EFFICIENCY CHARACTERISTICS OF A NEW QUASI-CONSTANT VOLUME COMBUSTION SPARK IGNITION ENGINE

by

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A zero dimensional model has been used to investigate the combustion performance of a four cylinder petrol engine with unconventional piston motion. The main feature of this new spark ignition engine concept is the realization of quasi-constant volume during combustion process. Presented mechanism is designed to obtain a specific motion law which provides better fuel consumption of internal combustion engines. These advantages over standard engine are achieved through synthesis of unconventional piston mechanism. The numerical calculation was performed for several cases of different piston mechanism parameters, compression ratio and engine speed. Calculated efficiency and power diagrams are plotted and compared with performance of ordinary spark ignition engine. The results show that combustion during quasi-constant volume has significant impact on improvement of efficiency. The main aim of this paper is to find a proper kinematics parameter of unconventional piston mechanism for most efficient heat addition in spark ignition engines.

Key words: *spark ignition engine, quasi-constant volume combustion, efficiency*

Introduction

Since the very birth of the spark ignition internal combustion (IC) engine by Leonardo da Vinci in 1509 [1], the desire to improve its performance was the prime mover force both for experimental and theoretical research. Although, over the decades, much progress was made on both the practical and the theoretical side, virtually all technical advances were achieved by intuition or experimental trial and error methods rather than by rigid derivations and implementations based on fundamental laws. It is well known that transport is almost totally dependent on fossil, particularly, petroleum-based fuels such as gasoline, diesel fuel, liquefied petroleum gas (LPG) and natural gas (NG). Fuel consumption, emission by the transport sector and overall motor vehicle efficiency are important topic these days. There is a strong drive towards legislation limiting the fleet average CO_2 emission [2]. Conventional IC engines are based on a relatively simple solution to achieve a thermodynamic cycle while providing mechanical power. While the performance, emissions and reliability of IC engines have been improved significantly, the fundamental principle of crank-rod-piston slider mechanism still remains largely unaltered. In theory, the most efficient thermodynamic cycle for IC engines is the Otto cycle [3], which consists of isentropic compression and expansion processes and constant volume heat addition and rejection processes [4-6]. A series of achievements have ensured since finite-time thermodynamics was used to analyze and optimize real heat-engines [7-11], from this publications can be

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concluded that the most important parts of the cycle which determine the efficiency are the constant volume heat addition at high compression ratios [12, 13]. This fact provides the highest thermal potential of the various possible thermodynamic cycles which are suitable for IC engines, and the subsequent expansion process, which converts the thermal potential into work. In reality, neither conventional spark ignition (SI) or compression ignition or even the modern developed homogeneous charge compression ignition (HCCI) and controlled auto ignition (CAI) combustion processes, can achieve the efficiency level suggested by the ideal thermodynamic cycles [14-16]. Only the Otto cycle delivers theoretical maximum thermal efficiency. The traditional design of IC engines using a simple slide-crank mechanism gives no time for a constant volume combustion which significantly reduces the cycle efficiency [17].

The optimal piston trajectories of the IC engine cycles for different optimization objectives have been studied by many researches. Mozurkewich and Berry described improved engine performance by optimized piston motion [18, 19]. Hoffman and Berry [20] and Blaudeck and Hoffman [21] defined optimal paths for diesel cycles. Teh and Edwards [22-24] and Teh *et al.* [25, 26] works on thermodynamic requirements for maximum IC engine cycle efficiency and optimal control approach to minimizing entropy generation. Optimal piston paths for diesel engines was also examined by Byrzler [27] and Byrzler and Hoffman [28]. The optimal path of piston motion of irreversible Otto cycle was presented by Ge *et al.* [29-31]. Optimizing piston velocity profile for maximum work output was done by Chen *et al.* [32], and engine performance improved by controlling piston motion was presented by Xia *et al*. [33].

Modeling the performance of IC engines has been a continuing effort over the years and many models have been developed to predict IC engine performance parameters. Zero dimensional (Zero-D or 0-D) models are the most commonly preferred analytical tools for IC engine development [34, 35]. For 0-D models, most properties are averaged over the total volume and no spatial information is available. A survey of thermodynamic models for cylinders are presented by Blumberg *et al*. [36]. Generally speaking 0-D models are one of the simplest and fastest methods to model engine combustion processes. Engine designers may find that experimentally based Zero-D codes are more useful for design and development applications.

On the basis of well known mathematical tool for 0-D modelling of SI engine a further step was made in this paper through the analysis of a new engine concept which is able to make longer dwell of piston near the top dead center (TDC). Quasi-constant volume (QCV) IC engine is able to provide longer dwell angle at TDC than ordinary IC engine. Also in this paper was presented basic description of the new engine that will be able to realize thermodynamic cycle with increased efficiency. The proper optimization criteria to be chosen for the optimum design of the heat engines may differ depending on their purposes and working conditions. If the heat engine design was done not to obtain maximum work or power, but to have maximum benefit from energy, then the design objective is to get maximum efficiency. Fuel consumption is main concern for heat engines so the maximum thermal efficiency criterion is very important. Despite the fact that for engines of race car, maximum power output criterion is significant, for engines of passenger car, both fuel consumption and crank moment gain may equally important, in such a case both the power and thermal efficiency criteria have to considered in the design. In this paper will be also investigated some power characteristics of QCV engine.

Quasi-constant volume spark ignition engine

Fact is that real engine is not able to make heat addition during constant volume. When the engine operating, the piston can only reciprocate continuously between TDC and bottom dead center (BDC) at a frequency proportional to the engine speed. The chemical reaction process associated with combustion events, however, essentially takes a fixed-time to-complete, which is relatively independent of the engine speed. In order to maximize the work obtained from the heat energy released by combustion, the air/fuel mixture has to be ignited prior to the piston reaching TDC, and the ignition timing should be adjusted according to the engine speed and the quality of the air/fuel mixture. Clearly, the early stage of the heat release before the piston reaches TDC results in negative work. During the combustion event, the piston movement is defined by the crank rotation, so that truly constant volume heat release is not achievable. The ideal scenario is to initiate and complete the combustion event while the piston remains at the TDC position. A practical method is to reduce the engine crank rotation velocity at the TDC position to provide extra time for completing the combustion. This will then generate a new combustion cycle, QCV, that sit between conventional IC engine combustion cycle and ideal Otto constant volume combustion cycle.

QCV SI engine is presented on fig. 1. As can be seen from the described illustration pistons make a movement conditioned by the mechanism consisting of crankshaft and connecting disc. In this article will not be presented detailed description of this concept, since it is not the intention of the authors to propose a kinematic analysis of a new internal combustion engine design but only thermodynamic features and advantages over ordinary SI engines.

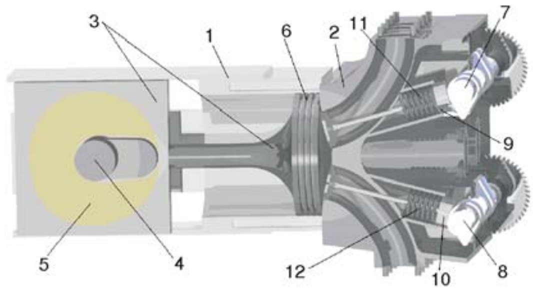

Figure 1. CAD model wtih basic parts of new QCV SI engine [37]
1 – engine block, 2 – engine head, 3 – piston, 4 – crankshaft, 5 – connecting disc, 6 – piston rings, 7 – intake camshaft, 8 – exhaust camshaft, 9 – intake valve, 10 – exhaust valve, 11 – intake valve spring, 12 – exhaust valve spring

Dwell time or dwell angle is important fact during combustion process. In conventional engine this dwell angle can be changed due to variations of ratio between connecting rod and crank radius. Piston dwell at TDC and at BDC are often mentioned, it should be noted that strictly, there is no dwell period in ordinary mechanism. The piston comes to rest at precisely the crank angle that the crank and rod are in line (TDC and BDC), and is moving at all other crank angles. At crank angles which are very close to the TDC and BDC angles, the piston is moving slowly. It is this slow movement in the vicinity of TDC and BDC that give rise to the term piston dwell. In this described concept there is also no piston dwell in classical sense, there is only very small changes of volume near TDC, especially for large λ_k ratio. If the piston dwells longer near TDC and ignition is initiated properly, there will actually be a longer period of time for the pressure created during combustion to press against the top of the piston. Also, if the dwell period is too long, there is a possibility for unfavorable energy conversion.

Unconventional piston motion

In the previously section was defined basic shape and parts of new QCV IC engine concept. From the presented fig. 1 it is clear that there is a certain similarity between ordinary and this unconventional engine design. First of all, this piston mechanism is one special case of ordinary piston linkage that can be found in conventional internal combustion engine, but with some special features that will be explained in this section. It may be noted that the kinematic scheme of this mechanism is very similar with ordinary piston mechanism, it can be said that

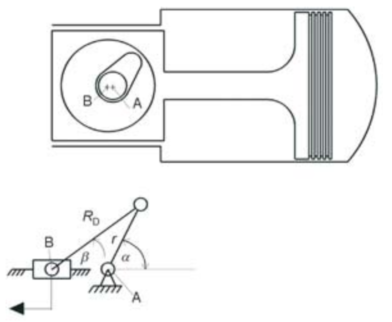

Figure 2. Kinematics of the unconventional crankshaft drive

there is only two main differences. First is due reverse setting of piston mechanism, and the second is feature that with such design of the mechanism is possible to achieve very large ratio between crankshaft radius and connecting rod length. Kinematics scheme of unconventional piston mechanism is presented on the following fig. 2.

For the piston path $s(\alpha)$, it follows from fig. 2, where R_D is radius of connecting disc, r – the radius of crankshaft, and D – the piston diameter:

$$s(\alpha)=\sqrt{(r+R_\mathrm{D})^2}-\sqrt{R_D^2[r\sin(\alpha-\beta)]^2}-r\cos(\alpha-\beta) \tag{1}$$

The derivative provides for the piston speed relation:

$$\frac{\mathrm{d}s}{\mathrm{d}\alpha}=r\sin(\alpha-\beta)+\frac{r[r\sin(\alpha-\beta)\cos(\alpha-\beta)}{\sqrt{R_D^2}-[r\sin(\alpha-\beta)^2} \tag{2}$$

From the definition of the cylinder volume:

$$V(\alpha)=V_\mathrm{C}+D^2\frac{\pi}{4}s(\alpha) \tag{3}$$

the alteration of cylinder volume follows in (4):

$$\frac{\mathrm{d}V}{\mathrm{d}\alpha}=D^2\frac{\pi}{4}\frac{\mathrm{d}s}{\mathrm{d}\alpha} \tag{4}$$

Presented mathematical description are reserved only for mechanism without eccentric ratio, there is also possibility to create this unconventional mechanism with eccentric ratio, but such case will not be discussed in this article. Finally we can obtain eq. (5) that represents the piston path:

$$s(\alpha)=r\left\{[R_\mathrm{D}-\cos(\alpha)]+\frac{1}{\lambda_\mathrm{k}}[1-\sqrt{R_D-\lambda_k^2\sin^2(\alpha)}]\right\} \tag{5}$$

In theory the ideal scenario is to initiate and complete the combustion event while the piston remains at the TDC position. In practice, too long piston dwell in TDC can lead to high heat loss through the cylinder walls [13]. It can be concluded that in real engine piston dwell at TDC is not necessary good solution for high efficiency energy conversion, only with optimisation of the time, *i. e.* angle, when the piston is near TDC will contribute to higher heat addition efficiency. Also, there are several more potential when it comes with dwell angle near TDC. Small changes of piston position near TDC provides the maximum thermal potential and eliminates the negative work due to early ignition which is well into compression stroke with conventional engine strategies. With piston dwell near TDC backflow due valves overlap could be avoid. Finally, if the combustion event completes at the TDC, the effective expansion stroke can be maximally extended to fully use the thermal energy as well as to provide sufficient time for post combustion reactions, thereby reducing partial burned emissions.

Cycle modelling

For the present study, a 0-D combustion model is employed. During numerical considerations in this paper several assumptions were carried out. First of all, in this paper was used constant specific heats behaviour. Also, the cylinder gas is air and it obeys the ideal gas law. One of the major assumptions as well is uniform crank speed. For simplification of the model we neglected that there is no gas leakage through valves or piston rings, this actually means that the mass in cylinder is constant. The contents of the cylinder are fully mixed and spatially homogeneous in terms of composition and properties during intake, compression, expansion, and exhaust processes. Finally, gas in cylinder moves through equilibrium states.

Thermodynamic model

The first law of thermodynamics to an open system yields the energy equation as eq. (6):

$$E' = Q' - W' + \sum_{v} m_{\mathrm{v}} h_{\mathrm{v}} \tag{6}$$

The rate of change of mass within any open system is the net flux of mass across the system boundaries. These boundaries in ideal engine are only intake and exhaust valves, in this case we have acepted assumption that there is no leakage of working fluid during compression or expansion process. As can be concluded, we can starting from general form of the first law for a closed, transient system:

$$\frac{\mathrm{d}E}{\mathrm{d}t} = Q' - W' = \frac{\mathrm{d}Q}{\mathrm{d}t} - \frac{\mathrm{d}W}{\mathrm{d}t} \tag{7}$$

Under the assumption we stated above now we can write the relations – eq. (8):

$$E = mu \tag{8}$$

and for the work done by system:

$$W = \int p\mathrm{d}V \Rightarrow \mathrm{d}W = p\mathrm{d}W \tag{9}$$

After inserting eq. (8) and eq. (9) into eq. (7) we obtain new relation – eq. (10):

$$\frac{\mathrm{d}(mc_{\mathrm{V}}T)}{\mathrm{d}t} = \frac{\mathrm{d}Q}{\mathrm{d}t} - \frac{p\mathrm{d}V}{\mathrm{d}t} \tag{10}$$

The equation of state for an ideal gas is [1]:

$$pV = m\mathrm{R}T \tag{11}$$

So we can substitute with eq. (11) to eliminate temperature from eq. (10):

$$T = \frac{pV}{m\mathrm{R}} \tag{12}$$

Now we have new relation:

$$\frac{c_{\mathrm{V}}}{\mathrm{R}} \frac{d(pV)}{\mathrm{d}t} = \frac{\mathrm{d}Q}{\mathrm{d}t} - p\frac{\mathrm{d}V}{\mathrm{d}t} \tag{13}$$

Using the product rule for the derivative of a product we can obtain:

$$\frac{c_{\mathrm{V}}}{\mathrm{R}} \left[V\frac{\mathrm{d}p}{\mathrm{d}t} + p\frac{\mathrm{d}V}{\mathrm{d}t} \right] = \frac{\mathrm{d}Q}{\mathrm{d}t} - p\frac{\mathrm{d}V}{\mathrm{d}t} \tag{14}$$

where E is the energy, m – the mass, u – the specific energy, W – the work, p – the pressure, V – the volume, c_{v} – the specific heat under constant volume, T – the temperature, R – the gas constant, t – the time, and Q – the heat. Part dV/dt depending on engine speed and engine geometry, and for this engine concept it will be different than in conventional engine. Although with obvious differences, there are also some certain similarities of these two approaches of engine design. From fig. 2 it can be concluded that this is actually ordinary piston mechanism, but with very large connecting rod and crankshaft radius ratio. In fact, in this research we studied one special case of piston mechanism where is length of connecting rod (connecting disc radius) very similar with value if the length of crankshaft radius. In other hand, this unconventional piston mechanism is opposite in relation to standard piston mechanism, inverted or opposite piston path can be described with fig. 4, where is represented piston motion law for different kinematic parameters.

In this paper we try to find optimal value of piston motion law, *i. e*. connecting disc radius and crankshaft radius ratio, in order to meet high efficiency combustion process. It is clear that with selecting the different piston motion law there would be changes in power and efficiency characteristics, like in ordinary engine, but in this concept there is larger field for finding the best solution.

The previous eq. (6) we can solve for dp/dt:

$$\frac{\mathrm{d}p}{\mathrm{d}t} = \frac{1}{V}\frac{\mathrm{R}}{c_\mathrm{V}}\frac{\mathrm{d}Q}{\mathrm{d}t} - \frac{p}{V}\left(1+\frac{\mathrm{R}}{c_\mathrm{V}}\right)\frac{\mathrm{d}V}{\mathrm{d}t} \tag{15}$$

Since we known that R can be expressed like:

$$\mathrm{R} = c_p - c_\mathrm{V} \tag{16}$$

and

$$k = \frac{c_p}{c_\mathrm{V}} \tag{17}$$

than we can obtain:

$$\frac{\mathrm{R}}{c_\mathrm{V}} = k - 1$$

In this research we have aceepted assumption that angle speed of engine is constant, and if instantaneous crank angle is:

$$\alpha = \omega t \tag{18}$$

relation (15) can be expressed through eq. (19):

$$\frac{\mathrm{d}p}{\mathrm{d}\alpha} = \frac{(k-1)}{V}\frac{\mathrm{d}Q}{\mathrm{d}\alpha} - k\frac{p}{V}\frac{\mathrm{d}V}{\mathrm{d}\alpha} \tag{19}$$

Heat release model

Thermodynamic models of the real engine cycle have served as effective tools for complete analysis of engine performance and sensitivity to various operating factors [38, 39]. In order to modeling combustion we use several approaches and mathematical models, which have the goal to describe the actual heat release via combustion as exactly as possible by means of the so-called substitute heat release rates which is shown in fig. 3. For SI engines, the mass fraction of burnt gases (x_b) is computed by using a Wiebe function [40, 41].

In order to adjust a real combustion by means of a Wiebe substitute heat release rate, there are various methods for determining the three Wiebe parameters: start of combustion, combustion duration, and form parameter. Fuel mass that was burned during combustion in relation to crank angle can be simulated with eq. (20):

$$x_{\mathrm{b}} = \frac{m_{\mathrm{b}}}{m_{\mathrm{f}}} = 1 - \exp\left[-c\left(\frac{\alpha - \alpha_0}{\Delta\alpha}\right)^{m+1}\right] \tag{20}$$

where α is the crank angle, α_0 – the crank angle at the start of combustion, $\Delta\alpha$ – the total combustion duration (from $x_{\mathrm{b}} = 0$ to $x_{\mathrm{b}} \approx 1$), and c and m are adjustable parameters which fix the shape of the curve. In eq. (20) m_{b} represents the mass of burnt fuel and m_{f} the mass of total fuel. The total value of the heat that can be released from the combustion of the quantity m_{f} of fuel is Q_{in} and is controlled by the combustion efficiency, η_{c}, described through eq. (21):

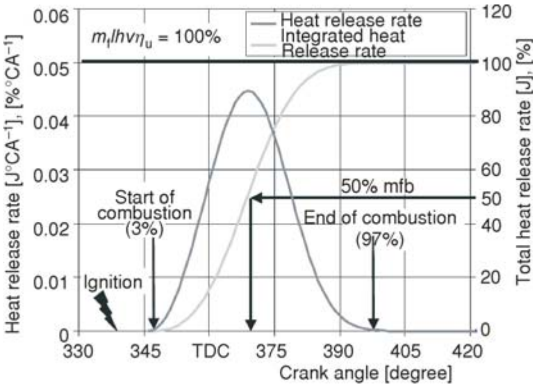

Figure 3. Characteristic attributes of the combustion path [24]

$$Q_{\mathrm{in}} = \eta_{\mathrm{c}} m_{\mathrm{f}} Q_{\mathrm{LHV}} \tag{21}$$

where Q_{LHV} is the lower heating value of the fuel and m_{f} – the total mass of fuel trapped into the cylinder.

Heat transfer

Heat transfers inside internal combustion engines are of convective and radiative nature. However, for SI engines, the radiative heat transfer are negligible since it accounts for only 3 to 4% of the total heat transfer [42, 43]. This cannot be applicable to diesel engines where the radiative heat transfer can represent up to 10% of the heat exchanges due to soot formation during combustion. During combustion, the burned gas temperature increases significantly with maximum which can reach about 2800 K. This induces gases expansion and thus an increase in their movement. It is during this period that heat transfers are the most important. Heat flux induced can reach several tens megawatts per square meter for some engines [44]. Net heat can be expressed through eq. (22):

$$\frac{\mathrm{d}Q}{\mathrm{d}\alpha} = \frac{\mathrm{d}Q_{\mathrm{in}}}{\mathrm{d}\alpha} - \frac{\mathrm{d}Q_{\mathrm{loss}}}{\mathrm{d}\alpha} \tag{22}$$

where $\mathrm{d}Q_{\mathrm{loss}}$ is the heat loss. From the definition of the heat transfer coefficient, we can obtain next relation:

$$\frac{\mathrm{d}Q_{\mathrm{loss}}}{\mathrm{d}\alpha} = h_{\mathrm{w}} A (T_{\mathrm{g}} - T_{\mathrm{w}}) \frac{1}{\omega} \tag{23}$$

The convective heat transfer coefficient is given by the Woschni model as [45-47]: where h is the heat transfer coefficient, A – the surface area in contact with the gases, T_{g} – the mean charge temperature, and T_{w} – the wall temperature.

Friction work

Friction work in SI engine can be divided into three major components. These components are pumping work, friction work and accessory work. The pumping work (W_{p}) can be described as net work per cycle done by the piston on the in-cylinder gases during the inlet and exhaust strokes. Friction work, *i. e.* rubbing friction work (W_{rf}) is the work per cycle dissipated in overcoming the friction due to relative motion of adjacent components within the engine. Finally accessory work (W_a) is the work per cycle required to drive the engine accessories [48]. Now, we can obtain relation of total friction work as follows:

$$W_f = W_p + W_{rf} + W_a \tag{24}$$

In this research we will examine only wide open throttle (WOT) operating situations. From the publications [48, 49] the total friction mean effective pressure (TFMEP) for different four stroke SI engine can be expressed as a function of engine speed are well correlated by an equation of the form:

$$TFEMP = 0.97 + 0.15\frac{n}{1000} + 0.05\left(\frac{n}{1000}\right)^2 \tag{25}$$

where n is the angular speed of crankshaft. The brake mean effective pressure of the standard engine can be found from:

$$IMEP = BMEP + PMEP + FMEP \tag{26}$$

This concept also eliminates contact between piston and cylinder, so there is no normal force on cylinder wall during piston motion, this feature of concept greatly reduces friction on the pistons and piston rings, on the other side unconventional IC engine design have some other friction losses. Normal force from cylinder wall was replaced with the normal force between piston and engine block. Such mechanism setting favors the piston-cylinder assembly because now we can assume contactless piston motion through cylinder.

Results and analysis

The constants and parameter values used in this paper are: D = 85 mm, r = 50 mm, $\lambda_\kappa =$ $= 0.33^{-1}$, $h_c = 0.9$, Q_{LHV} = 44400 J/kg, $PMEP + FMEP$ = 446 kPa, T_W = 400 K, c = 6.9, m = 2, $m_h =$ $= 0{,}8$, D_a = 50 deg., e = 8-12, n = 1000-6000 rpm. Cases were studied numerically for values of the n = 1000, 2000, 3000, 4000, 5000, and 6000 per min, e = 8, 9, 10, 11 and 12 and λ_κ = 0.33, 0.4, 0.5, 0.55, 0.71, 0.78, and 0.86.

With this movement, the piston is able to make such motion where heat addition can be done during piston dwell. The design geometry creates a pause or dwell in the piston's movement near TDC, while the output shaft continues to rotate. Surely, very long piston dwell near TDC (cases when λ_κ is very near values of 1) will decrease efficiency and overall engine processes. Only few values of λ_κ are interesting for application in SI engine. Adding these constant volume dwell cycles improves fuel burn, maximizes pressure, and increases cylinder charge. With changing λ_κ ratio fuel burn can be precisely controlled by maintaining a minimum volume (TDC piston dwell) throughout the burn process, containment maximizes pressure and burn efficiency.

After solving necessary equation for piston motion it is easy to plot results of piston path, speed and acceleration for several investigated cases. Piston path for several most interesting investigated cases was presented with fig. 4. On the fig. 5 was plotted only piston speed in relation of selected kinematic ratio λ_κ. It can be concluded that all selected cases have longer piston dwell near TDC. Although there is obvious similarity between these two piston mechanism there is a difference between ordinary and unconventional piston mechanism for the same kinematics ratio, these differences in piston motion are represented through two path curves noted with $\lambda_\kappa^* = 0.33$ and $\lambda_\kappa = 0.33$, where is with λ_κ^* noted piston motion law of ordinary IC engine.

It is interesting to see how this unconventional piston motion have impact on piston velocity and acceleration. These results of numerical analysis was shown in figs. 5 and 6. It can

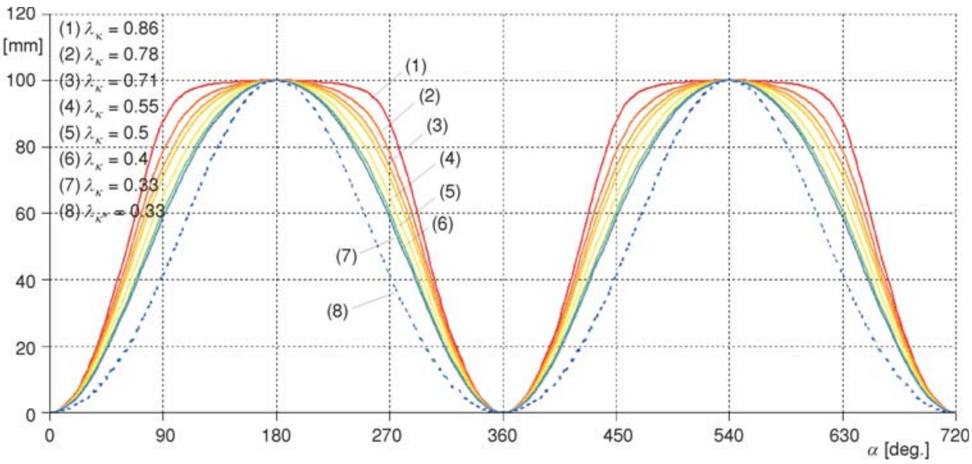

Figure 4. Piston path in relation of crank angle and selected ratio λ_κ

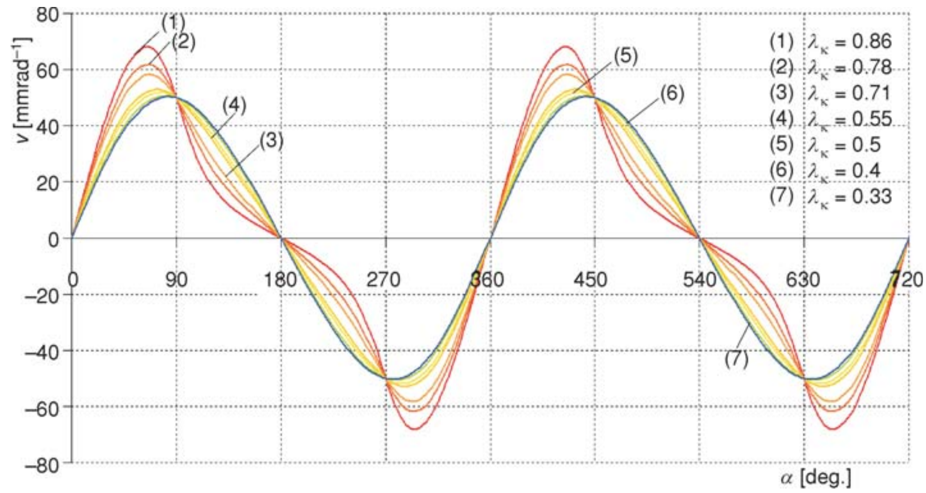

Figure 5. Piston velocity in relation of crank angle and selected ratio λ_κ

be concluded from the presented results that there is a noticeable differences between piston motion in classical and new concept. Compared to the conventional, this concepts of engine provides some 25 to 35% longer piston dwell at combustion dead center (CDC), *i. e.* additional time for the preparation of the mixture and the combustion of the fuel at better thermodynamic conditions. Even though the piston accelerates slower in transition, the piston ultimately reaches higher speeds to cover the additional stroke. This increase in piston speed means greater component strain.

After kinematics consideration, which is necessary part of other thermodynamic equation, we can calculate efficiency and power characteristics of QCV engine. In the next fig. 7 are plotted power-efficiency curves for several λ_κ ratios in relation of compression ratios.

From the plotted results is evident that the high efficiency regimes are those with middle range power. Also, the most powerful operating points are contributed with smaller effi-

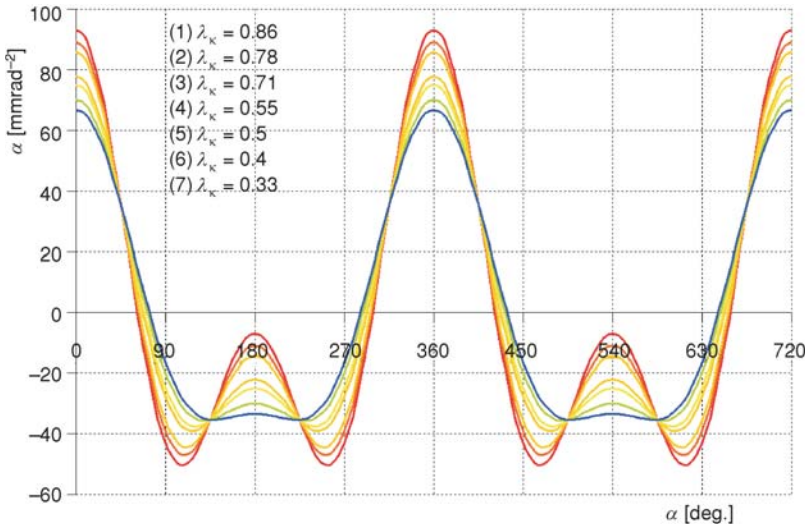

Figure 6. Piston acceleration (a) in relation of crank angle and selected ratio λ_κ

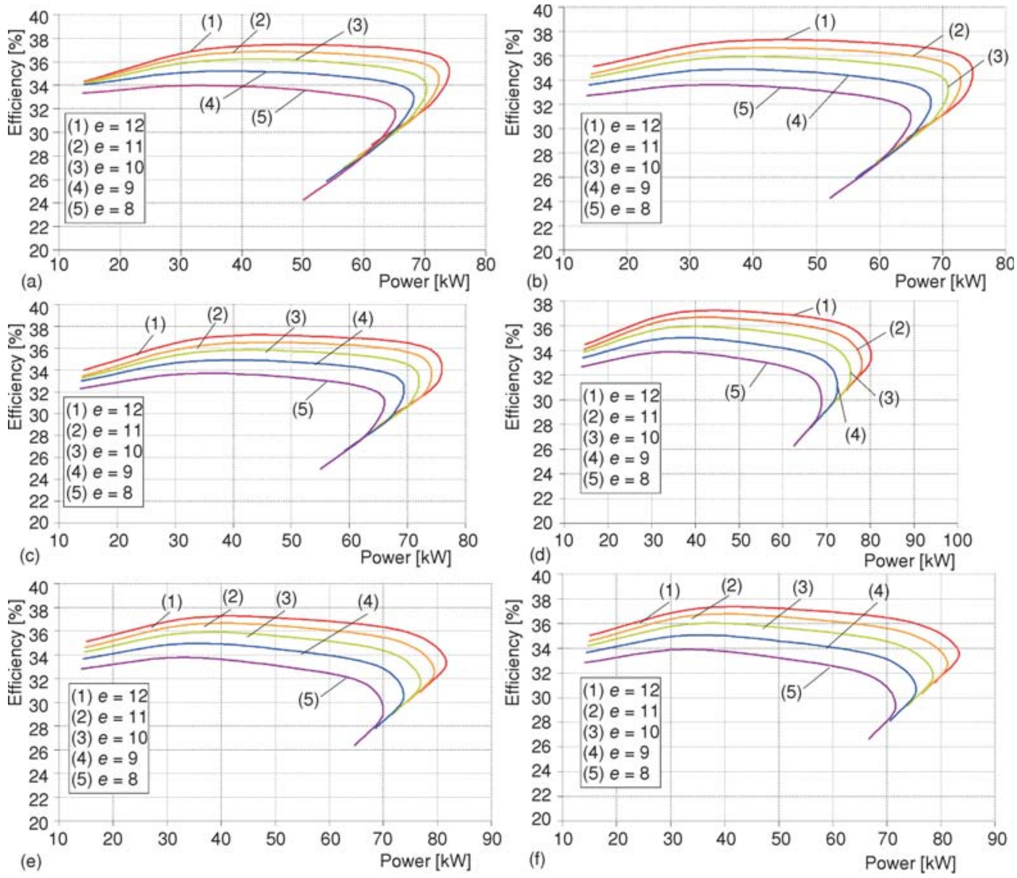

Figure 7. Efficiency-power characteristics for (a) λ_κ = 0.86, (b) λ_κ = 0.78, (c) λ_κ = 0.71, (d) λ_κ = 0.5, (e) λ_κ = 0.4, and (f) λ_κ = 0.33

ciency. It is well known that with the increasing of compression ratio there is also improvement in engine efficiency and power, some results of impact of compression ratio on engine efficiency are represented in fig. 8. It can be noted that the largest improvement of efficiency are reserved for increasing of compression ratio for smaller values of compression ratio, this feature is well known from theoretical cycles. It can be seen that the indicated thermal efficiencies and, hence, the work done increase with increasing the compression ratio, higher compression ratio means compression into a smaller volume at TDC, raising the pressure and temperature at the end of compression.

Figure 9 provide evidence that in relation to engine speed there is also one optimal value for engine efficiency, and this value of speed are similar for different values of compression ratio. Now, we can compare different piston dwell motion law in order to clarify best solution, such comparison was presented in fig. 10.

Finally, overall improvement of QCV combustion cycle over ordinary SI engine cycle can be seen from diagram described in the previous fig. 11. Where are η_u – efficiency of unconventional engine and η_o – efficiency of the ordinary engine. This chart and previous fig. 10 provides evidence that smaller piston dwell are better option than very long piston dwell. As can be seen with very long piston dwell gives higher heat loss than smaller dwell angles which contribute to reduced efficiency. The chosen QCV cycle changes the piston movement profile and produces longer residual time at TDC to favour the combustion optimisation.

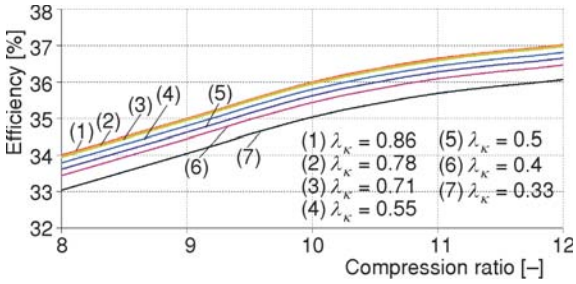

Figure 8. Efficiency of unconventional IC engine in relation to selected compression ratio and λ_κ for a constant engine speed of 3000 rpm

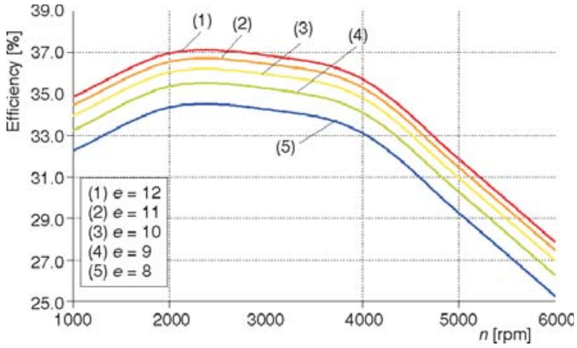

Figure 9. Efficiency of QCV engine for λ_κ = 0.5 in relation of engine speed

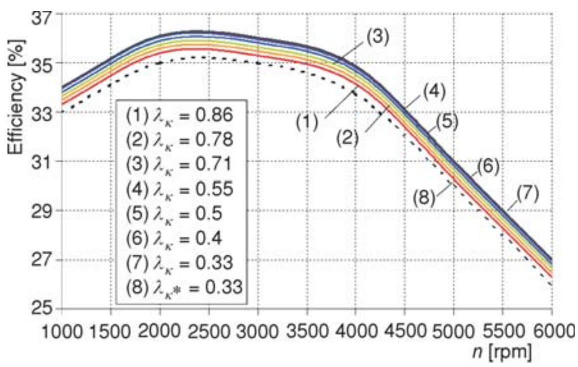

Figure 10. Efficiency comparison between ordinary SI engine and QCV SI engine

We can also noted some certain limits of efficiency improvement due the piston dwell, from the same chart van be concluded that value of efficiency improvement comes to the saturation point with reducing dwell angle. From the presented results may lead to the conclusion that opti-

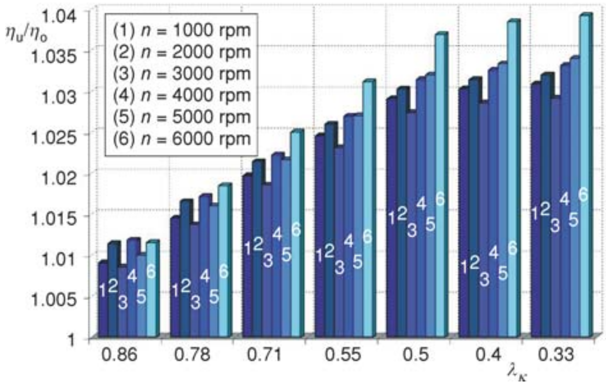

Figure 11. Overall improvement of efficiency through QCV combustion cycle in relation of *n* and λ_K

mal value of λ_K are around value 0.3. It should be mentioned that is much easier to perform an engine construction with greater value of λ_K, because in that case dimension of piston mechanism parts are smaller than in option when is required very small value of λ_K.

Conclusions

The engine used for most contemporary motor vehicles is the four-stroke SI internal combustion engine. A novel Otto cycle engine concept, in which intake and compression are carried out through unconventional piston mechanism, is presented. Numerical simulations were performed to optimize the cranking mechanism for achieving high thermal efficiency. The performance of a unconventional Otto heat engine is investigated by considering the 0-D modelling. It is found that there are optimal values of the piston dwell at which both the power output and efficiency attain their maxima, respectively. The thermal efficiencies and power output of the cycle are, in general, dependent not only on temperature and the thermal conductance between the working substance and the cylinder wall but also on the volume and other parameters, which are presented in this article. The system enabled reductions in piston velocity around the TDC region to a fraction of its value at constant crankshaft angular velocity typical in conventional engines. A QCV combustion has thus been successfully achieved, leading to improvements in engine fuel consumption and power output which are discussed in detail.

In this article was presented one approach for improvement of SI engine efficiency. Described concept has several advantages over ordinary SI engines. First of all, this engine is able to provide heat addition during constant volume, than it is possible to achieve smaller values of SI angle advance. Also with this concept there is no need for valve overlap. All of these mentioned advantages show that the potential to increase the efficiency of the SI engine conditions is not yet exhausted. As shown in the research results above, with the constant volume combustion cycle, the piston movement is significantly slower around TDC and BDC. Overall, the pressure integral of the QCV combustion cycle is about 4% higher than that of the conventional cycle at full load.

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Nomenclature

A	– area, [m^2]	c_P	– specific heat under constant pressure, [Jkg^{-1}K^{-1}]
CA	– crank angle, [deg.]	D	– cylinder diameter, [m]
C_V	– specific heat under constant volume, [Jkg^{-1}K^{-1}]	E	– energy, [J]

e – compression ratio, [–]
H – enthalpy, [J]
h – specific enthalpy, [Jkg^{-1}]
h_w – heat transfer coefficient, [$Wm^{-2}K^{-1}$]
m – mass, [kg]
n – engine speed, [rpm]
p – pressure, [Pa]
Q – heat, [J]
R – gas constant, [$Jmol^{-1}K^{-1}$]
R_D – connecting disc ratio, [m]
r – crankshaft radius, [m]
s – piston path, [m]
T – temperature, [K]
TFMEP– total friction mean effectic pressure, [bar]
t – time, [s]
u – specific energy, [Jkg^{-1}]
V – volume, [m^3]
W – work, [J]

Acronyms

BDC – bottom dead center
BMEP – brake mean effective pressure
CAI – controlled auto ignition
CDC – combustion dead center
FMEP – friction mean effectic pressure
HCCI – homogeneous charge compression ignition
IC – internal combustion
IMEP – indicated mean effective pressure
LHV – lower heating value
PMEP – pumping mean effectic pressure
QCV – quasi-constant volume
SI – spark ignition
TDC – top dead center

Greek symbols

α – angle of crankshaft [deg.]
η – efficiency [–]
λ_κ – ratio of crankshaft radius and connecting disc radius [–]
ω – angular speed [rad^{-1}]

Subscripts

b – burnt gas
c – combustion
g – gas
o – ordinary
u – unconventional
v – valve
w – wall

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