

EFFICIENCY OF A NEW INTERNAL COMBUSTION ENGINE CONCEPT WITH VARIABLE PISTON MOTION

by

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This paper presents simulation of working process in a new internal combustion engine concept. The main feature of this new internal combustion engine concept is the realization of variable movement of the piston. With this unconventional piston movement it is easy to provide variable compression ratio, variable displacement and combustion during constant volume. These advantages over standard piston mechanism are achieved through synthesis of the two pairs of non-circular gears. Presented mechanism is designed to obtain a specific motion law which provides better fuel consumption of internal combustion engines. For this paper Ricardo/WAVE software was used, which provides a fully integrated treatment of time-dependent fluid dynamics and thermodynamics by means of 1-D formulation. The results obtained herein include the efficiency characteristic of this new heat engine concept. The results show that combustion during constant volume, variable compression ratio and variable displacement have significant impact on improvement of fuel consumption.

Key words: *simulation, variable compression, variable displacement, constant volume combustion*

Introduction

The internal combustion (IC) engine is the favoured propulsion system for passenger and freight traffic. A significant reduction of CO₂ emission in mobility sector is a major challenge for the next years. Global concerns on the limitation of energy and reduction of the CO₂ emission force automotive engineers to develop more energy efficient and environmentally friendly alternative powertrain technologies. Considering the present development trends, trends for more efficient use of fuel resources and the well known problem of global warming and other environmental factors, development of IC engines will certainly move towards the reduction of fuel consumption. In this paper one of the possible ways of reducing thermodynamic losses in the IC engine is shown.

Relatively low efficiency of today's internal combustion engine is the consequence of several factors. First, ordinary spark ignition (SI) IC engines during running at low loads have their thermal efficiency reduced due to the effect of the throttle valve that controls the engine load and by the fact that the compression starts at low pressure [1]. Under part load conditions, engines use some of the work to pump air across the partially closed throttle valve. One of the possible solutions for improving efficiency at part load is to reduce the stroke volume by selectively shutting of several cylinders of an engine at the part load conditions. As early as 1916, the

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potential of using a variable displacement engine to increase the fuel efficiency at part load conditions was known and tested. This means that instead of reducing the air-fuel mixture charge by the throttle valve at part load conditions, the stroke volume of the engine is reduced by disabling some of the working cylinders [2]. Also, the compression ratio of the engine should be varied according to the load and speed conditions in order to improve efficiency [3-5].

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 [6], which consists of isentropic compression and expansion processes and constant volume heat addition and rejection processes [7, 8]. It is generally known that the most important parts of the cycle which determine the efficiency are the constant volume heat addition at high compression ratios [9, 10]. 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 nor compression ignition or even the modern developed homogeneous charge compression ignition or controlled auto ignition combustion processes, can achieve the efficiency level suggested by the ideal thermodynamic cycles [11]. Only the Otto cycle delivers theoretical maximum thermal efficiency. The traditional design of internal combustion engines using a simple slide-crank mechanism gives no time for a constant volume combustion which significantly reduces the cycle efficiency [11].

Variable displacement and variable compression engines are gaining attention by scientist and automobile manufactures because of their fuel consumption economy advantage. One of the successfully constructed IC engine with variable compression ratio is certainly made by SAAB [1]. In conventional IC engines the load regulation is balanced by throttling the intake mixture [6]. Variable displacement concepts have been analyzed in many different scientific publications. Siewart [12] reported a fuel economy approaching 20% for variable stroke engines over fixed stroke engines. Also there is a several patents about mechanisms which provides variable stroke, one of them are patented by Freudenstein and Maki [13]. Several authors [14-16] have proposed different complex mechanisms to achieve variable displacement engine. In the paper of Yamin and Dado [17] was investigated the effect of a variable stroke mechanism on the engine performance, the conclusion showed that the engine performance was improved with this novel design. Also Pouliot *et al.* [18], proposed, constructed and studied a five-cylinder, four-bar linkage engine and Filipi and Assanis [19] theoretically, investigated the effect of varying the stroke length on a homogeneous charge engine's combustion, heat transfer and efficiency using gasoline as fuel. Wong *et al.* [20] presented and analyzed a four cylinder engine with Alvar cycle that utilizes secondary pistons and auxiliary chambers.

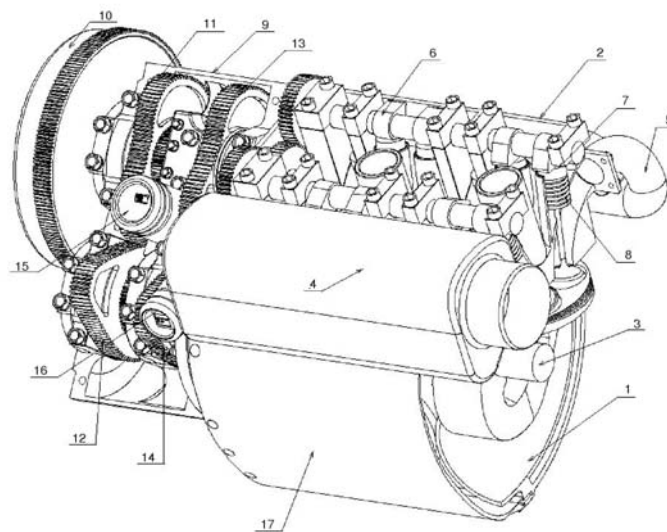
On the basis of these references a further step made in this paper is to make analysis of a new engine concept which is able to make variable piston motion. Variable piston motion (VPM) IC engine [21] is not only able to provide variable compression ratio and displacement but also with this concept it is easy to achieve dwell angle at top dead center (TDC) and bottom dead center (BDC). With piston dwell at bottom dead point more complete expansion can also be achieved. In this paper was used Ricardo/WAVE software to obtaining the improvement between this new cycle and the standard Otto cycle. Also in this paper was presented basic description of the new engine that will be able to realize thermodynamic cycle with increased efficiency.

Variable piston motion IC engine

In the following section will be presented basic parts and shape of a new IC engine concept. VPM IC engine is presented on the fig. 1. As can be seen from the described illustration toroidal piston make a movement conditioned by the mechanism consisting of two pairs of non-circular gears. 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 spark ignition engines.

Figure 1. Basic parts of VPM IC engine [21]

1 – engine block, 2 – engine head, 3 – toroidal piston, 4 – intake manifold, 5 – exhaust manifold, 6 – camshaft, 7 – valve, 8 – valve spring, 9 – housing, 10 – flywheel, 11, 12, 13, 14 – non-circular gears, 15, 16 – stepper motors, 17 – crankcase



VPM IC engine has two pairs of non-circular gears (NCG). A NCG is a special gear design with special characteristics and purpose. While a regular gear is optimized to transmit torque to another engaged member with minimum noise and wear and with maximum efficiency, a non-circular gear's main objective might be ratio variations, axle displacement oscillations and more. In fact this feature of NCG is very important for synthesis of mechanism where is intermittent-motion required. This intermittent-motion mechanism combines circular gears with non-circular gears in a planetary arrangement. With such planetary differential gear it is possible to achieve very complex movement, where toroidal piston is able to provide motion with variable displacement and variable compression, also because of the characteristics of NCG, piston dwell at TDC and BDC is also feasible.

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. 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. This process occurs within the engine and its part of the thermodynamic cycle of the

device. In all IC engine useful work is generated from the hot, gaseous products of combustion acting directly on moving surfaces of the engine, such as the top of a piston. This moving boundary of combustion chamber is the focus of this paper. In generally moving of the piston is responsible for the volume changing during process of combustion. In this paper was presented IC engine where this boundary, *i. e.* top of the piston, actually not moving in a large portion of heat addition.

The four stroke spark SI engine pressure-volume diagram (p - V) contains two main parts. They are the compression-combustion-expansion (high pressure loop) and the exhaust-intake (low pressure or gas exchange loop) parts. The main reason for efficiency decrease at part load conditions for these types of engines is the flow restriction at the cross-sectional area of the intake system by partially closing the throttle valve, which leads to increased pumping losses and to increased low pressure loop area on the P - V diagram. Meanwhile, the poorer combustion quality, *i. e.* lower combustion speed and cycle to cycle variations, additionally influence these pressure loop areas, illustrated in detail on fig. 2.

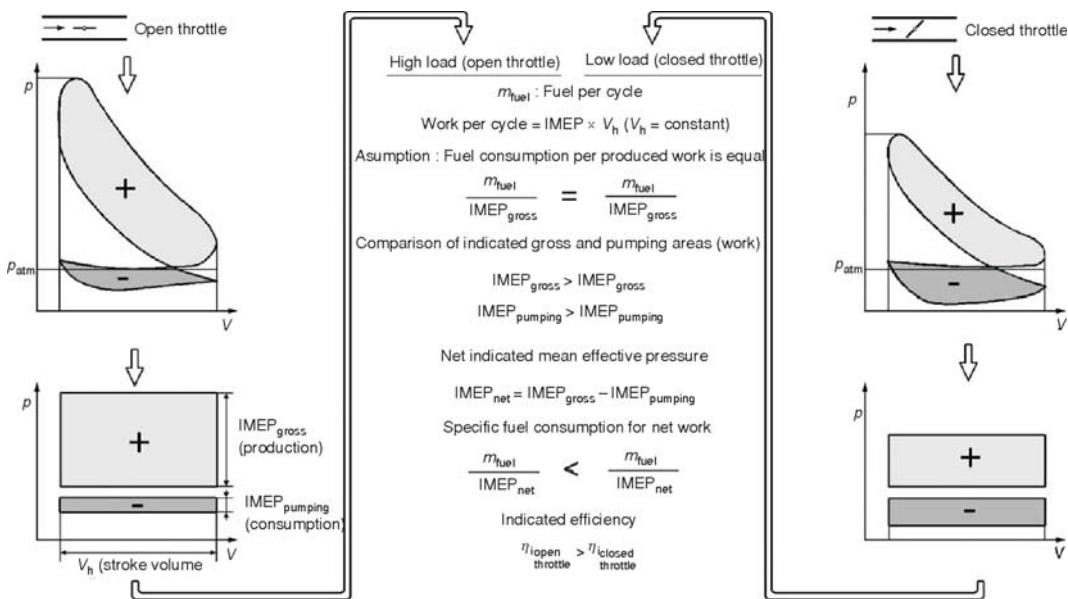


Figure 2. Schematic comparison of gross, pumping, net IMEP and their effect on indicated efficiency in high and low load conditions in SI engines [2]

Cylinder deactivation is initialized by cutting off the fuel supply to the selected cylinders. There are also several systems that shut off the valves of the deactivated cylinders too. In these systems, the reduction in pumping losses is more than that achieved by cutting off the fuel supply only [22]. In this study, methods for increasing efficiency at part load conditions and their potential for practical use are also investigated, in fact in this article was examined case where classical approach of engine throttling was replaced with variable displacement piston motion. In fig. 3 is presented piston motion law that was used for simulation of working processes in variable piston motion IC engine.

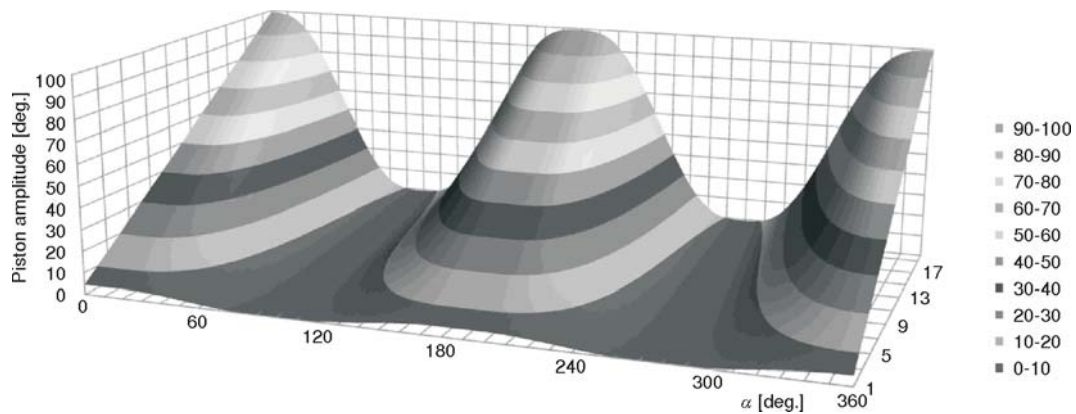


Figure 3. Complex motion of the toroidal piston that allow infinitely many displacement of the engine in the range from 678 cm^3 to 4000 cm^3

Unconventional piston motion-new four stroke cycle

The ideal scenario is to initiate and complete the combustion event while the piston remains at the TDC position. This provides the maximum thermal potential and eliminates the negative work due to early ignition which is well into compression stroke with conventional engine strategies. In addition, 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. During operation of conventional IC engines, the piston can only reciprocate continuously between TDC and 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.

In this section, the new unconventional piston motion law will be presented. 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 at the TDC and the BDC, while the output shaft continues to rotate for up to 35 degrees. Adding these constant volume dwell cycles improves fuel burn, maximizes pressure, and increases cylinder charge. Fuel burn can be precisely controlled by maintaining a minimum volume (TDC piston dwell) throughout the burn process, containment maximizes pressure and burn efficiency. Furthermore, holding the piston at maximum volume (BDC piston dwell) provides additional time for the cylinder to fully charge before closing the intake valves. The design creates unconventional four stroke cycle process. This unconventional cycle consists of the following strokes and processes.

The first stroke consists of forced and free intake. During the forced intake, piston travels from TDC to BDC, which draws fresh mixture into the cylinder. This part of the stroke is the same as the intake stroke in the ordinary IC engines, the second part is the free intake. After

the piston comes into BDC, it stops there for a while, this dwell time depends on the optimization of the intake process and it will not be explained in detail in this paper. However, it is very important that the piston dwell does not last longer or shorter than the optimal calculated value. After the piston comes into BDC, the column of fresh gases continues to flow into the cylinder by inertia, until the intake valve closes. In this way the intake volumetric efficiency is increased. The second stroke consists of the compression process and a combustion during constant volume. In the first part of this second stroke, the piston travels from BDC to TDC. The ignition occurs at TDC without any spark advance, thus saving the accumulated energy of the flywheel. Ignition begins when the piston is stopped at the TDC, while the piston stop lasts for the time calculated by optimization to complete combustion and prevent any back-pressure caused by the spark advance. Consequently, the use of energy obtained from the fuel is maximized and the fuel consumption is decreased. The third stroke is an expansion stroke, during which the piston comes from TDC to BDC like in a standard mechanism but with the exception that piston again makes a dwell in BDC. In this new unconventional four stroke cycle, the entire expansion stroke occurs between TDC and BDC. Compared to standard IC engine, in the new piston motion movement there is no exhaust valve opening advance, which determines loss of possibly resulting work. In the second part of this third stroke, the piston comes on BDC and stays in the same position for a while. During this time high-pressure gases are spontaneously evacuated, while

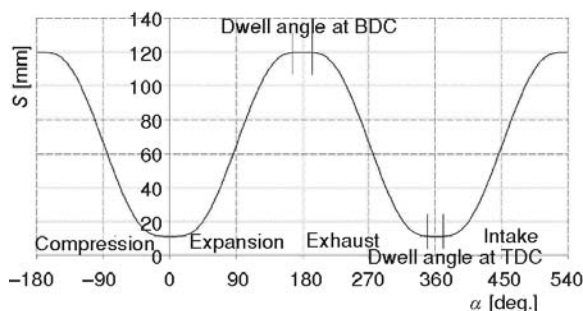


Figure 4. New unconventional motion of piston for the selected dwell angle of 20° [23]

the piston is stopped at the BDC. The last stroke is exhaust stroke, during which the exhaust gas is actually a low pressure gas, so the piston will not require a big pumping effort going up towards TDC. In the last phase of exhaust stroke, exhaust gases can freely leave compression volume. At the same time intake valves slowly open and fresh charge comes into the cylinder, while the piston is still in the dwell mode at TDC. Previously described unconventional four stroke cycle can be illustrated by fig. 4.

Simulation

Within the automotive industry the most widely adopted technique for gas exchange studies is to solve the 1-D coupled set of non-linear equations using the finite volume or finite difference method. This technique is used in several commercial softwares *e. g.*, Ricardo/WAVE, GT-Power and AVL/BOOST. In this paper, Ricardo/WAVE software was used, which provides a fully integrated treatment of time-dependent fluid dynamics and thermodynamics by means of 1-D formulation. Internal combustion engine simulation modeling has long been established as an effective tool for studying engine performance and contributing to evaluation and new developments [24, 25]. Thermodynamic models of the real engine cycle have served as effective tools for complete analysis of engine performance and sensitivity to various operating factors [26, 27]. WAVE is the primary program and solver for all simulations of fluid dynamic systems, this software can be used to model the complete internal combustion engine. The piping and manifolds of the intake and exhaust systems are modeled using the basic WAVE flow elements. These networks are then linked together through engine elements and

sub-models, which have been calibrated to provide accurate driving inputs for the intake and exhaust pressure-wave dynamics.

The details of the flow (as calculated in the flow network) are obtained as a solution of quasi-one dimensional compressible flow equations governing the conservation of mass, momentum and energy-eq. (1-3). The flow network of both conventional and unconventional piston movement is discretized into a series of small volumes and the governing equations are then written in a finite difference form for each of these elementary volumes. A staggered mesh system is used, with equations of mass and energy solved for each volume and the momentum equation solved for each boundary between volumes. The equations are written in an explicitly conservative form as:

– mass continuity equation

$$\frac{dm}{dt} = \sum_{boundaries} m_{flux} \quad (1)$$

– conservation of momentum equation

$$\frac{d(m_{flux})}{dt} = \frac{dPA + \sum_{boundaries} (m_{flux} u)}{dx} - \frac{4C_f \frac{\rho u^2 dx A}{2D} - C_p \left(\frac{1}{2} \rho u^2 \right) A}{dx} \quad (2)$$

– conservation of energy equation

$$\frac{d(me)}{dt} = P \frac{dV}{dt} + \sum_{boundaries} m_{flux} H - h_g A (T_{gas} - T_{wall}) \quad (3)$$

If the engine cylinder element has one zone, the entire cylinder is treated as one region. In the latter, the cylinder is divided into two regions (unburned and burned), which share a common pressure. The two-zone model is used to capture the chemical processes taking place during the combustion period in more detail. Combustion models may be used either with a single or two-zone engine cylinders, but for this research two-zone models were used because of the problem with knock combustion that was also examined. For the single zone model there is the energy equation refer to (4) as below:

$$\Delta(mu) = \sum_{i=1}^{nvalves} m_i h_i - Q - P\Delta V \quad (4)$$

During combustion, the only term of enthalpy flow is $m_i h_i$ due to propagation of the flame front to the unburned zone. For the two-zone, refer to model (4), in the unburned zone we have:

$$m_{u1} u_{u1} - m_{u0} u_{u0} + P(V_{u1} - V_{u0}) + Q_u - \Delta m_{ui} h_{ui} = 0 \quad (5)$$

Using the equation of the state, it becomes:

$$m_{u1} u_{u1} - m_{u0} u_{u0} + m_{u1} R_{u1} T_{u1} - P V_{u0} + Q_u - \Delta m_{ui} h_{ui} = 0 \quad (6)$$

Similarly, for the burned zone we have:

$$m_{b1} u_{b1} - m_{b0} u_{b0} + m_{b1} R_{b1} T_{b1} - P V_{b0} + Q_b - \Delta m_{bi} h_{bi} = 0 \quad (7)$$

As a constraint, the volumes of the unburned and burned zones are summed up to the total cylinder volume:

$$m_{u1} R_{u1} T_{u1} + m_{b1} R_{b1} T_{b1} - P V_c = 0 \quad (8)$$

The last three equations are a complete set and are solved by using the Newton iteration method.

Since this article investigates unconventional piston motion, classical approach for solving problems of volume changes cannot be applied. When the piston position differs from standard crank piston motion, the imposed piston motion sub-model can be used for modeling the engine. The formulation to calculate the instantaneous cylinder volume is identical to the one used in the standard WAVE model, with the exception that the piston position, is linearly interpolated between points in the user-entered profile. Smooth piston motion depends on the fine spacing of the crank angle array. In this case enough large arrays were used to enable one-degree spacing. As far as the high-pressure part of the cycle is considered, the most important process is the combustion. Without in-cylinder pressure measurements, the combustion model had to be predicted based on typical forced induction Wiebe function parameters. WAVE allows for three parameters in the Wiebe correlation to be input: 10-90% burn duration, 50% burn point, and the Wiebe exponent, described by eq. (9). In this program, Ricardo Wave model of combustion can be selected between several options, ranging from theoretical models with constant volume or constant pressure heat release, over Wiebe-function based heat release models, to quasi-dimensional two-zone model of turbulent flame propagation. The SI Wiebe function is widely used to describe the rate of fuel mass burned in thermodynamic calculations [28]:

$$W = 1 - e^{\left[-AWI \left(\frac{\Delta\theta}{BDUR} \right)^{WEXP+1} \right]} \quad (9)$$

Table 1. Main engine data

Engine type	Spark ignition
Engine cycle	Four-stroke
Number of cylinders	2
Number of valves per cylinder	4
Bore	120 mm
Stroke	30-177 mm
Intake valve diameter	44 mm
Exhaust valve diameter	40 mm
Valves path	15 mm
EVDUR	235 deg.
IVDUR	230 deg.
EVMP	253.3-245 deg.
IVMP	479.3-471 deg.
Octane number	98
Compression ratios	8-16

This relationship allows the independent input of function shape parameters and of burn duration. The experimentally observed trends of premixed SI combustion are represented quite well. In this paper the Wiebe one stage model of heat release has been chosen. The parameters of Wiebe function were selected to achieve good agreement between modeled and experimentally recorded pressure. Selected parameters have been successfully applied in the research [29-31]. Engine data that was chosen for this research was presented in tab. 1. It can be noticed that valves open duration are constant values, but position of maximum valve opening (EVMP and IVMP) are in certain ranges. That is because of variability of piston motion, mechanism is constructed in that way that allow different piston displacement and in the same time adjustment of valvetrain open phase.

The valve train was modeled by setting up the appropriate number of valves per cylinder and entering details about valve size, lift, and flow, for this purpose was chosen values which are different from the conventional valvetrain. Reason for that can be found in the fact that piston dwell have impact on valves open duration. So, in this concept, because of the piston dwell there is no need for valve overlap, this can be seen from fig. 5. Valve data for each cylinder must be entered referencing a valve model. The lift valve model was used in this example, so that the valve would follow a set profile. The intake and exhaust valves were modeled using

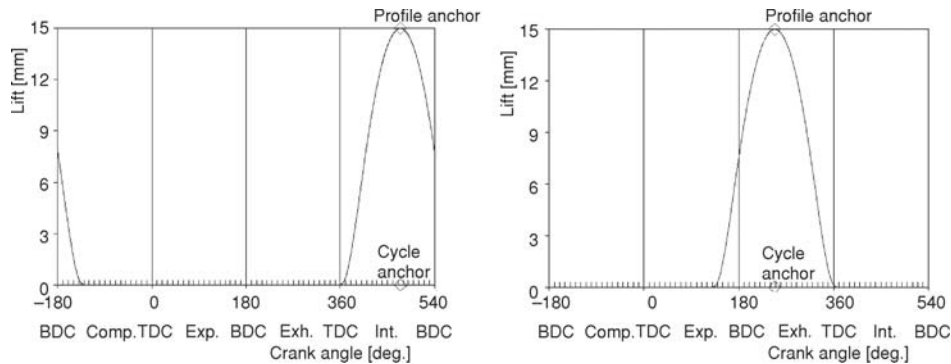


Figure 5. Valves lift without valve overlap for intake and exhaust valve respectively

ducts and junctions, where geometry such as length, orientation, and cross sections are specified. Heat transfer and friction data must also be entered, in this model the selected values are similar to the standard SI engine.

Since in this paper was investigated only virtual engine model, for the purpose of model calibration in this study was examined influence of selected input parameters for simulation of ordinary IC engine. The calibration of simulating model was performed on ordinary spark ignition engine on a test stand with adequate experimental equipment. It was realized through the comparison of experimental and calculated results and tuning some model parameters and constants. Following the procedure prescribed in the WAVE user manual the average values of all important values was compared to test data. In order to validate the model with high degree of precision, it is important to have as much engine test data as possible. For this research model was calibrate to match experimental data for 50 different operating conditions at full and partial load. In order to validate the parameters calculated by Ricardo/WAVE software, engine data was recorded at a range of engine speeds between 2000 and 6000 rpm. The pressure histories were recorded in first engine cylinder and in two characteristic points in inlet pipe of relating cylinder and compared with calculated curves. TDC must be determined within 0.1 deg. in order to accurately calculate work (IMEP), so in order to avoid serious error in the TDC determination caused by torsional vibration the test cylinder must be chosen in multi-cylinder engine as the one immediately next to the crankshaft encoder. Piezoelectric pressure transducer was used for the purpose of acquiring in-cylinder pressure data. For this experimental investigation was used a special category of ECU (engine control unit) which is programmable in order to achieve different working parameters (air-fuel ratio, ignition timing, fuel injection, etc.).

The calibration of simulating model was performed and some results are described on fig. 6. For this purpose the overall engine operation parameters were considered: volumetric efficiency, power and torque output, mixture strength, fuel consumption, engine mechanical losses and flow losses in engine intake and exhaust systems, results of torque and power are shown on fig. 6. Since in this engine concept

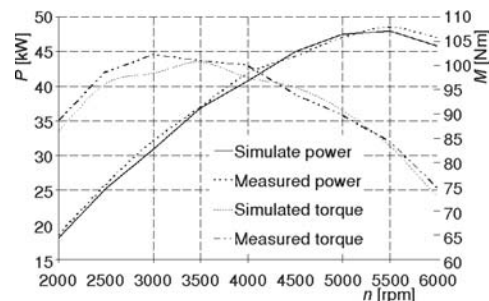


Figure 6. Comparison of experimental and modelled engine parameters for ordinary SI engine

there are several pairs of gears these mechanical losses must also be taken into account. 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. With eq. (10) it is easy to calibrate all necessary losses by changing constants (A_{cf} , B_{cf} , C_{cf} , and Q_{cf}) in order to simulate all mechanical losses that would exist in the virtual engine model.

$$FMEP = A_{cf} + \sum_{i=1}^{ncyl} [B_{cf} (P_{max})_i + C_{cf} (S_{fact})_i + Q_{cf} (S_{fact})_i^2] \quad (10)$$

Results and analysis

One of the major features of the described engine is combustion during constant volume. It can be concluded from the results in fig. 7 that there is a noticeable differences between heat addition part of P - V diagram in classical and new concept. On the same figure is presented P - V diagrams in linear and log-log graphs, log-log graph were selected because of better view on gas exchanges loop.

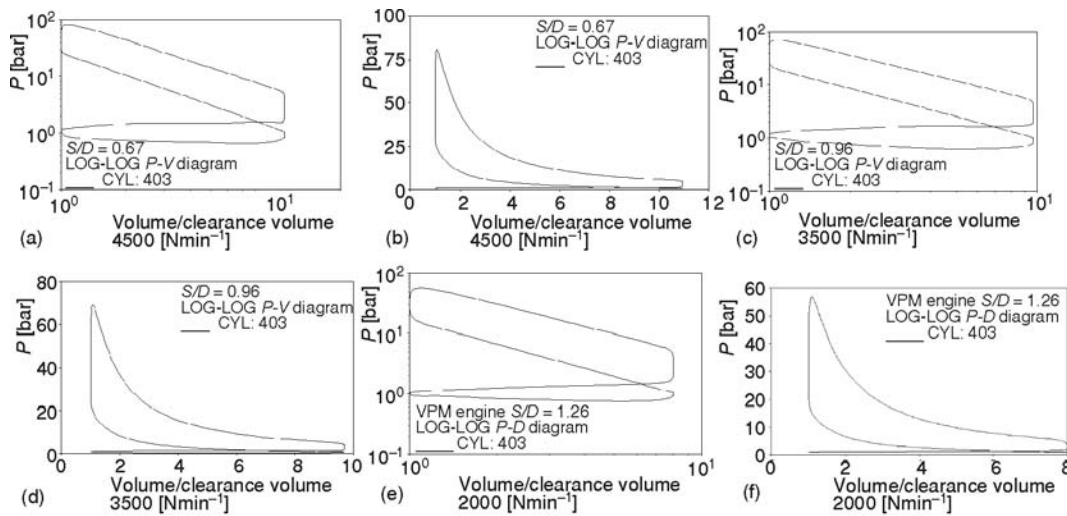


Figure 7. Pressure-volume changes in linear and log-log digrams for the different values of S/D ratio and engine speed

Impact of piston dwell on P - V diagram, especially on heat addition part, is shown on fig. 8. In the previous fig. 9 variations of efficiency is shown for various values of S/D ratio, engine speed and compression ratio. To gain into the efficiency at different load and S/D ratio it is important to activate knock model which is based on induction time and calculate in seconds ignition delay at every timestep using the following eq. (11):

$$\tau = \frac{0.01869}{A_p} \left(\frac{ON}{100} \right)^{3.4107} P^{-1.7} \exp \left(\frac{3800}{T} \right) \quad (11)$$

where: A is the pre-exponential multiplier, ON – the fuel octane number, P – the cylinder pressure, At – the activation temperature multiplier, and T – the unburned gas temperature. In general, this induction time continually decreases as combustion progresses and the unburned zone temperature rises. The end-gas auto-ignites (knocks) if the induction time is less than the flame arrival time. When knock occurs, a spontaneous mass burning rate due to knock is determined and fed

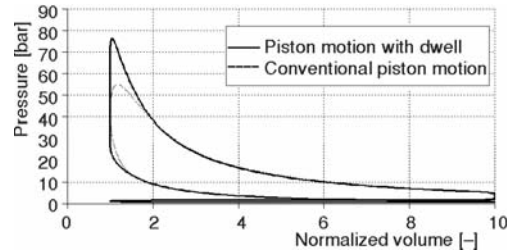


Figure 8. Comparison of P - V diagrams during conventional motion and motion with dwell

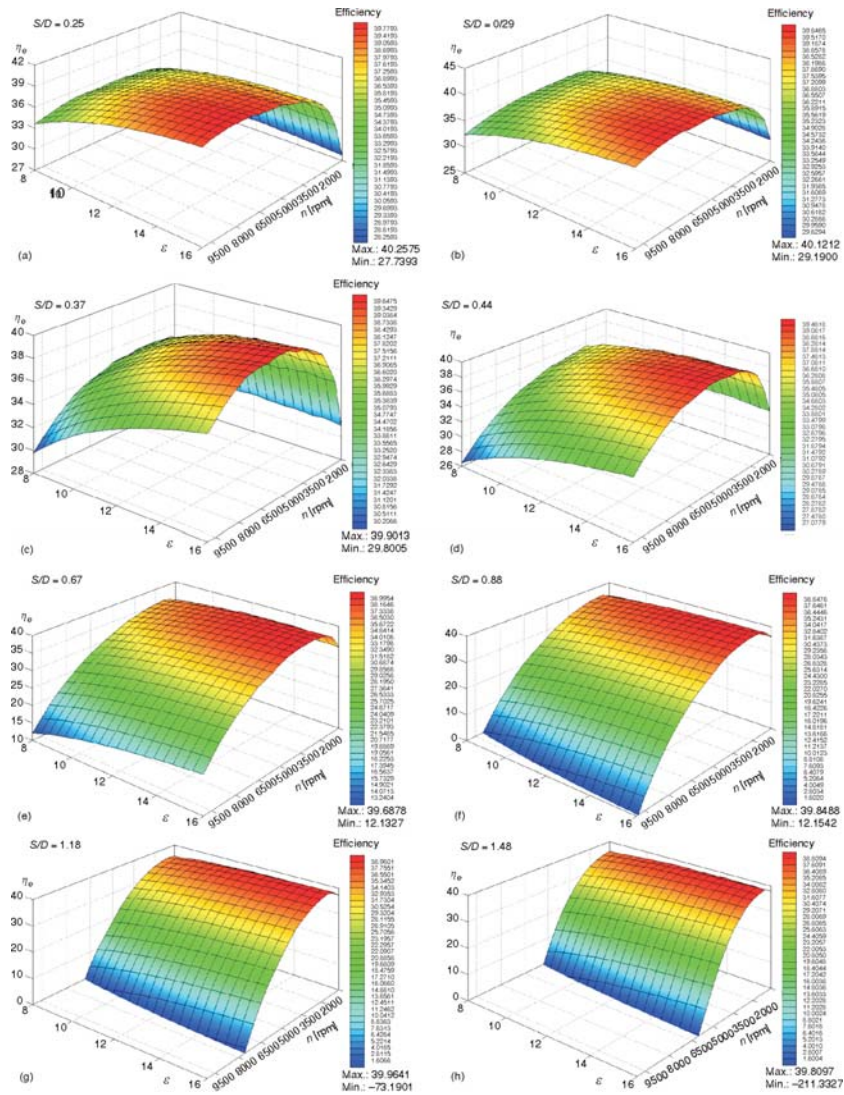


Figure 9. Efficiency flow of VPM IC engine in relation to S/D , compression ratio and engine speed (for color image see journal web site)

back to the cylinder, leading to rapid rise in cylinder pressure and temperature. The in-cylinder heat transfer coefficient is also increased during knock. The model assumes that auto-ignition occurs when eq. (12) is satisfied:

$$\int_{t_0}^{t_i} \frac{dt}{t} = 1 \tag{12}$$

In the eq. (12) are mentioned following parameters: t_0 – start of end-gas compression, t_i – the time of auto-ignition, and t – the induction time. After solving all necessary simulation cases, efficiency curves for all examined S/D ratios can be drawn, such graph is presented on the following fig. 10.

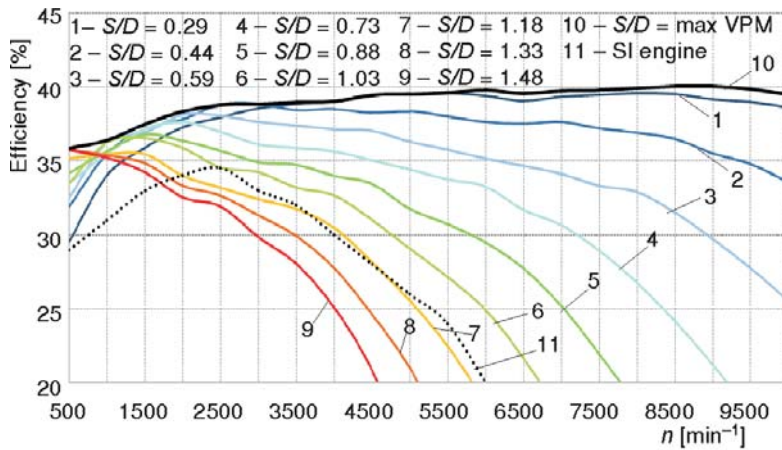


Figure 10. Efficiency curves for different S/D ratio in relation to engine speed in comparison with efficiency of ordinary spark ignition engine

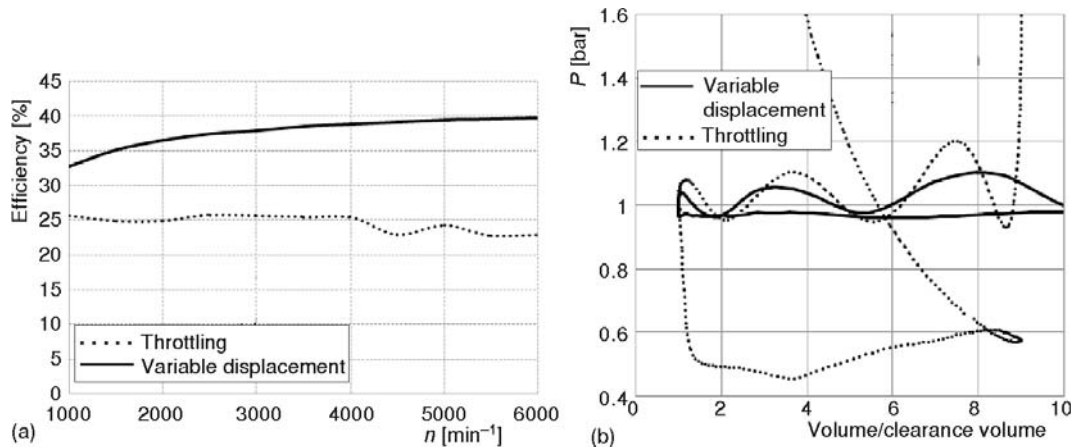


Figure 11. Comparisons of: (a) efficiency during conventional regulation of load and with VPM IC engine, (b) cylinder pressure diagrams for these two approaches (gas exchanges loop only)

It is interesting to see the impact of variable displacement on efficiency in relation to conventional throttling operation mode, such analysis was performed and results can be seen from fig. 11.

In conventional engine during exploitation only two parameters can be changed, load and speed. Unlike conventional engines in VPM engine there is one more parameter that can be changed during operation-stroke. In fig. 12 are presented changes of in-cylinder pressure during operation at constant speed and constant full load but with variable stroke (variable displacement).

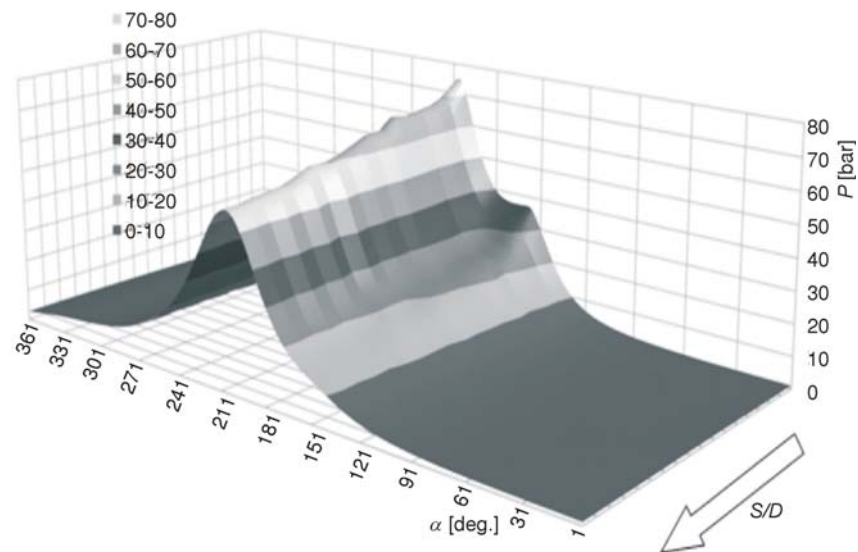


Figure 12. In-cylinder pressure changes in relation to crank angle and engine displacement at constant engine speed

Conclusions

In this article was presented one approach for improvement of spark ignition engine efficiency. Described concept has several advantages over ordinary SI engines. First of all, this engine have variable compression ratio, than with this concept it is possible to avoid classical approach for partial load operation via variable displacement. Finally presented concept is able to provide heat addition during constant volume. 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, variable displacement methods have the best potential to increase the efficiency of the engine at part load conditions. To avoid engine operation below the unthrottled load limit, facilitate smooth mode changes and further improve the vehicle fuel economy. With the constant volume combustion cycle, the piston movement is significantly slower around TDC and BDC, in fact piston actually stops for a while, this have significant impact on volumetric efficiency and engine efficiency. Overall, the pressure integral of the constant volume combustion cycle is about 11% higher than that of the conventional cycle at full load, but with the feature of variable displacement this concept can reach almost 80% greater efficiency in relation to standard engine at part load. An advanced engine system design, combining variable displacement, variable compression and constant volume combustion has been explored with the aid of the physics-based computer simulation. The main objective was to develop a system capable of operating unthrottled throughout the torque-speed range. Regulat-

ing the load via reduced displacement while keeping the throttle wide open produces very significant efficiency gains at low-load, but there is some sort of limit. The minimal engine displacement is about 680 cm³ and the maximal around 4000 cm³, so for the really low load the throttling would still be necessary. However, even at such low loads and low displacement there would be an improvement in fuel consumption because engine throttling would not be so drastic like in cases when average engine operate.

Acknowledgment

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Nomenclature

A	– flow area, [m ²]
A_p	– pre-exponential multiplier, [–]
A_t	– activation temperature multiplier, [–]
C_f	– friction coefficient, [–]
C_p	– pressure loss coefficient, [–]
D	– cylinder diameter, [m]
e	– energy, [J]
H	– enthalpy, [J]
h	– specific enthalpy, [Jkg ⁻¹]
h_g	– heat transfer coefficient, [Wm ⁻² K ⁻¹]
m	– mass, [kg]
n	– engine speed, [rpm]
P	– pressure, [Pa]
Q	– heat, [J]
R	– crankshaft radius, [m]
S	– piston path, [m]
T	– temperature, [K]
t	– time, [s]
t_0	– start of end gas compression, [s]
t_1	– time of auto-ignition, [s]
u	– specific internal energy, [Jkg ⁻¹]
V	– volume, [m ³]
V_h	– displacement, [m ³]
x	– co-ordinate, [m]

Greek symbols

α	– angle of crankshaft, [deg]
Δ	– difference, [–]
ε	– compression ratio, [–]

η	– efficiency, [–]
θ	– degrees past start of combustion, [deg.]
ρ	– density, [kgm ⁻³]

Subscripts

b	– burnt gas
c	– chamber
e	– engine
i	– indicated
u	– unburnt gas

Acronyms

AWI	– internally calculated parameter to allow BDUR to cover the range of 10-90%
BDC	– bottom dead center
BDUR	– combustion duration
EVDUR	– exhaust valves open duration
EVMP	– exhaust valve maximum open point
FMEP	– friction mean effective pressure
IC	– internal combustion
IMEP	– indicated mean effective pressure
IVDUR	– inlet valves open duration
IVMP	– inlet valve maximum open point
NCG	– non-circular gear
ON	– fuel octane number
SI	– spark ignition
TDC	– top dead center
VPM	– variable piston motion
WEXP	– exponent in Wiebe function

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