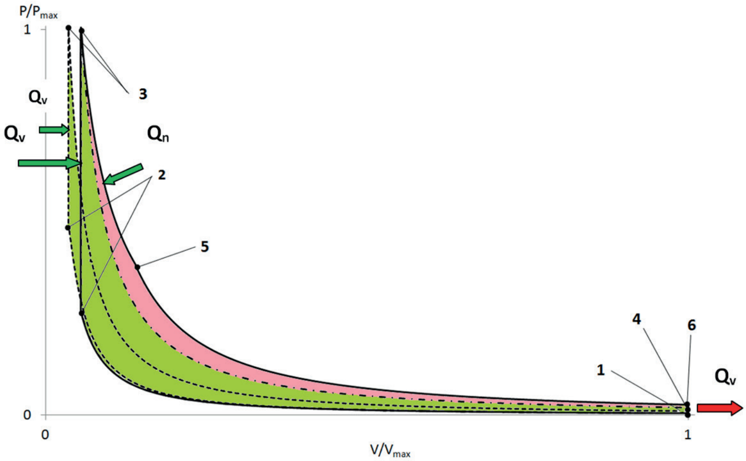

Fig. 12. Energetic concept for new theoretical cycle

Such division implies that during energy supply, as long as the maximum pressure Q_1=30% is not exceeded, the theoretical ACC engine will work with its maximum compression ratio of 30:1. When this value is exceeded, the compression ratio will drop down to the value of 20:1. In such case the ACC engine works according to Otto cycle.

Efficiencies of proposed cycle:

$$(\gamma = 1) \Rightarrow \eta_{prop} = \eta_1$$

$$(\gamma < 1) \Rightarrow \eta_{prop} = \eta_1\gamma + \eta_2(1-\gamma)$$

Table 3.

n	η_1	η_2	α
$n = 0$	$1 - \frac{1}{\varepsilon^{(\kappa-1)}}$	$1 - \varepsilon^{(1-\kappa)}\frac{(\alpha^{\kappa}-1)}{\kappa(\alpha-1)}$	$\left(\frac{1}{\kappa}\right)\left(\frac{1}{\gamma}-1\right)\left(1-\frac{1}{\beta}\right)+1$
$0 < n < 1$	$1 - \frac{1}{\varepsilon^{(\kappa-1)}}$	$1 - \varepsilon^{(1-\kappa)}\frac{(n-1)}{(n-\kappa)}\frac{(\alpha^{\kappa-n}-1)}{(\alpha^{1-n}-1)}$	$\left[\left(\frac{n-\kappa}{n-1}\right)\left(\frac{1}{\gamma}-1\right)\left(1-\frac{1}{\beta}\right)+1\right]^{\left(\frac{1}{n-1}\right)}$
$n = 1$	$1 - \frac{1}{\varepsilon^{(\kappa-1)}}$	$1-\varepsilon^{(1-\kappa)}\frac{(\alpha^{\kappa-1}-1)}{(\kappa-1)ln(\alpha)}$	$e^{\left[\left(\frac{1}{\kappa-1}\right)\left(\frac{1}{\gamma}-1\right)\left(1-\frac{1}{\beta}\right)\right]}$
$n = \infty$	$1 - \frac{1}{\varepsilon^{(\kappa-1)}}$	-	-

The efficiency course in theoretical ACC engine, calculated based on the formulas presented in Table 3 and chosen polytrophic indexes, are presented in Fig. 13.

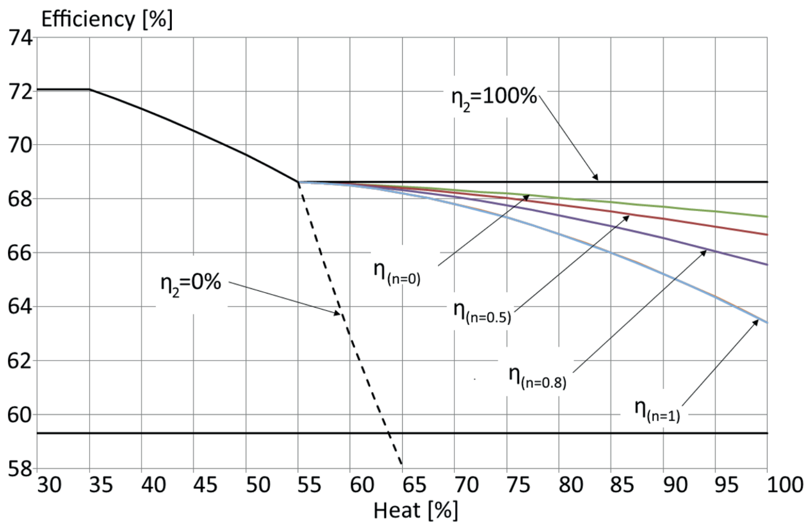

Fig. 13. Efficiency of the new Otto cycle as a function of load

Based on graph in Fig. 14 an assumption can be made that maximum efficiency will be reached when compression ratio will be equal to 30:1, and loading will be lower than 35%. Then it will decrease gradually as the loading will be increasing to 55%.This will be accompanied by decrease of compression ratio down to 20:1. When new energy supply appears, after exceeding 55% of load, this decrease will propagate further and will be dependent on polytrophic index. The efficiency will reach minimal value for the full loading, with polytrophic index equal to 1, as in isothermal process. Such course of theoretical efficiency gives the cycle of the ACC engine, with partial load of 55%, a significant advantage of 9-13% over the engine with constant compression ratio equal to 11:1.

Based on conducted theoretical calculations, Authors prepared also Fig. 14 with graphs of α, β and γ factors, which were used in calculations.

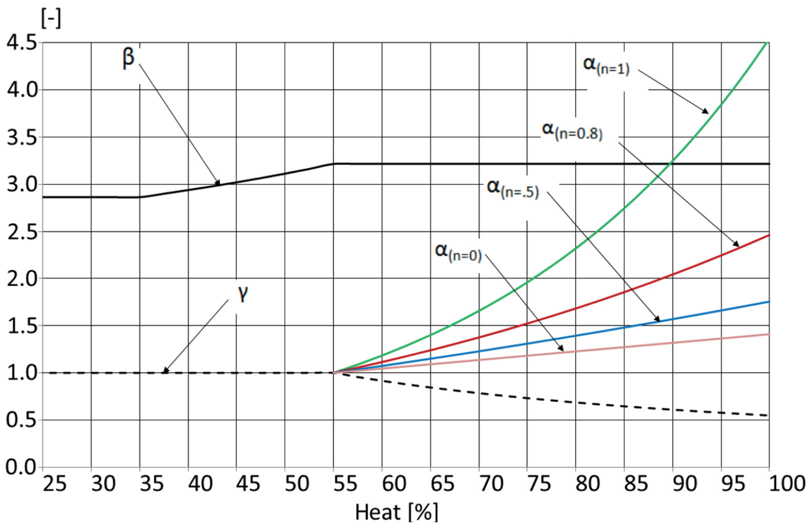

Fig. 14. Change of values for α, β and γ factors as a function of load

As expected, along with increase of polytrophic index n, also factor α is increasing, what is accompanied by decrease of complementary cycle efficiency η_2, the bigger, the smaller the γ factor is.

Table 4 presents the theoretical efficiency of ACC engine again, but it also includes the fact that maximal compression ratio in investigated ACC engine equal to 17,5:1 and is decreasing to value of 14:1, when a new energy supply occurs. Experimental value of 14:1 was determined based on the presence of detonative combustion and it was used to calculate the efficiencies presented in Table 4 for optimal response of the system.

Table 4. Changes to theoretical efficiencies after decrease of compression ratio to 14:1

Figure	Compression ratio	Efficiency Propose cycle	Comparison
Fig. 6	14,8 : 1	62,60%	3,21%
Fig. 7	11,8 : 1	58,37%	7,45%
Fig. 8	14,0 : 1	61,83%	3,98%

Such low compression ratio was due to specific design of the combustion chamber – taking into account some technical aspects resulted in non-compact structure.

In conclusion, the above calculations of the efficiency for the new proposed theoretical cycle have to be correlated with the efficiency of ACC system by itself and it is necessary to emphasize that it is a system contributing in additional loss, what is shown on Fig. 15. In this figure one can also notice the difference between red area, corresponding to energy accumulation, and green area, corresponding to energy recuperation. This difference resulted in effciency of η_{ACC}=66%.

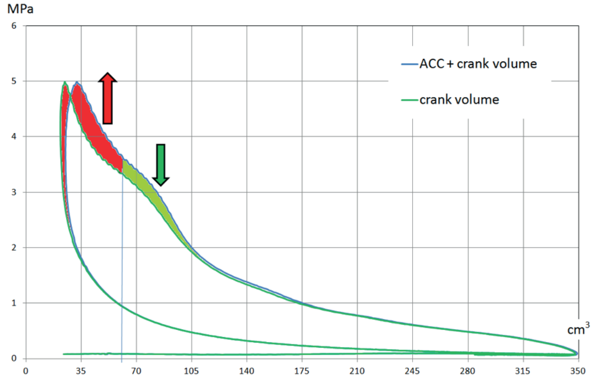

Fig. 15. Real energy balance for ACC system with VCR function for closed cycle

In the case of theoretical ACC engine with VCR function, energy accumulation is more beneficial than a system without this function since it takes place in compression stroke before top dead center.

The effect of this profiting balance can be noticed in Fig. 16, where theoretical ACC engine with VCR function with ACC engine without this function are compared.

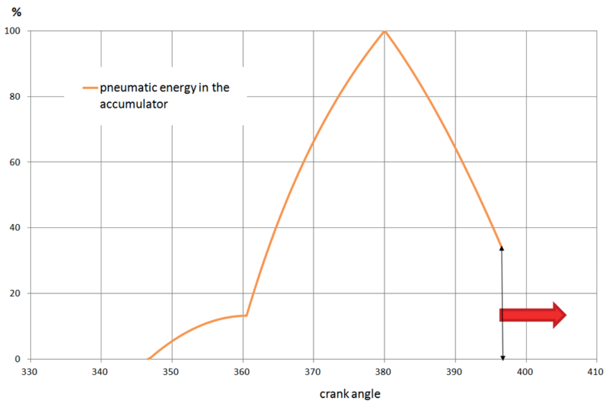

Fig. 16. The course of energy accumulation and recuperation in theoretical (frictionless) ACC system

Now, a simplified analysis will be performed for the cases with premature response of additional piston. As mentioned, this reaction can be induced twofold. For the purpose of calculations, the less beneficial possibility with advanced ignition was taken. Its efficiency has to be additionally decreased by negative loop of crank-piston system. In this new variant, implementation of additional point in theoretical cycle is necessary. A division of energy put into cycle is implemented, to better describe theoretical circuit with premature response of ACC system.

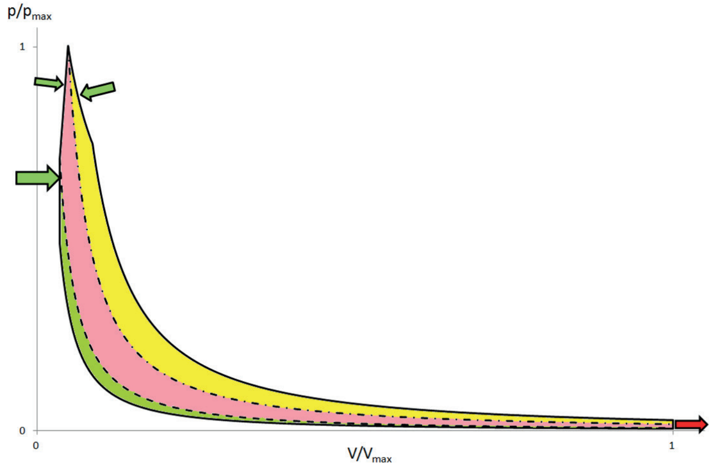

Fig. 17. Theoretical cycle of ACC engine with premature response

This cycle is presented in Fig. 17 together with the division of energy in three ways.

Fig 18, 19 and 20 are showing the course of the ACC system reaction in time unit, which correlates better with combustion process than the crankshaft rotation angle. From these graphs one can observe that reaction of ACC system is dependent on the load and should not be longer than few milliseconds. In Fig. 18, 19 and 20 the value of reaction was found in the range of 0,5 and 3 ms. This is a quick reaction for mechanical system working in straight-line motion with direction change.

In Fig. 18, despite a quick reaction of approximately 3 ms, it is not quick enough at 70% due to detonative combustion occurrence.

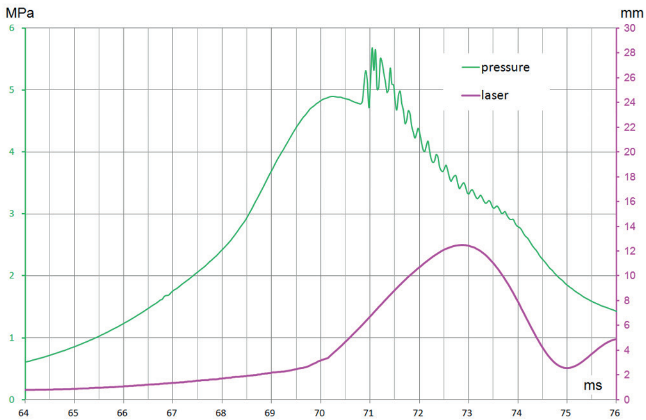

Fig. 18. Delayed response of ACC system

The second graph in Fig. 19, with premature reaction of 0,5 ms which is extraordinarily quick, shows a decrease in the efficiency. This graph is important also because it allows to compare the reaction times of ACC system from Fig. 18 with the time of detonative combustion (0,125 ms). The difference if fourfold, but ACC system is subjected to optimization, which can decrease this difference down to 3:1. With higher rotational speed this can eliminate the detonative combustion altogether.

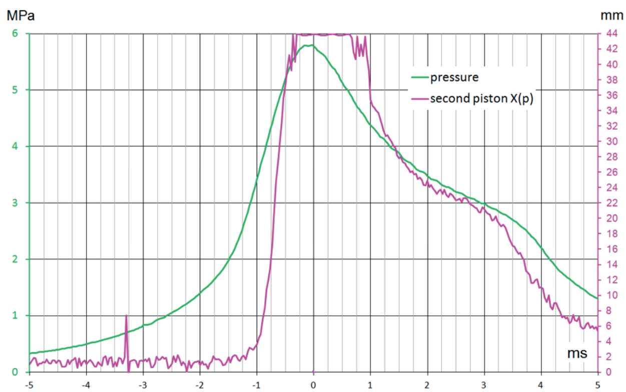

Fig. 19. Premature reaction of ACC system

The most favorable reaction of ACC system is presented in Fig. 20. In reality is not different from operation time of delayed reaction in Fig. 18, however here detonation does not occur. The essence of this reaction is earlier reaching the maximum, because already in 2,3 ms after top dead center in comparison with 4 ms after top dead center in Fig 20.

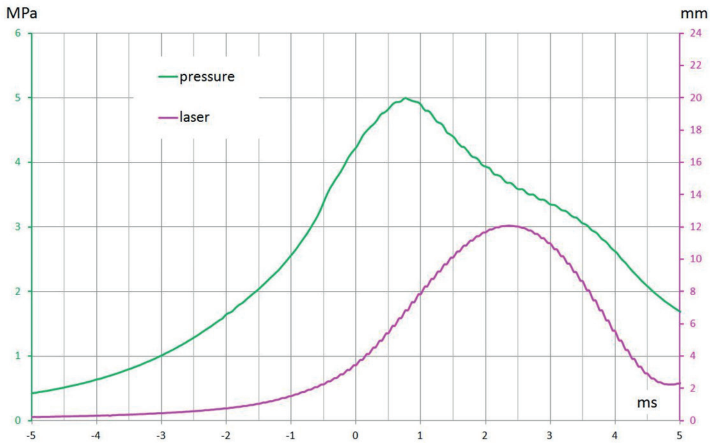

Fig. 20. Optimal reaction of ACC system in time

4. Conclusions

It can be stated, that additional kinematic ACC system, coupled with crank-piston system, without changing the engine construction, can at partial load successfully realize the function of compression ratio change. Basic feature of ACC engine, differentiating it from the engines with VCR function, is the fact that compression ratio in this engine, despite of conducted regulation, is never constant. Usually, it oscillates around an average value, being the result of the ignition and ACC system adjustment. The change of the compression ratio, defined in previous section, does not include all of the possible cases and is not entirely accurate, because of the lack of isolating the energetic power stroke and compression stroke before measurement.

Another important aspect, which differentiate such defined compression ratio from typical definitions, is that its value also influences the combustion process. Authors imply that with partial load the regulation of compression ratio can take place in wider range than for VCR engines. In presented calculations Authors included the value interval between 30:1 and 20:1, while in presented empirical investigations, because of construction limitations (this is the use of typical elements from spark ignition engines with low compression ratio of 8:1) and endurance limitations, maximal compression ratio was limited to the value of 17,5:1 in four-stroke ACC engine and 22,5:1 in two-stroke ACC engine.

Following conclusions can be drawn from conducted analyses:

1. VCR function in ACC engine proves to exert a beneficial influence on efficiency course. This can be observed in Fig. 14. This is why VCR function will be implemented in ACC engines. It is especially vital when it is working as a traction engine, which has partial load characteristic.

2. ACC engine with VCR can run with higher compression ratio, due to its kinematics-advanced response of ACC system. Higher initial speed provides shortest reaction times, especially with the upward movement of additional piston, when it reaches its first maximum. If we want to compare it with the work of crank-piston system, with the following parameters: displacement of 44 mm and cylinder diameter of 38 mm, such system would have to rotate with the speed of 60000 rpm.

3. ACC system should be appropriately tuned, implying that its reaction should be optimal, so nor advanced nor delayed. This is why the criterion of reaching maximal torque without detonative combustion occurrence is reasonable.

4. Advanced ignition is typical for ACC engine, but is not as adverse, because of the overlap between compression stroke and power stroke. The latter tends to dominate after a few tenths of a millisecond (Fig. 19, Fig. 20 and Fig. 21).

5. Optimal reaction time is not a rigid concept, because it depends on the rotation speed, load and combustion chamber shape.

6. More beneficial for ACC with VCR, is energy recuperation process, because accumulation process starts already with lower pressure in compression stroke, what reduces the differences between these two processes.

7. The efficiency of theoretical ACC system is mainly dependent on the kinematic system, which supplies the system with energy. Therefore, for its work, as the most important part for the general efficiency, all actions leading to increase of the efficiency for theoretical cycle of ACC engine should be related with it.

8. The aim of using ACC system requires that energy should be absorbed and returned in the most suitable moment that the balance was beneficial for the whole cycle.

9. ACC system, from its principle of operation, allows the temperature and pressure regulation during energy supply to the cycle. To be exact, this occurs the final phase.

10. In the calculations, the friction forces were neglected. Therefore, the loss of energy has to be interpreted as kinetic energy of ACC system. This energy in real engine is dissipated in form of damped vibrations.

11. The difference between theoretical and real cycle of ACC engine is the final temperature during decompression (in power stroke). It is caused by energy dissipation that occurs earlier than in other cycles. It takes place during the energy recuperation and influences the maximal temperature of the cycle.

12. In considerations of ACC's theoretical cycle, Authors skip the way of energy recuperation, which results in a lack of exact solution. It is connected with the fact that the means of energy dissipation are not defined.

In presented analyses, Authors have assumed the most probable behavior of ACC system in different circumstances. However, this does not rule out other interpretations i.e. with more detailed energy division scheme, than presented in this paper. Authors are hoping to start a general discussion about the definition of compression ratio in kinematic systems, where compression and decompression strokes are overlapping, as it takes place in ACC engine.

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