EFFECT OF ENGINE CYLINDER DEACTIVATION ON FUEL ECONOMY AND CRANKSHAFT SPEED VARIATIONS

Miroljub V. TOMIĆ*[*1], Dragan M. KNEŽEVIĆ[2], Miloljub S. ŠTAVLJANIN[1]

[1]TehnoLab CMV d.o.o., Belgrade, Serbia

[2] University of Belgrade, Faculty of Mechanical Engineering, Belgrade, Serbia

[*] Corresponding author; E-mail: mtomic@mas.bg.ac.rs

Abstract: In this paper the cylinder deactivation technique in spark ignition engines was investigated. The potential of this concept was analysed using engine working cycle simulation model AVL Boost. The engine power output regimes corresponding to the vehicle moderate constant driving speeds were considered and the results show that at that low load engine operating regimes cylinder deactivation can enable fuel economy improvement of about 6-13% and corresponding CO_2 *emission reduction. The angular speed variations of engine crankshaft under cylinder deactivation conditions were also analysed and it was found that the speed variations increase several times compared to the standard operation of the engine with all active cylinders.*

Key Words: Spark ignition engine, Cylinder deactivation, Fuel economy, Crankshaft speed variations

1. Introduction

The main subject of interest of researchers in the field of automotive and internal combustion (IC) engines engineering in recent decades has been fuel economy and environmental protection. These problems are interrelated and cannot be considered separately. In the last decades, the exhaust emission of automobile engines has been remarkably reduced, by improving and precise control of engine working process (wide application of electronic control), and, on the other hand, by applying after treatment of exhaust emissions. In spark ignition engines, the application of 3-way catalytic treatment exceptionally reduced the emission of toxic gases: carbon monoxide (CO), unburned hydrocarbons (HC) and nitrogen oxides (NO_x). In diesel engines the wide applied technologies: oxidation and Selective Catalytic Reduction (SCR) treatment and solid particles (PM) filter with regeneration reduce harmful emissions to the smallest possible level. However, the problem of carbon dioxide (CO_2) emissions remains. This gas is not toxic, but it is responsible for the problem of global warming due to the greenhouse effect. As the result of complete combustion of carbon (C), CO_2 cannot be reduced by engine process improvement. In the case of hydrocarbon fuel use, the only possibilities of CO_2 emission reduction remain the use of fuel with lover carbon content and fuel consumption reduction. Even the application of natural gas, with a content of methane (CH_4) above 90%, only partially reduces the problem [1]. The use of hydrogen (H_2) as a fuel eliminates the problem, but although being researched, this is still far from mass application. The problem is high energy required for hydrogen production and complex manipulation.

Therefore, in the case of hydrocarbon fuel use, the only way to reduce CO_2 emission is the reduction of fuel consumption. During the development of IC engines, especially in the last decades, fuel consumption has been significantly reduced, as the result of working process improvement, better engine design and reduction of mechanical losses. For example, in spark ignition engine, the compression ratio, as one of the main factor influencing cycle efficiency, is significantly increased.

That is enabled by improving combustion process, and also by applying variable valve timing technique and so-called Miller's (Atkinson's) cycle.

1.1. Cylinder deactivation in IC engines

Unfavorable engine fuel economy is specially accentuated at low load engine operation. In spark ignition engines, poor fuel economy at low load conditions is mainly the result of the method of power regulation – throttling the intake air flow. This causes high pressure drop during the induction and high pumping losses. Additionally, the thermodynamic efficiency is reduced due to lower pressure and temperature. In the case of diesel engines, the problem is significantly less expressed because there is no intake air throttling, but the drop in thermodynamic efficiency still exists.

Cylinder deactivation (CDA) in multi-cylinder engines, as the method of engine fuel economy improvement at low load engine operation, has been known even in early engine development period. It is sometimes called “engine variable displacement” or “temporally downsizing”. The essence of the method is to deactivate part of engine cylinders during engine low load operation preventing their pumping losses, and in that way enable the rest of active cylinders to operate at higher load. Good review of CDA techniques and also potential and limitations can be found in [2], [3], [4], [5] and [6].

The first known application of some kind of CDA system is on the Sturtevant 38/45 HP 6-cylinder car produced in Boston 1905 [2]. The driver could deactivate 3 cylinders by stopping one of the magnetos and permanently opening the exhaust valves of the respective cylinders. There is no data of the system efficiency.

Enger Twin-Unit Twelve debuted in 1917, with a 3.7l 60° V12 engine with a lever on the steering column, which once operated, kept the exhaust valves open on one of the two banks of 6 cylinders, in order to avoid compression and at the same time, close the intake manifold on the same side of the respective cylinders [2].

Later, in second half of XX century and in XXI century more well-known car manufacturer applied CDA technique on some of their engines.

General Motors (GM) applied CDA on Cadillac 8-cylinder engine in 1981 and in 2005 and 2007, on 5.3L V8 and a 3.9L V6 engines using hydraulically deactivated valve lifters [5], [6]. GM states that fuel consumption improvement of 12% can be achieved.

Mitsubishi [5] launched the 1.4L 4-cylinder engine, which deactivates the intake and exhaust valves through hydraulically switching finger followers. This system proved to be imperceptible during activation, and a reduction in fuel consumption of up to 20% was possible.

Daimler Chrysler applied Active Cylinder Control for the first time in 2001, on a 5.8 l V12 used by Mercedes-Benz, using hydraulic lifters with deactivation by hydraulically operated bolts [2].

In 2008, Honda launched the 3.5L V6 variable cylinder engine with i-VTEC system [2], [6]. The i-VTEC mechanism applies pressure to a sliding bolt to connect the rockers operating the valves. When the bolt is not actuated, the rocker arm does not open the valves and the cylinder is deactivated.

Figure 1. VW cylinder deactivation system [2]

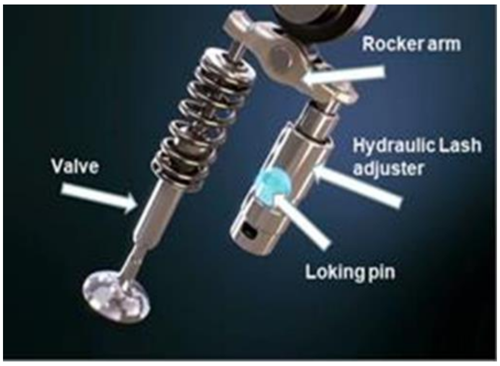

Figure 2. Mazda cylinder deactivation system [8]

In 2013, VW introduced Active Cylinder Technology (ACT) in its 1.4 TSI engine [2], [6], [7] and the system is shown in Fig.1. The camshafts are of complex structure since the cams actuating the inlet and exhaust valves of cylinders 2 and 3 (which are to be deactivated) are on grooved hollow shaft mounted on the base camshaft. Those hollow shafts have complex cam design: normal cams for valve opening and "zero opening" (circle) cams and they can be axially moved using electromagnetic actuators. So, if the hollow camshafts are in the "active" position (upper part of Fig. 1.) cams act on rocker arms and the valves are opened. By the movement hollow camshafts in "inactive" position (down Fig. 1.), the circle cams do not actuate rocker arms and the valves remain closed. Cylinders 2 and 3 are deactivated according to vehicle required driving power in the rage of 1400 – 4000 rpm and torque 25 – 75 Nm. VW claims that fuel consumption is reduced for 0.4 l/100km in NEDC driving cycle.

An Interesting newer solution of cylinder deactivation is used by Mazda in their SkyActiv spark ignition engines, and that technique is shown in Fig. 2. [8]. The hydraulic lash adjuster can be locked with the locking pin. In that case, the rocker arm acts on the valve and opens it. Otherwise, if locking pin unlocks the hydraulic lash adjuster, the rocker arm acts on it and the valve remains closed. Engine control system deactivates cylinders 1 and 4 at engine low load operating conditions i.e. when vehicle driving power requirements are low.

Cylinder deactivation has been previously applied mainly on the engines with a larger number of cylinders, 6 to 12. In recent decades this technique has been used on 4 cylinders and even on 3 cylinder engine - Ford Eco Boost 1.0 l turbocharged engine, the first 3 cylinder engine with cylinder deactivation. The system uses engine oil pressure to activate a special valve rocker and interrupt the connection between the camshaft and the valves of cylinder No. 1. on up to engine 4500 rpm and when running under light loads. The potential to improve fuel efficiency by up to 6 % is claimed [9].

As already mentioned, in the case of diesel engines, the reduction in fuel efficiency when operating at low load is less, but with modern automotive diesel engines the problem occurs in the after treatment of exhaust gases [10]. Namely, SCR catalyst operates most efficiently when temperatures are between 250 and 400 ^{o}C. Diesel cylinders deactivation can enable exhaust gas temperature on required level at low engine load, since active cylinders operate on higher temperature level. These can also be used at road loads to achieve efficient diesel particulate filter (DPF) regeneration [11].

The results of experimental investigation of diesel engine fuel consumption using CDA, can be found in [12]. The experiments were performed with 4 cylinder tractor engine with conventional fuel injection system on test bench in laboratory conditions and in driving conditions. In some engine low load operating conditions fuel consumption improvement of even to 20% was observed. In [13] a new concept has been investigated for deactivating only one of four cylinders of a commercial vehicle diesel engine with common rail fuel injection. For this purpose experimental investigation on test bench were carried out and also 1D process simulation were applied to analyze pumping losses.

2. Simulation of engine working process with cylinder deactivation

For numerical investigation of cylinder deactivation in IC engines the AVL BOOST IC engine working process simulation model was used [14]. All analyses were performed with the simulating model of modified spark ignition engine "FIAT 159A2.000" that was previously used for the "Zastava Florida" vehicle. The engine main technical data are shown in Tab. 1.

Engine modeling scheme formed in BOOST pre-processor is given in Fig. 3 and shows all engine elements that were taken into account in the simulation. All engine construction characteristics and equipment were specified in detail: cylinder design, intake and exhaust valves timing and lift, valves flow characteristics, throttles, flow restrictions, intake and exhaust pipes, fuel injectors, air cleaners, plenums, catalyst, etc.

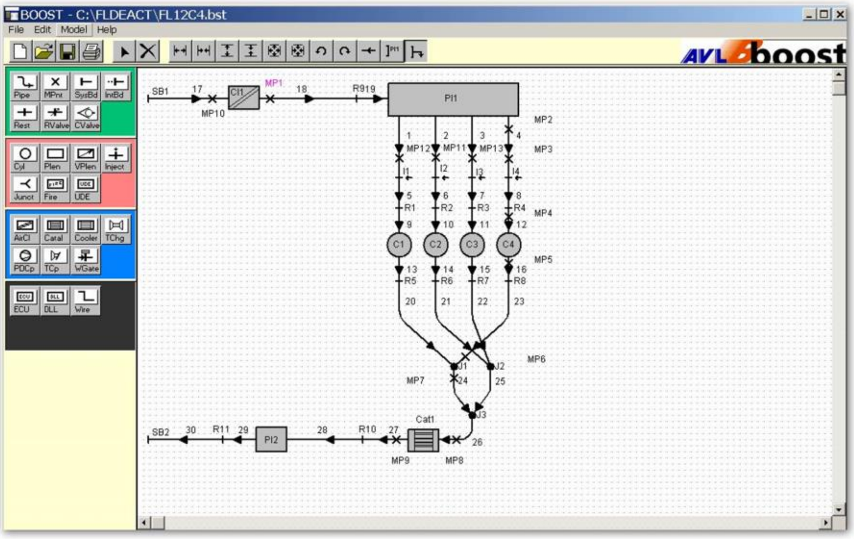

Figure 3. Engine scheme formed in pre-processor of AVL Boost simulation model

Table 1. Main engine data

Engine type	Spark ignition 4-stroke 4 cyl. in line	
Bore	mm	80.5
Stroke	mm	67.4
Swept volume	dm^3	1.372
Compression ratio		9.2
Brake power	kW	57 /6000 rpm
Brake torque	Nm	108 /3000 rpm

The engine operating regimes were chosen to meet the vehicle's power needs at constant speeds: 40, 60, 80 and 100 km/h, i.e. according to the vehicle total tractive resistance vs speed. This approximately corresponds to reduced engine load regimes when cylinder deactivation can be applied. At vehicle speeds greater than 100 km/h, the required power exceeds the possibility of cylinder deactivation.

Fig. 4 shows the vehicle tractive resistance force (F) at above mentioned constant speeds and required engine effective power (P_e), calculated with vehicle transmission system efficiency of 0.9. Corresponding engine working regimes are shown in Tab. 2.

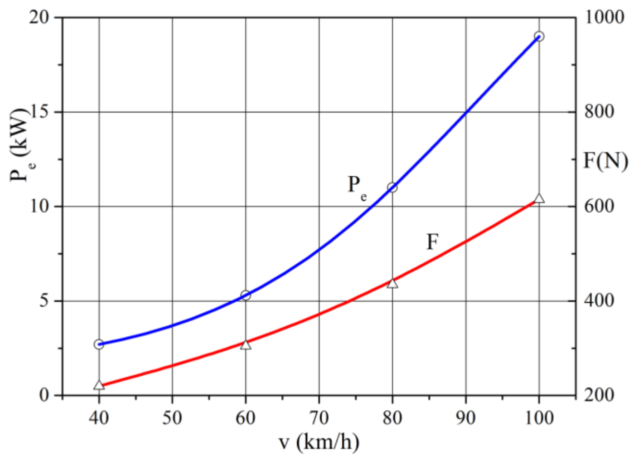

Figure 4. Vehicle tractive resistance force (F) and corresponding engine power (P_e)

Table 2. Engine operating regimes

Vehicle speed	n	P_e	BMEP
km/h	rpm	kW	bar
40	1500	2.7	1.57
60	1800	5.4	2.62
80	2400	11.2	4.08
100	3750	18.7	4.36

All engine working process simulations were performed with stoichiometric mixture (air excess ratio, λ=1). The strategy how to start deactivation and regarding what remain trapped in deactivated cylinder (combustion products or fresh air), and also the problems of cylinders reactivation have been investigated in [6], [15]. Experimental VW engine fueled with natural gas [15] was adapted and equipped with electro-hydraulic valve train, enabling wide optimization of intake and exhaust valve timing. The performed analyses show that the unfavorable case is to trap exhaust gases as the "gas spring", especially at high pressure and temperature [6]. In our particular case the initial conditions for deactivated cylinders (pressure and temperature) are adjusted so that the state at the start of compression is as in the case of normal engine operation.

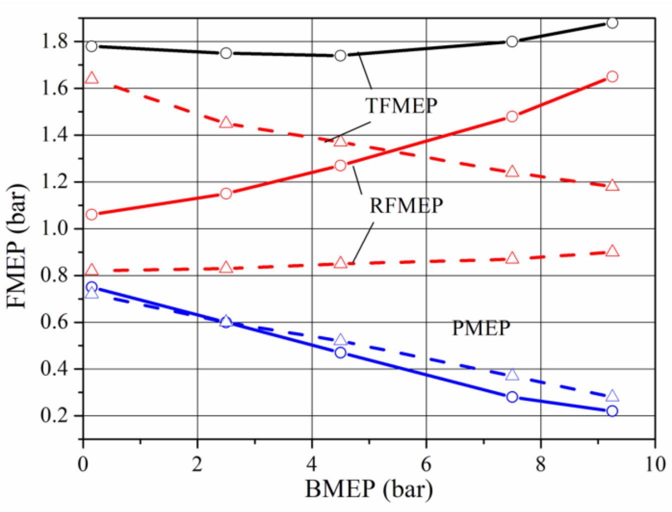

Figure 5. Mean effective pressure of losses vs engine load; BMEP- brake mean effective pressure, PMEP - pumping mean effective pressure, RFMEP- rubbing friction mean effective pressure, TFMEP – total friction mean effective pressure (PMEP +RFMEP +AMRP); full lines are for firing condition and dashed lines for motoring (non firing) conditions; 4-cylinder 3.26 dm^3 spark ignition engine, n=1600 rpm [16]

An insufficiently reliable assumption refers to the change of engine mechanical losses under the conditions of cylinder deactivation. Compared to standard engine operation, active cylinders work with higher pressure and temperature, while deactivated cylinders are motored and work with lower pressure and temperature. This has the repercussions on forces, surface temperatures, lubricating oil viscosity, oil film thickness and consequently on rubbing friction losses. Fig. 5 shows a classic dependence of rubbing friction losses and pumping losses on engine load [16]. In the case of motoring conditions (BMEP=0), the throttle positions are the same as in firing conditions for appropriate load. For intended comparative analyses the pumping losses (PMEP) are taken under account in engine process simulation, but the change of rubbing friction losses (RFMEP) should be analyzed. As can be seen, RFMEP decreases with load decrease, while under motoring conditions (without firing) it is almost independent of load.

Tribology and frictional aspects of cylinder deactivation were studied in [17]. It was shown that the friction between cylinder liner and piston rings in active cylinders in CDA operation is greater than in standard engine operation. This is even the case in inactive cylinders, in a smaller percentage.

In general, the auxiliaries drive losses (AMEP) are approximately independent of cylinder deactivation. In the case of CDA, active cylinders work with increased load and have greater RFMEP (Fig. 5). The inactive cylinders are without load, they are practically motored, but without pumping losses (valves are closed). Compression and expansion occurs with certain small energy loss. Pressures and temperatures are smaller and consequently the friction losses are reduced. However, some investigations show that this reduction cannot fully compensate the increase of mechanical losses in active cylinders [3]. The real total change of mechanical losses of the whole engine is difficult to be accurately estimated and that is why the mechanical losses were taken equally in simulation of standard engine operation and during cylinder deactivation.

2.1. Results of engine process simulation

The main results for all considered engine working regimes are shown in Tab. 3. Deactivation was realized in cylinders 2 and 3. The first two rows show the average values for the whole engine in standard operation (4 cyl.) and with CDA (2 cyl.). Next two rows show the result in single cylinders during CDA operation: CDA-1- active cylinder 1, CDA-2 - deactivated cylinder 2. As can be seen, the brake mean effective pressure (BMEP) of active cylinders is more than twice greater than in the case of all cylinders running. The maximum pressures (p_{max}) of the cycle are significantly greater in active cylinders, while the differences in maximum temperatures (T_{max}) are smaller.

The mean effective pressures of pumping losses (PMEP) are much smaller in the case of CDA engine operation. In deactivated cylinder there are no pumping losses and small negative indicated mean effective pressure (IMEP) is the difference between compression and expansion twice a cycle. In this case PMEP is formally considered as ½ of IMEP (Tab. 3). Brake specific fuel consumptions (BSFC), and the relative fuel consumption reduction expressed in % (ΔBSFC) are also specified in Tab. 3. As can be seen, the improvement of fuel economy is between 6 and 13 %, depending on engine load and speed. The less engine load, fuel consumption improvement is greater, which is the logical consequence of increased pumping losses at low load engine operating regimes. Engine brake specific fuel consumption and required power for considered vehicle constant speeds are shown on Fig. 6.

Table 3. The results of engine process simulation; P_e- brake power; IMEP-indicated mean effective pressure; BMEP-brake mean effective pressure; PMEP mean effective pressure of pumping losses; p_{max} maximum cycle pressure; T_{max} maximum cycle temperature; BSFC brake specific fuel consumption; ΔBSFC - BSFC reduction

	n	P_e	IMEP	BMEP	PMEP	p_{max}	T_{max}	BSFC	ΔBSFC
	rpm	kW	bar	bar	bar	bar	K	$\frac{g}{kWh}$	%
4 cil	1500	2.7	2.44	1.57	-0.698	16.3	2191	481.4	
CDA	1500	2.71	2.45	1.58	-0.289			417.7	13.2
CDA-1	1500		5.14	4.27	-0.513	27.6	2323		
CDA-2	1500		-0.09	-0.96	-0.048	5.94	699.4		
4 cil	1800	5.4	3.55	2.616	-0.601	19.7	2285	368.4	
CDA	1800	5.4	3.55	2.615	-0.182			335.7	8.9
CDA-1	1800		7.28	6.34	-0.308	36.5	2386		
CDA-2	1800		-0.1	-1.03	-0.05	6.75	700		
4 cil	2400	11.2	5.13	4.083	-0.546	27.0	2390	315.5	
CDA	2400	11.2	5.14	4.085	-0.144			295.7	6.3
CDA-1	2400		10.36	9.31	-0.245	49.1	2486		
CDA-2	2400		-0.11	-1.16	-0.056	8.8	697		
4 cil	3750	18.7	5.42	4.36	-0.678	28.1	2464	308.2	
CDA	3750	18.7	5.42	4.362	-0.229			284.0	7.8
CDA-1	3750		10.9	9.84	-0.407	51.6	2515		
CDA-2	3750		-0.1	-1.15	-0.049	8.9	702		

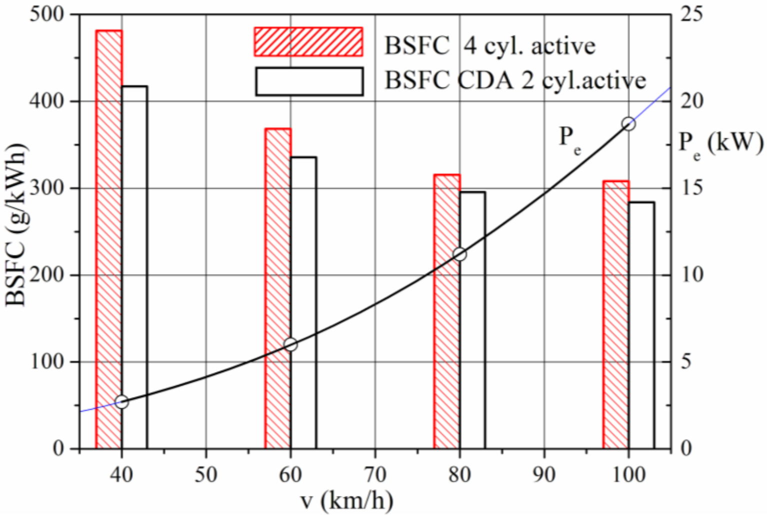

Figure 6. Required engine power (P_e) and BSFC

For one of the operating regimes considered (n=1800 rpm and BMEP=2.62 bar) p-α and T-α diagrams are shown in Fig. 7 and 8. The results for active cylinder 1 and inactive cylinder 2 are shown as the function of cylinder 1 crank angle. The pressure difference between the active cylinder in standard and CDA operation is very large. In inactive cylinder compression and expansion of trapped charge occurs (gas spring effect) with decreasing amplitude caused by heat transfer and blow by. The investigations curried out in [6] and [15] show that stationary state can be reached after approximately 100 cycles. The differences in temperature between active cylinders are smaller. Maximum temperature of the cycle in active cylinder in CDA operation is about 100 K higher than in standard engine operation.

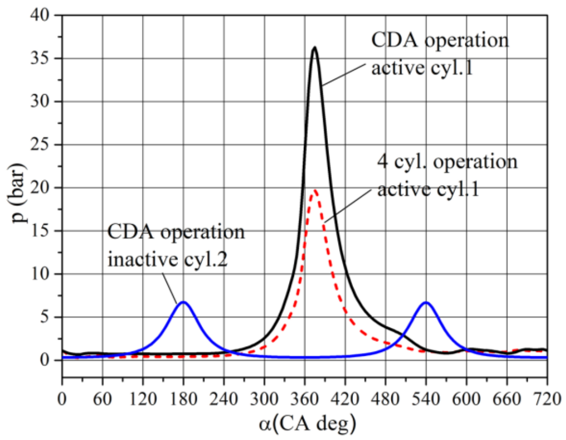

Figure 7. Pressures vs crank angle for engine standard and CDA operation; n=1800 rpm, BMEP= 2.62 bar

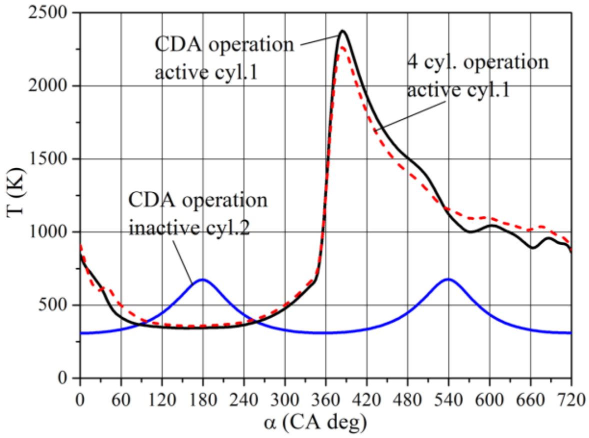

Figure 8. Temperatures vs crank angle for engine standard and CDA operation; n=1800 rpm, BMEP= 2.62 bar

Indicator p-V diagrams of active cylinders 1, for standard engine operation and for CDA operation, are shown on the Fig. 9. and 10. In the case of CDA engine operation positive indicator work of the cycle in active cylinder is more than twice higher (IMEP=7.28 bar), than in standard engine operation (IMEP=3.55. bar).

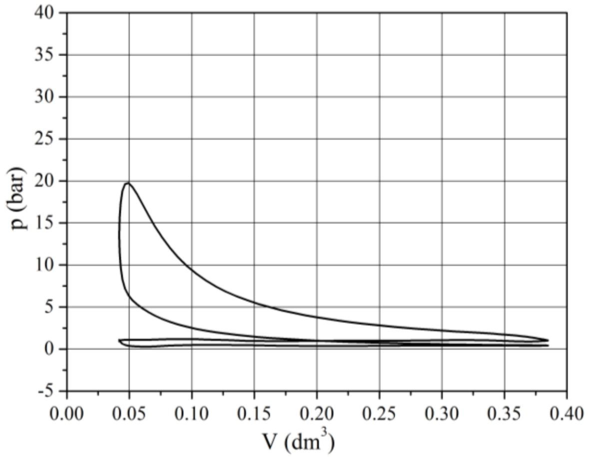

Figure 9. Indicator p-V diagram; n=1800 rpm, BMEP=2.62 bar; standard engine operation, cyl. 1, IMEP=3.55 bar

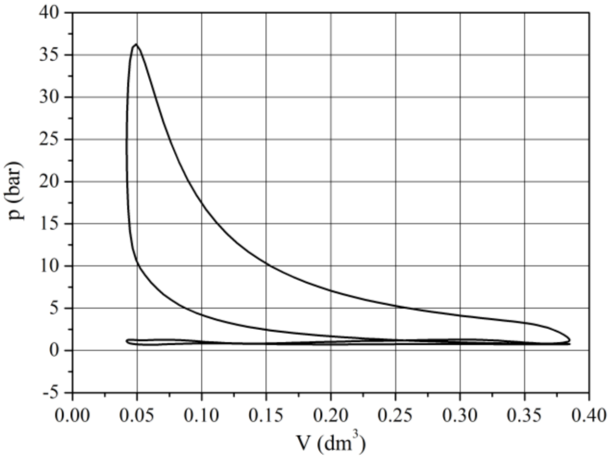

Figure 10. Indicator p-V diagram; n=1800 rpm, BMEP=2.62 bar; CDA engine operation, active cyl. 1, IMEP=7.28 bar

In Fig. 11. and 12. diagram scale is adjusted to clearly show the negative work of gas exchange (pumping loss) in low pressure part of the cycle. It is considerably smaller in the case of CDA engine operation, and in active cylinder is PMEP = -0.308 bar, while in standard engine operation PMEP = -0.601 bar. Higher pressures in active cylinder cause higher gas speed and pressure wave during the exhaust (Fig. 12.). In standard engine operation (Fig. 11.) exhaust speeds are lower and the pressure wave intensity is much smaller.

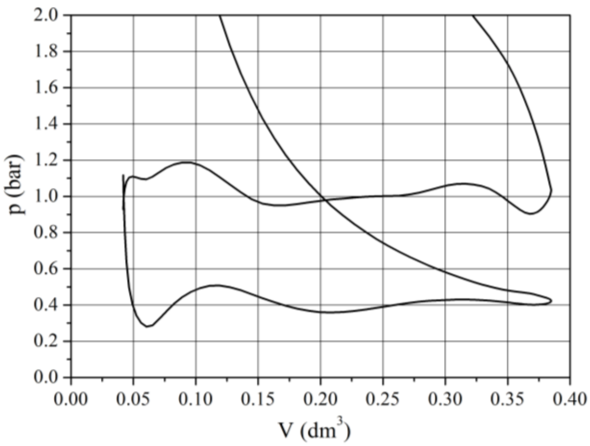

Figure 11. Low pressure part of indicator p-V diagram (pumping work); standard engine operation; cyl. 1; n=1800 rpm, BMEP=2.62 bar, PMEP = -0.601 bar

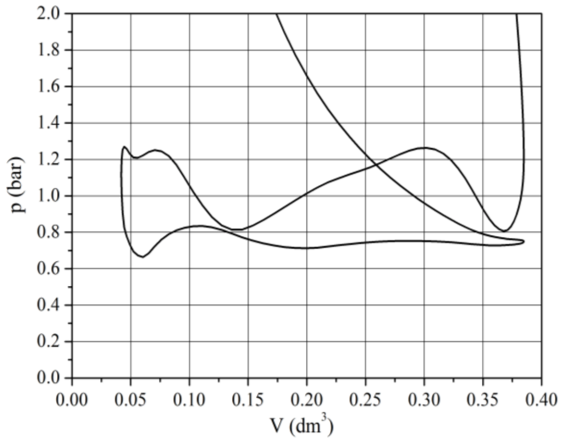

Fig.12. Low pressure part of indicator p-V diagram (pumping work); CDA engine operation; active cyl. 1; n=1800 rpm, BMEP=2.62 bar, PMEP = -0.308 bar

3. Analysis of crankshaft rotational speed variations

Under CDA engine operation conditions, the firing interval is doubled (360 instead of 180 CA degrees) and the variations of tangential forces and torques on the crankshaft are much greater. This has repercussion on crankshaft rotational speed variations during the cycle. Those problems, including possibility of torsion vibrations, were considered and analyzed in more studies and papers [18], [19].

The resulting force of gas pressure and of inertia of oscillatory masses produces tangential force (F_t) acting on the crankshaft. Those forces from all engine cylinders produce the current engine torque (M_t) which is variable according to the variations of tangential forces. The mean value of engine torque (M_{tm}) is in equilibrium with the constant torque of external resistance. During the period when M_t is greater than M_{tm} crankshaft accelerates and otherwise, when M_t is less than M_{tm}, it decelerates. Rotational speed of the crankshaft varies and reaches their local extreme values in the positions when $M_t = M_{tm}$. (Fig. 13. and 14.).

For the analyses of engine crankshaft rotational speed variations the differential equation of torque equilibrium can be formulated as:

$$M_t = M_{tm} + J_0 \frac{d\omega}{dt} \tag{1}$$

Here, J_0 is the engine equivalent mass moment of inertia, ω crankshaft rotational speed and t time. The current engine torque is in equilibrium with external resistance torque (equal to M_{tm}) and the inertia of engine rotating masses. This equation enables evaluation of engine rotational speed:

$$M_t - M_{tm} = J_0 \frac{d\alpha}{dt}\frac{d\omega}{d\alpha} = J_0 \omega \frac{d\omega}{d\alpha}; \qquad (M_t - M_{tm})d\alpha = J_0 \omega d\omega \tag{2}$$

The equation (2) can be integrated in the interval ω_{min} to ω_{max} (Fig. 13. and 14.)

$$\int_{\alpha 1}^{\alpha 2} (M_t - M_{tm})d\alpha = J_0 \int_{\omega min}^{\omega max} \omega d\omega = J_0 \frac{1}{2}(\omega_{max} + \omega_{min})(\omega_{max} - \omega_{min}) \tag{3}$$

where α_1 and α_2 are angular positions of ω_{min} and ω_{max}.

$$\int_{\alpha 1}^{\alpha 2} (M_t - M_{tm})d\alpha = J_0 \delta \omega^2 \tag{4}$$

Here δ is so called „coefficient of fluctuation of speed“.

$$\delta = \frac{2(\omega_{max} - \omega_{min})}{(\omega_{max} + \omega_{min})} = \frac{(\omega_{max} - \omega_{min})}{\square} = \frac{W_t}{J_0 \omega^2} \tag{5}$$

`The expression:

$$W_t = \int_{\alpha 1}^{\alpha 2} (M_t - M_{tm})d\alpha \tag{6}$$

is called „excess work of tangential force“ and it produces acceleration of the crankshaft from ω_{min} to ω_{max}.

The crankshaft rotational speed can be evaluated by the integration of equation (2) for the whole engine cycle (4π rad). The forces of gas pressure are obtained by engine process simulation, and inertia forces of oscillating masses m_0 (mass of piston group and part of connecting rod) are calculated with m_0=0.683 kg. Further, tangential forces acting on engine cranks and corresponding torque were calculated and then crankshaft rotational speed evaluated by numerical integration of equation (2).The calculation was carried out with the value of the equivalent mass moment of inertia of the engine J_0=0.07 kgm^2.

Table 4. The results of crankshaft rotational speed variations evaluation

	n	BMEP	M_{tm}	□	W_t	δ	$□_{min}$	$□_{max}$
	rpm	bar	Nm	s^{-1}	J	-	s^{-1}	s^{-1}
4 cil	1500	1.57	17.2	157.08	18.23	0.0105	156.2	157.9
CDA	1500	1.58	17.25	157.08	90.43	0.0524	153.0	161.2
4 cil	1800	2.616	28.6	188.5	22.7	0.0091	187.6	189.4
CDA	1800	2.615	28.6	188.5	140.8	0.0566	183.2	193.8
4 cil	2400	4.085	44.56	251.33	20.86	0.0047	250.7	251.9
CDA	2400	4.085	44.56	251.33	198.0	0.045	245.7	257.0
4 cil	3750	4.36	47.56	392.7	108.65	0.01	390.7	394.7
CDA	3750	4.362	47.58	392.7	299.4	0.0227	387.3	398.1

The results for engine standard operation and with CDA are shown in Tab.4. As can be seen, crankshaft rotational speed variations are significantly greater in the case of engine CDA operation. The values of excess work of tangential force (W_t) and coefficient of fluctuation of speed (δ) are approximately 3 to 10 times greater, depending on the engine operating regime. Figs. 13 and 14 show engine torque and crankshaft rotational speed variations for operating regime n=3750 rpm and BMEP=4.36 bar (corresponding for vehicle speed of 100 km/h). As can be seen, the variations of crankshaft rotational speed are almost three times greater in the case of CDA application. In the operating regimes with lower rpm the difference in variations of crankshaft rotational speed are even greater. Therefore, when applying CDA, the increased vibrations of engine on the support should be considered and taken into account.

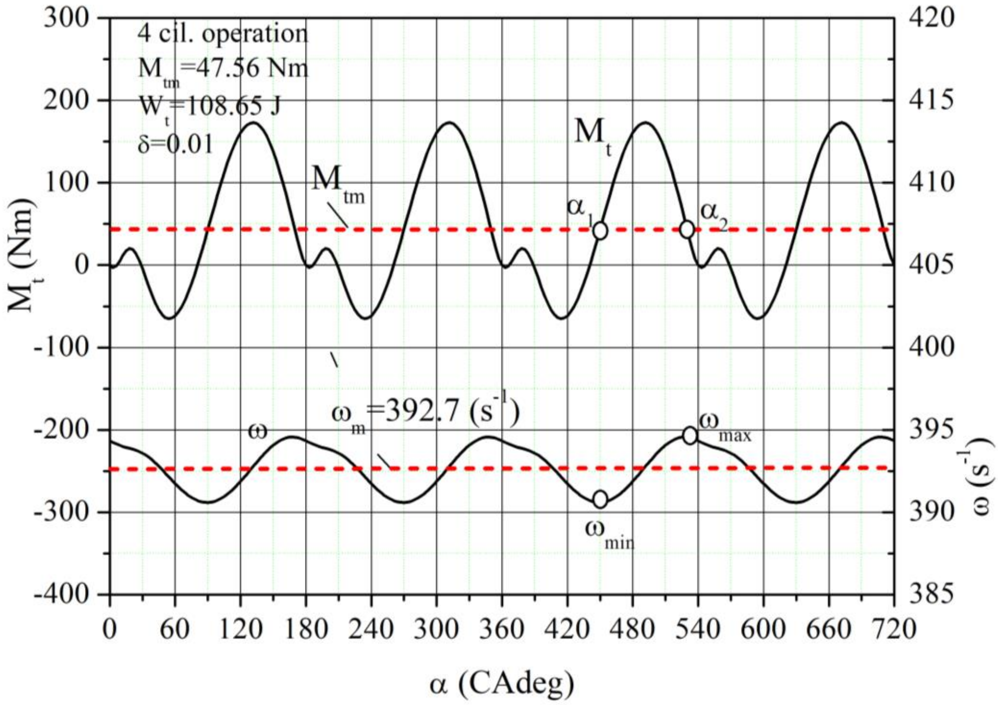

Figure 13. The variations of engine torque (M_t) and rotational speed (ω); standard engine operation; all cylinders active; n=3750 rpm, BMEP=4.36 bar

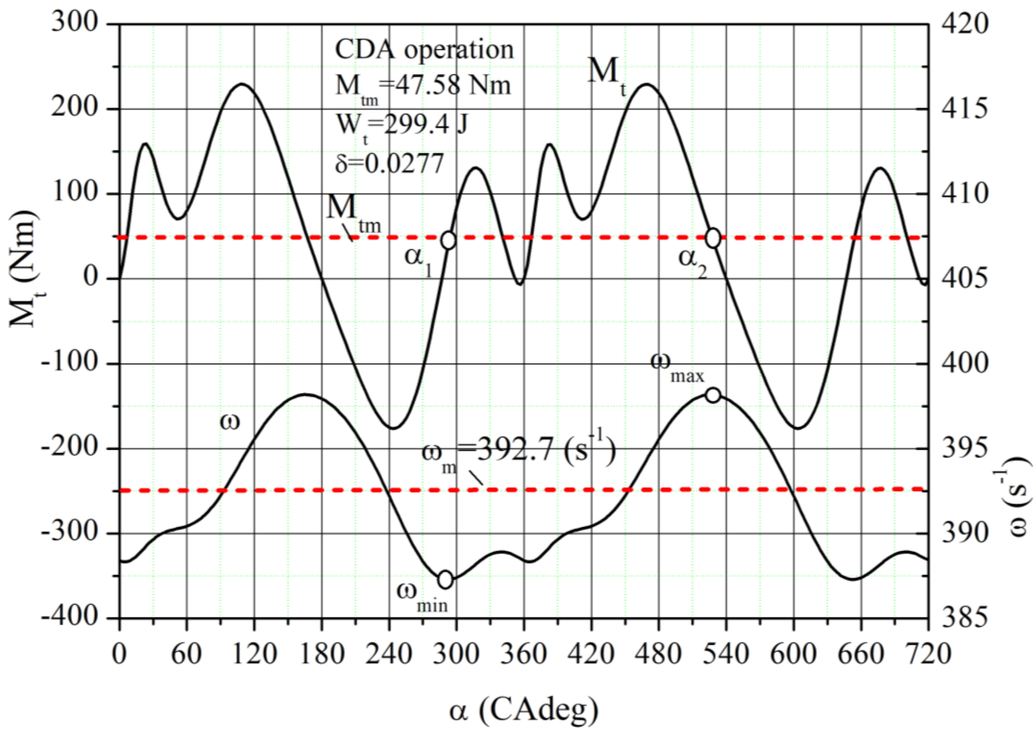

Figure 14. The variations of engine torque (M_t) and rotational speed (ω); CDA engine operation; 2 and 3 cyl. deactivated; n=3750 rpm, BMEP=4.362 bar

4. Conclusions

The comparative investigation of engine standard operation (all cylinders active) and operation with CDA (cylinders 2 and 3 deactivated), was carried out by engine process modeling, using AVL Boost engine simulating model. Based on obtained results, the following conclusions can be formulated:

1. Cylinder deactivation is an effective way to improve engine fuel economy at low load engine operation. The brake specific fuel consumption is reduced thanks to less intake throttling and reduced pumping losses in active cylinders, which operate with increased load, and to the absence of pumping losses in inactive cylinders. Higher pressure and temperatures also contribute to thermodynamic efficiency.

2. Simulation of the engine working process shows that when the engine is operating at light load, which corresponds to low power requirements at moderate vehicle speeds, CDA can enable fuel consumption to be reduced by 6-13%. The simulation was performed with equal friction losses in the case of standard and CDA engine operation. A slightly different improvement in fuel consumption is to be expected due to a change in total friction losses of the entire engine. A more comprehensive investigation of frictional losses under cylinder deactivation conditions is needed.

3. Under CDA engine operating conditions, the firing interval is doubled and the increased engine torque variations produce a greater level of crankshaft rotational speed variations. The performed analyzes show that the variations of crankshaft rotational speed are increased several times.

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