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SPARK IGNITION ENGINE PART LOAD FUEL ECONOMY IMPROVEMENT: NUMERICAL CONSIDERATION

The possibilities that enable the improvement of a spark ignition engine fuel economy at low load conditions has been analysed: decrease of engine intake losses by intake valve lift and timing fully control and low load operation without throttling; compression ratio increase (i.e. the application of variable compression ratio) and the operation with extremely lean mixture. The analysis has been performed numerically, using engine cycle computer simulation by a very sophisticated program AVL-Boost, applied on the model of 1.4 *litters spark ignition engine (Yugo Florida). The above-mentioned possibilities has been quantified and compared and the combination of these measures has also been considered.*

Key words: *spark ignition engine, fuel economy, variable valve timing.*

1. INTRODUCTION

New concepts of spark ignition (SI) engines are very actual nowadays. One of the reasons is too high fuel consumption of SI engine at part load comparing with diesel engine. The drawbacks of SI engines in regard with relatively high fuel consumption are well known. They are: knock limited compression ratio, unstable lean mixture running at part load and engine throttling at part load which increases pumping losses.

Theoretically, there are three technical possibilities for the improvement of SI engine part load fuel economy. The first is the design of variable compression engine. Since the value of SI engine compression ratio is limited by knock at full load, it is reasonable to use the high compression ratio at part load and moderate one at full load. Unfortunately, the practical realization of this possibility is very difficult, though there are some trials and prototype solutions [1].

The second possibility is ultra lean mixture running of SI engine. The operation of SI engine with lean homogeneous mixture is limited by irregular running in the vicinity of lean mixture limit. However, the overall lean mixture limit can be high enlarged by stratified charge combustion. Though the idea of stratified charge is known for a couple of decades ago, its practical realization occurred recently. The realization of this solution is enabled by the use of gasoline direct injection with precise electronically controlled mixture. Using appropriate airflow in the combustion chamber (the tumble) and late injection timing, it is possible to obtain relatively rich mixture near spark plug with overall lean mixture at low load engine running. At the same time, using early injection (during the intake) with homogeneous near stoihiometric mixture at middle and full load operation, it is possible to keep all good advantages of conventional SI engines, i.e. higher specific output. This concept is recently relatively successfully applied in standard SI engine production [2]. At the moment, it is useful to note that lean mixture running concept has in itself double potential for fuel economy improvement: lean mixture has better thermodynamic properties and at the same time enables reduction of intake throttling and, therefore, the decrease of pumping losses.

Third possibility of SI engine part load fuel economy improvement is the reduction of pumping losses (exhaust and induction flow losses) through valve timing control. Nowadays, leading vehicle car manufacturers use different designs of variable valve timing (VVT). The most frequently used solution is based on so called “cam phaser”, device that enables relative angular movement of intake camshaft. Therefore, late closure of intake valve at high engine speed can be obtained. This approach is relatively simple but results are also very moderate. The systems with variable valve timing and variable valve lift (VVTL) are more complex but also much more promising. However, the system with discrete change of valve timing and lift, usually with two different cams, still uses throttling for engine power control. Therefore, pumping losses at part load are still very high reducing fuel economy.

Throttleless solution, as BMW “Valvetronic” design [3], can realize continual change of valve opening duration and lift which enables engine power control with reduced intake pumping losses. For the future, much more promising solution is camless system with electromagnetic control of valve opening and lift [4]. This system enables absolute flexibility of valve timing and lift optimisation using electronic control of valve opening with big reduction of pumping losses.

This paper tries to evaluate and quantify the effect of aforementioned solutions on the possibility of SI engine

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part load fuel consumption reduction. This evaluation is done numerically using computer simulation of engine process with the application of different method of engine output control.

2. COMUTER SIMULATION OF ENGINE CYCLE

The analysis of the potentials of aforementioned measures on fuel economy at part load engine conditions has been carried out by engine cycle mathematical simulation. For this purpose the AVL BOOST program for engine working process simulation has been used [5].

Program code BOOST contains so called ”pre-processor" which enables the forming of engine calculating scheme, selection of model type for processes and sub processes, specification of all parameters and starting values and boundary conditions etc. The calculating scheme of the engine is formed graphically using offered standard elements such as: cylinders, receivers, pipes, pipe conjunctions, cleaners, injectors etc. For all elements must be specified their geometrical and other characteristics according to real engine conditions, and they are assembled to so called ”engine calculation scheme". Fig.1 shows the calculating scheme of the relating engine.

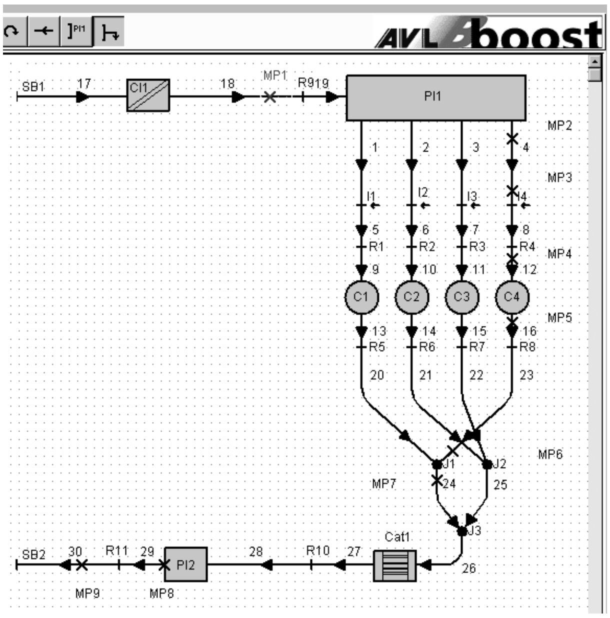

Figure 1. The scheme of calculation model.C-engine cylinder; Cl - air cleaner; Pl-receiver; SB - system boundary; 1 - 30 pipes; MP - measuring (control) point in pipe; I - Injector; R - restriction point; J-junction; Cat-catalyst.

Since the main aim of this mathematical simulation is the analysis of engine fresh mixture filling process and load control, a special care has been taken to the low-pressure part of engine cycle, i.e. gas exchange process. In program BOOST the flow through the pipes (intake and exhaust) has been modelled according to one-dimensional gas dynamics including wall friction forces and wall heat transfer. Pressure wave formation, one-dimensional propagation through the pipes by sound velocity, and reflection has been taken under account [5].

Table 1. Main engine data

swept volume (l)	1.372
bore/stroke (mm/mm)	80.5/67.4
compression ratio (-)	9.2
inlet valve diameter (mm)	36.5
inlet valve opens (CA deg. before BDC)	7
inlet valve closes (CA deg. after TDC)	35
exhaust valve sit diameter (mm)	30
exh. valve opens (CA deg. before TDC)	37
exh.valve closes (CA deg. after BDC)	5
fuelling system	Bosch M4.6

The flow through intake and exhaust valves is modeled as isentropic flow through the restrictions including real valve lift and timing. The flow coefficients for the valves had been previously determined using usual test procedure of stationary valve blowing [6].

Conventional engine load control, i.e. “throttle” has been simulated by R9 (flow restriction No. 9), simply by adjusting the level of flow restriction to obtain the desired engine output. Load control using fully variable valve timing has been simulated by adapting intake valve timing so to obtain the same engine output without any throttling. In the cases of analysis the effects of variable compression ratio and lean mixture operation on fuel economy, the necessary throttling for desired engine output has also been adjusted.

As far as the high-pressure part of the cycle is considered, the most important process is combustion. In program BOOST model of combustion can be selected between several options, from theoretical models with constant volume or constant pressure heat release, over Viebe-function based heat release models, to quasidimensional two-zone model of turbulent flame entraintement and propagation. In this investigation a Viebe one stage model of heat release has been chosen. The parameters of Viebe function were selected to achieve good agreement between modelled and experimentally recorded pressure lines taken earlier at the same engine [7].

The object of investigation is 4-stroke, 4-cylinder in line, 2 valve per cylinder, liquid cooled spark ignition engine, with MPI fueling system, produced by DMB and used to power Zastava-Florida 1.4 passenger car. The main engine data are shown in table 1.

The calibration of simulating model was performed earlier. It was realized through the comparison of experimental and calculated results and tuning some model parameters and constants. 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 (Fig.2). Also, the pressure histories were recorded in fourth engine cylinder and in two characteristic points in inlet pipe of relating cylinder and compared with calculated curves [7].

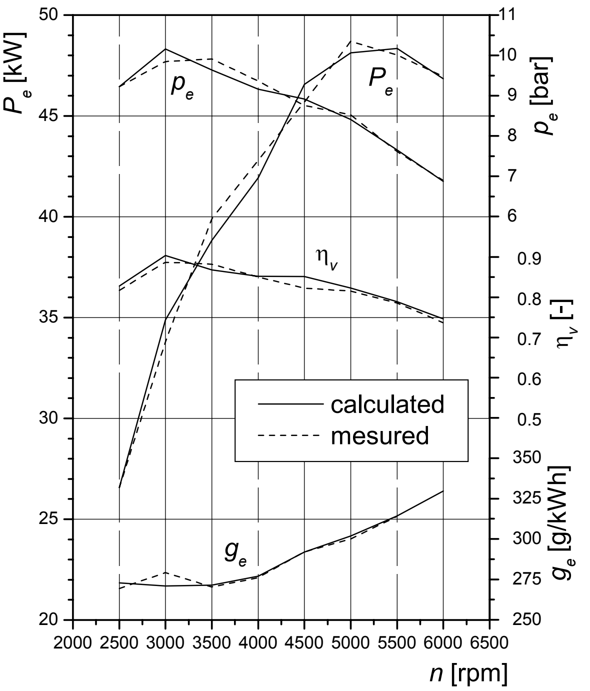

Figure 2. Comparison of experimental and modelled overall engine parameters: P_e -power output, p_e -mean effective pressure, g_e -specific effective fuel consumption, η_v -volumetric efficiency.

3. RESULTS OF ENGINE CYCLE SIMULATION

Using described mathematical model based on AVL BOOST code, different possibilities of engine control are analysed. All simulations have been carried out at the same working point which was considered as a representative for afore mentioned engine, i.e. for engine power P_e=12 kW at 3000 rpm (approximately one third of full load at this engine speed) which corresponds to the vehicle speed of about 80 km/h.

a) Effects of variable valve timing and lift

The main point in the cycle simulation was the possibility of intake pumping losses reduction using engine load control by variable intake valve timing and lift without additional throttling and the comparison with conventional engine with power control by throttle. Standard engine compression ratio ε=9.2 was used during this simulation, as well as stoihiometric mixture λ=1.0.

Fig. 3 shows the cylinder pressure during exhaust and intake, as well as the effective flow area of intake valve for the cases with the load control by throttling and by valve timing control (without throttling) at the same engine load. Such intake valve control can be easily realized by the application of electromagnetic valve opening. These two cases are marked at the figure by "throttle" and "EFVVT" (electrically fully variable valve timing). The difference in the cylinder intake pressure is evident.

However, this difference for aforementioned two cases is more evident in "p-v diagram" shown at Fig. 4, where low-pressure part of engine cycle is drawn for both cases. The figure shows, also the differences in pumping losses and their reduction by EFVVT control. Generally speaking, the applied strategy is as follows: at the beginning of induction intake valve opens as quickly

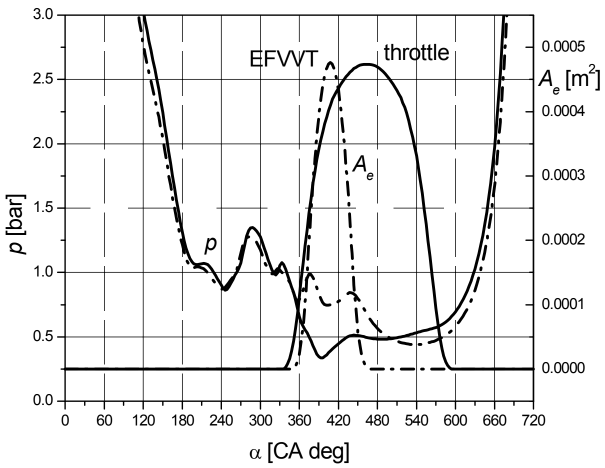

Figure 3. Cylinder pressure (p) and intake valve flow area (A_e) vs. crank angle with load control by throttle and variable valve timing (EFVVT)

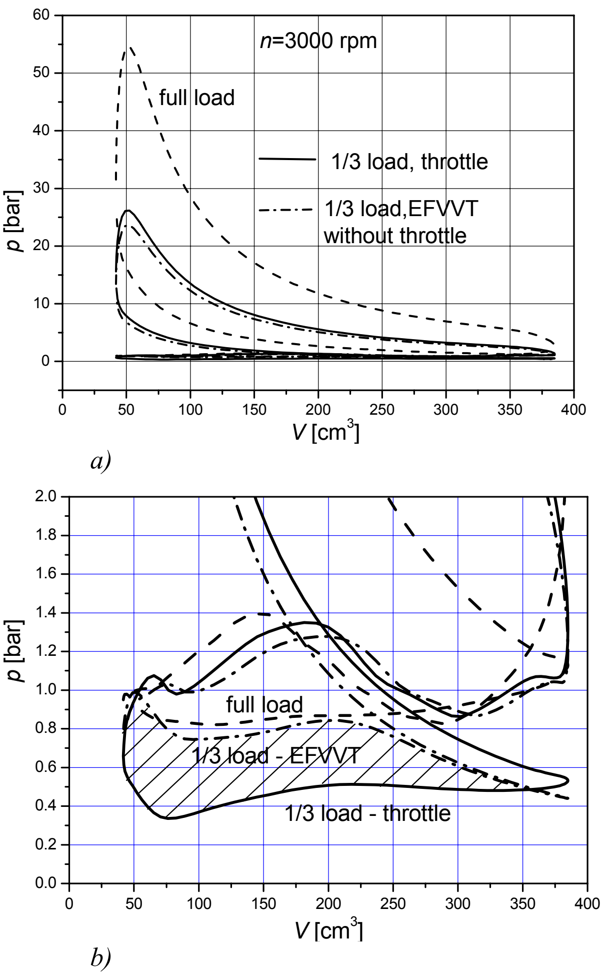

Figure 4. Cyliner pressure diagrams for full load and part load controlled by throttle and EFVVT (a) and their low pressure parts (b)

as possible reaching full lift. This gives small pressure loss. Intake valve closes when engine admits necessary quantity of mixture for this load, what happens almost at the half of intake piston stroke. Cylinder pressure rapidly decreases after intake valve closure, but this negative work is fully regenerated at the beginning of compression stroke. The hatched area at Fig. 4 (difference between EFVVT and throttle) represents the reduction of work required for the induction of appropriate quantity of mixture. Calculation of complete cycle data shows 5 % improvement of effective specific fuel consumption, i.e. fuel economy.

If, instead of throttling, the induction control is done by the decrease of intake valve lift, with corresponding decrease of valve opening angle (with the same angular position of maximum valve lift), no reduction of pumping losses can be achieved.

This case is shown at Fig. 5 and is indicated as MVVL (mechanically variable valve lift). Fuel consumption would be even increased for about 3 %.

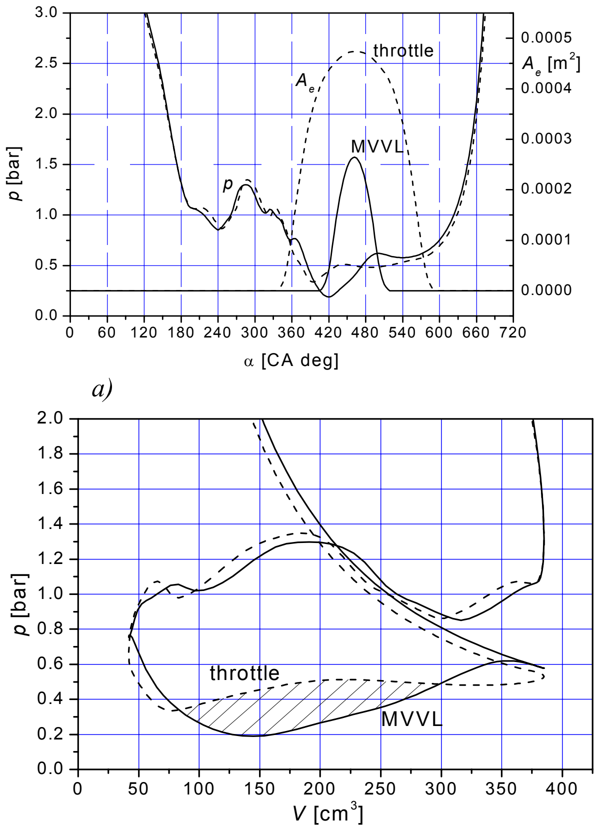

Figure 5. Cylinder pressur (*p*) and intake valve flow area (Ae) vs. crank angle with load control by throttle and mechanically variable valve lift (MVVL) (a), and corresponding low pressure *p*-*V* diagrams (b)

Fig. 5b shows that in the MVVL case induction work is even bigger than in the case of throttling.

However, if, with the decrease of intake valve lift, angular advance of camshaft is applied, i.e. if the mechanically variable valve lift and timing is applied (MVVLT), than the results are similar as in the case of EFVVT shown at Fig. 4.

b) Effect of lean mixture running

The biggest reduction of fuel consumption can be achieved using lean mixture at part throttle. Fig. 6 shows complete *p-V* diagrams and their low pressure parts for the engine running at part throttle with lean mixture λ=2.1 (the value which can be realized by charge stratification [2]). The load reduction has been simulated partly by mixture leaning and partly by smaller throttling (adjusted to obtain considered load - 12 kW at 3000 rpm). One should have in mind that lean mixture application means, at the same time, the needs for smaller throttling, i.e. better volumetric efficiency and smaller quantity of residual gases. Because of these reasons, the achieved fuel improvement economy of 17 % has been obtained.

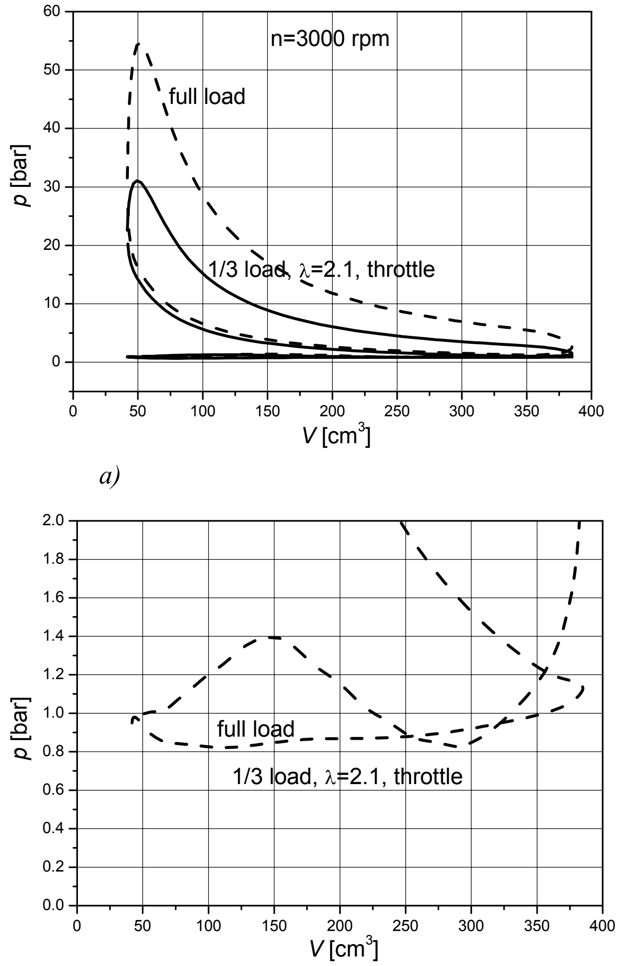

Figure 6. Cyliner pressure diagrams for full load and lean mixture part load (a) and their low pressure parts (b)

c) Effect of variable compression ratio

It is very known that it is possible to increase engine fuel economy using variable compression ratio. Though still very complicated, recent experiments and technical solutions, using moderate knock limited compression ratio at full load and higher compression ratio at part load, have proven this possibility [1]. Therefore, for the same previous mentioned working conditions (3000 rpm, 1/3 load), engine cycle simulation has been carried out using increased part load compression ratio ε=14 (approximately optimal value for this load [1]). Throttling, similar as in the case of conventional engine, has simulated a load decrease. Fig. 7 shows the computed pressure diagrams for full and part load of such engine. Comparing part load pressure diagrams of standard engine (Fig. 4) and variable compression engine (Fig. 7) it can be noticed much better shape of latter (increased combustion pressure, combustion close to TDC and lower pressure during expansion). The reduction of calculated specific fuel consumption is about 7 %.

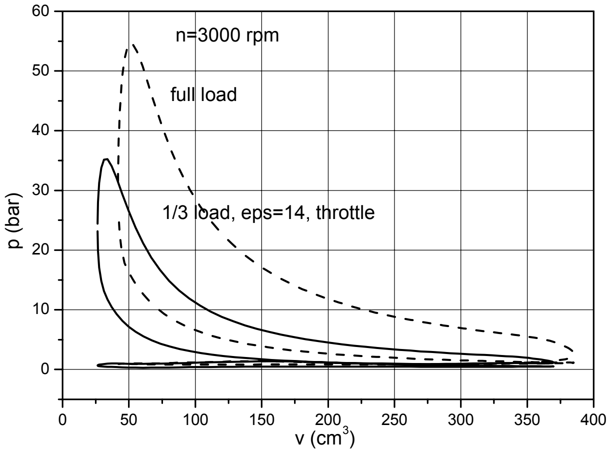

Figure 7. Cylinder pressure diagram for full load (ε=9,2), and part load (ε=14).

Fig. 8 summarizes the overall results of computed part load fuel consumption g_e and its reduction Δg_e for different afore-mentioned approaches: conventional fixed valve timing with part load throttling (THROTTLE), mechanically variable lift with fixed valve timing (MVVL), mechanicaly controled variable valve timing and lift (MVVLT), electronicaly fully variable valve timing (EFVVT), increased compression ratio (Eps=14) and lean mixture running (Lamb=2.1). The last bar (ALL) shows the effect of all combined measure: full valve timing control (EFVVT), increased compression ratio ε=14 and lean mixture λ=2.1. In the latter case part load fuel consumption reduction related to the conventional engine has been about 24 %. This value is very close to the actual part load diesel engine fuel consumption.

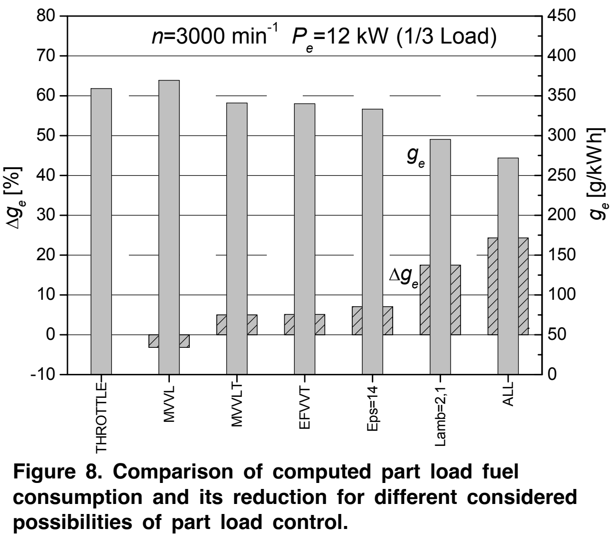

Figure 8. Comparison of computed part load fuel consumption and its reduction for different considered possibilities of part load control.

3. CONCLUSIONS

a) The theoretical analysis has been carried out using simulation model based on AVL BOOST code, which has been previously experimentally proved. The evaluation shows that there are significant possibilities of spark ignition engine fuel economy improvement at part load operation conditions.

b) Using load control by fully variable intake valve timing, without any throttling, a fuel economy improvement of approx. 5% can be achieved. The solutions with conventional camshaft and with electronically controlled mechanism for valve lift and timing variation are already commercialised. The systems with electromagnetic valve actuation that enable full flexibility of valve timing and lift has been widely investigated and their commercial application can be expected in near future.

c) The increase of compression ratio at part load operation conditions yields fuel consumption decrease of approx. 7%. There are several proposals and attempts for realisation of spark ignition engine with variable compression ratio, but commercial serial solution has not yet been realised.

d) The greatest potential of fuel economy improvement offers lean mixture combustion at low load operation conditions, according to the performed analysis approx. 17%. The solutions enabling this approach, such as spark ignition engines with direct gasoline injection and charge stratification are already commercialised.

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ПОБОЉШАЊЕ ЕКОНОМИЧНОСТИ ОТО МОТОРА НА ДЕЛИМИЧНИМ ОПТЕРЕЋЕЊИМА: НУМЕРИЧКА АНАЛИЗА

М. Томић , С. Петровић

У раду се анализирају могућности побољшања економичности рада ото мотора на парцијалним оптерећењима: смањење пумпних губитака (рада измене радне материје) путем контроле оптерећења изменом шеме развода и без пригушивања лептиром; повећање степена сабијања (односно примена варијабилног степена сабијања) и примена екстремно сиромашне смеше. Анализа је изведена теоријски, путем компјутерске симулације радног циклуса применом софистицираног програма AVL-Boost на моделу ото мотора од 1.4 *l* (Југо Флорида). Наведене могућности су квантификоване и упоређене, при чему је размотрена и комбинација ових мера.