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Influence of filling pressure on formation of total efficiency in the 2SZ-FE engine

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Investigation results of analysis of filling pressure influence on total engine efficiency at variable phases of the valve timing system setting were presented in this paper. Investigations were performed on a combustion engine of Toyota Prius with Atkinson cycle. Making use of an adequate inlet system and proper choice of timing gear setting in relation to load and rotational speed of the engine 2SZ-FE considerable increments of volumetric efficiency were obtained. Profiting from filling pressure measurements performed by means of an electro optic pressure sensor at constant loads and rotational speeds, and prolonging the charging process increase in general engine efficiency was obtained. Using a phase shifter its direct influence on the working field of the whole engine 2SZ-FE work cycle was shown. A further possibility of filling efficiency increase by changing the inlet system at higher rotational speeds of the engine was indicated.

Nomenclature

BDC - Bottom Dead Center
TDC - Top Dead Center
OIV - opening of inlet valves
CIV - closing of inlet valves
MVF - mean value of filling traces
Nu - Nusselt number
Re - Reynolds number
η_o - *total efficiency*
η_v - *volumetric efficiency*
M_1 - *quantity of inflammable mixture before combustion [kmol/kg]*
W - lower heating value [kJ/kg]
p_o - *surrounding pressure [MPa]*
T_o - *surrounding temperature*
p_e - *mean effective pressure [MPa]*

1. Introduction

Among the structural factors which exert most significant influence of the quality of the filling coefficient information of the inlet system and proper choice of timing gear phases setting should be mentioned. Rotational speed and engine loading are here the most important exploitation factors. Formation of the inlet system acts on volumetric efficiency by the flow resistance value and by its influence on dynamic phenomena during charging process [1, 2]. In order to obtain small flow resistances, hence, a high filling coefficient possibly high valves should be used. Aiming at getting low flow resistances, hence, a high filling coefficient in valves should be possibly big with short head channels and short inlet pipes with a small number of bends and sufficiently big cross sections should be applied in engine designing. As dynamic phenomena are concerned no general conclusions and consequently no structural instructions can be formulated. Final formation and dimensions of the inlet system were chosen during test-bed investigations. Setting up of the timing gear i.e. coordination of work of the timing gear mechanism and the movement of the crankshaft shows considerable influence on the filling coefficient quantity. The difference between the traces of cylinder filling pressure values at constant rotational speed, constant load, and fixed throttle opening is shown in Figs 1 and 2 in function of the opening angle of inlet valves.

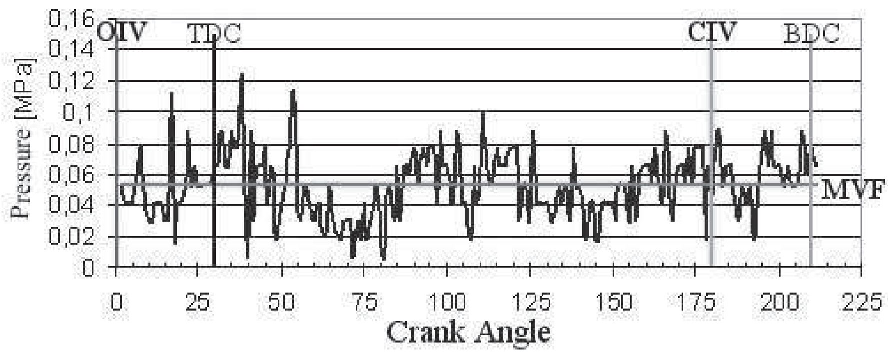

Fig 1. Traces of pressure change during filling stroke at rotational speed 1500 rpm, 25% of throttle opening and inlet valve opening phase 30^{o} CA BTDC.
Rys. 1. Przebieg zmian ciśnienia podczas suwu napełniania przy prędkości obrotowej 1500 obr/min, 25% otwarciu przepustnicy i fazie otwarcia zaworów dolotowych 30^{o}OWK przed GMP.

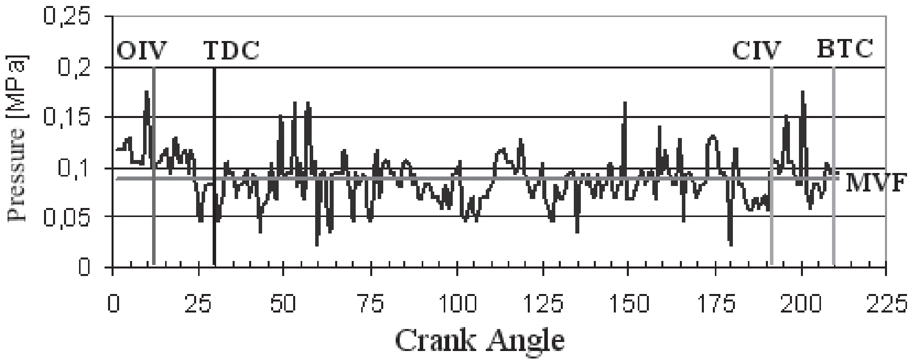

Fig 2. Traces of pressure changes during filling stroke at rotational speed 1500 rpm. 25% of throttle opening and inlet valve opening phase 12° CA BTDC.
Rys. 2. Przebieg zmian ciśnienia podczas suwu napełniania przy prędkości obrotowej 1500 obr/min, 25% otwarciu przepustnicy i fazie otwarcia zaworów dolotowych 12°OWK po GMP.

For possibly exact clearing of the cylinder from exhaust gases and its proper filling with a fresh charge the valves are opened with certain delay. Sooner opening of the inlet valves assures a properly large flow section of the valve already at the starting moment of the inlet stroke and permits scavenging of the whole combustion space. Delay in the inlet valve closing enables prolongation of the charging process beyond BDC due to making use of dynamic phenomena in the inlet system [3].

Sooner opening of the inlet valve creates favorable conditions to fuller and quicker removal of delayed closing leads to reduction of the rests of exhaust gases due to their additional removal in consequence of the sucking off action of the exhaust gases column in the outlet pipe. As a matter of front prolongation (of course in determined limits) of the time of valve opening is conducive to better cylinder filling. With increase in rotational speed the time corresponding to valve opening decreases. Therefore advance angles of valve opening and closing must be respectively bigger in high speed engines than in low speed ones. Increase in semi-opening angle of valves i.e. angle of crankshaft revolution corresponding to the period when inlet and outlet valves of one cylinder are open at the same time, improves clearing of the combustion space of the rests of exhaust gases due to better scavenging with a fresh charge [4]. Application of considerable semi-opening of valves is possible only in engines of inner formation of mixture (compression ignition and fuel injection engines) since in the case of outer mixture formation unacceptable fuel loss would take place.

In traditional engines total efficiency is limited by two basic factors [5]:

- filling losses,
- expansion ratio which is not constant and its value is established by the maximum ratio of compression where the threshold limit of mixture self-ignition is within the limits 1:10 [6].

However, efficiency increase is observed with increase in expansion ratio (to the value 17:1). Increase in efficiency related to increase in expansion ratio is no more significant in the range 25:1. Atkinson cycle aims at adaptation of parameters of engine work for maintenance of the compression ratio in its maximal value below the limit of knock and maintenance of an effective expansion ratio.

The cycle elaborated by Atkinson offers solution of two complicated problems:

- it assures that supply of charge to the cylinder proceeds without filling losses and the compression degree is differentiated from the expansion ratio,
- this cycle was also called "the cycle of five strokes" where five separate piston strokes can be distinguished: inlet reciprocal flow (partial removal aiming at control charging) compression, work, outlet.

2. Characteristics of Atkinson cycle

The indicatory diagram (Fig. 3) shows a comparison of Otto and Atkinson cycle at 50% load.

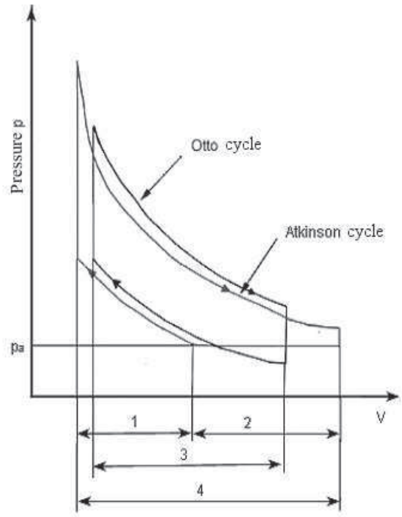

Fig. 3. Comparison of Otto and Atkinson cycle: 1-compression stroke of Atkinson cycle, 2-delay in inlet valve closing, 3-compression and expansion stroke of Otto cycle, 4-expansion stroke of Atkinson cycle, p_a - surrounding pressure.

Rys. 3. Porównanie obiegów Otto i Atkinsona: 1 - suw sprężania obiegu Atkinsona, 2 - opóźnienie zamknięcia zaworu dolotowego, 3 - suw sprężania i rozprężania obiegu Otto, 4 - suw rozprężania obiegu Atkinsona, p_a - ciśnienie otoczenia.

At this load the pressure of the final, part of the filling stroke is in a conventional Otto engine lower than atmospheric pressure. Whereas, in Atkinson engine this process proceeds at atmospheric pressure but with compression phase which starts half way of piston stroke. Basing on the above diagram the following conclusion can be drawn:

- maximal pressure gained by a conventional engine is lower than the pressure obtained in Atkinson cycle,
- no filling losses appear since the sum of the charge trapped in the cylinder was determined by delayed closing of the inlet valve,

- compression stroke and combustion chamber volumetric capacity were adapted to adequate filling but the compression degree remained uncharged (at 50% load).

A high degree of expansion of the cycle causes a high caloric effect in Atkinson cycle. The increasing expansion ratio - caused by reducing the combustion chamber capacity – makes the engine use up the total combustion energy. The relation of the expansion ratio in a conventional engine and engine of high expansion degree is shown in Fig. 4.

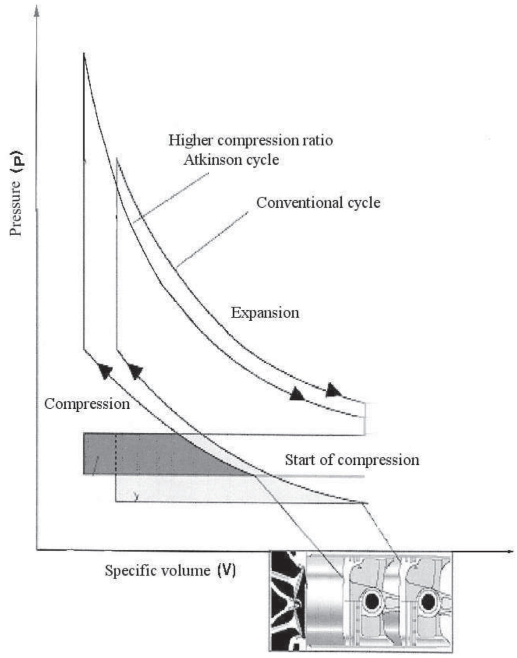

Fig. 4. Comparison of parameters of the conventional and Atkinson cycle [7].
Rys. 4. Porównanie parametrów obiegu konwencjonalnego i Atkinsona [7].

A considerable increase in compression and expansion ratio causes knock combustion and roughness of engine work. In order to avoid these problems closing of the inlet valve was delayed (and in the initial stage of compression due to what a part of the charge return to the inlet manifold.) Due to it the expansion ratio was increased without increase in the actual compression ratio [8]. The time of inlet valve closing is dependent on engine work condition. Fig. 5 shows some variants of valve closing in dependence on the crankshaft position [9].

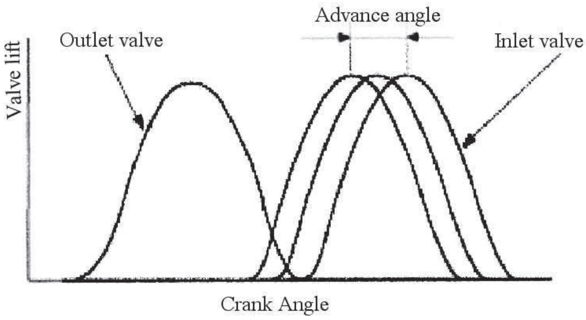

Fig. 5. Inlet valves lift in dependence on engine work conditions.
Rys. 5. Wznios zaworów dolotowych w zależności od warunków pracy silnika .

Variable Valve Timing – of the inlet valve is chosen in such a way as to obtain during engine work highest efficiency in a large range at rotational speed of the engine. Variable phases of the timing gear in Toyota Prius are shown in Fig. 6.

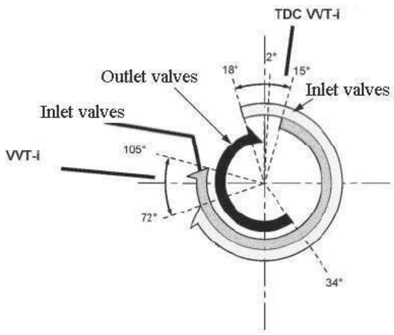

Fig. 6. System of timing gear phases in Toyota Prius II generation [103].
Rys. 6. Układ faz rozrządu w Toyocie Prius II generacji [103].

The heat stream reaching the walls during outlet and inlet of the charge is considerably smaller than during combustion. Nevertheless it is commonly known that heat exchange during the inlet phase influences greatly the volumetric efficiency.
Zaph [10] presented empirical correlation based upon the Woshny's formula [6] describing the influence of preliminary whirl on the coefficient of heat exchange during charge exchange.

$$\mathrm{Nu} = \mathrm{Nu}_0\left(1 + 0{,}0062\frac{V_z}{V_{tł}}\right), \tag{1}$$

where:

Nu_o – is the Nusselt's number for a whirlless case expressed by Worshiny's formula

The heat exchange coefficient described by Worshiny has an averaged in time global value for the inlet and outlet phase. Nishiwaki [10] et al elaborated heat exchange coefficients averaged in time separately for inlet and outlet:

- for the inlet process $Nu_d = 0,168\ Re^{0,867}$,
- for the outlet process $Nu_w = 1,69Re^{0,578}$, where Re is calculated on the basis of mean piston velocity.

3. Determination of general efficiency basing on pressure measurement results

Due to pressure measurements in the combustion chamber and application in investigations of a computational programme in agreement with the block scheme of cycle calculations algorithm shown in Fig. 7 the following is possible:

- quick analysis of efficiency of full engine work cycles,
- observation of closed indicator diagrams during engine work,
- quick analysis of total efficiency for determined engine's working parameter.

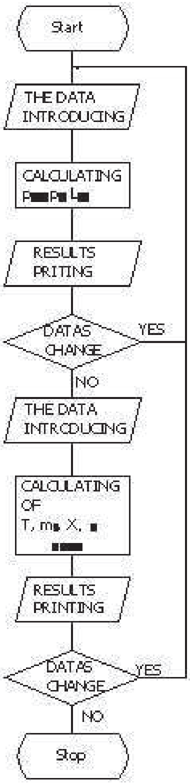

Fig. 7. Block scheme of cycle calculation algorithm.
Rys. 7. Schemat blokowy algorytmu obliczania obiegów.

Total efficiency is determined from equation [11]:

$$\eta_o = 8{,}311 \frac{M_1 p_e T_0}{W \eta_V p_o}, \tag{2}$$

where:

M_1 – quantity of inflammable mixture before combustion [kmol/kg]
W – lower heating value 43000 [kJ/kg]
p_o – surrounding pressure [MPa]
η_v – volumetric efficiency
T_o – surrounding temperature 288 [K]
p_e – mean effective pressure [MPa]

The computational scheme of cycle parameters consists of three independent blocks:
1 – block of data entrance, calculations and listing of computational values,
2 – graphic block of real cycle presentation in coordinates p – V,
3 – block of determination of particular circulation efficiency of full engine work cycles.

4. Determination of effective pressure by use of full engine work cycle measurements

Effective pressure may be determined from formula:

$$p_e = p_i - p_m \tag{3}$$

where:

p_i – indicated pressure [MPa]
p_m – mechanical losses pressure [MPa]

According to formula 3 and exemplary diagram 8 where indicated pressure is the relation of indicated work (L_i) and displacement volume (V_s).

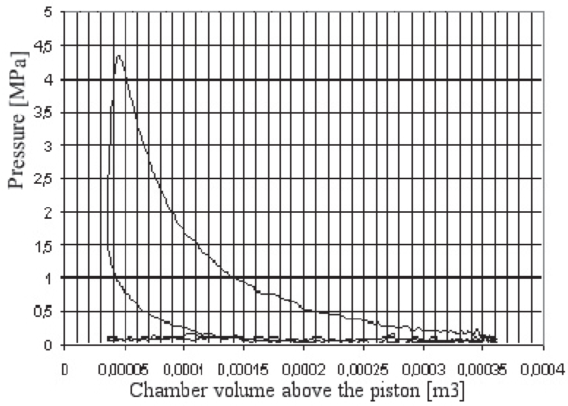

Fig. 8. Closed indicated diagram in form of example as result of engine 2SZ-FE investigations.
Rys. 8. Przykładowy zamknięty wykres indykatorowy wynikający z badań silnika 2SZ-FE.

In calculations of indicated work the computational programme takes into consideration also angular ranges in which particular strokes proceed i.e.:

- filling stroke from $\alpha_{1p} = 0^o$ to $\alpha_{1k} = 210^o$ CA,
- compression stroke from $\alpha_{2p} = 180^o$ to $\alpha_{2k} = 360^o$ CA,
- combustion expansion stroke from $\alpha_{3p} = 360^o$ to $\alpha_{3k} = 510^o$ CA,
- outlet stroke from $\alpha_{1p} = 510^o$ to 720^o CA.

Particular ranges show extreme position of piston strokes of engine 2SZ-FE used in test bed investigations. Work of particular strokes is determined by numerical integration of the full engine work cycle resulting from a closed indicated diagram.

In order to take into consideration mechanical losses within the range of rotational speed 1000-3500 rmp, for engine 2SZ-FE, use was made of relation [12]:

$$p_m = 0{,}0980666 \cdot (0{,}3 + 0{,}1 \cdot c_{śr}) \tag{4}$$

where:

$c_{śr}$ – mean piston speed [m/s]

for the examined cycle $c_{śr}$ may be expressed as:

$$c_{śr} = \frac{Sn}{30} \tag{5}$$

where:

S – piston stroke 0,0797 [m]

n – rotational speed (rmp).

Pressure of mechanical losses within the scope of performed investigations is shown in speed characteristics (Fig. 9)

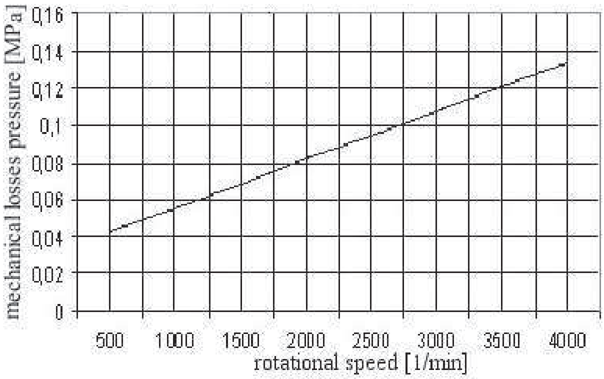

Fig. 9. Characteristics of friction pressure losses within the scope of investigations.
Rys. 9. Charakterystyka strat tarcia ciśnienia w zakresie badań.

5. Determination of filling efficiency within the scope of engine 2SZ-FE investigations

Making use of pressure measurements in the cylinder volumetric efficiency is determined and presented in form of relation (6) and an example of determination of average efficiency for rotational speed 3000 rpm of the phase shifter with CA in TDC and load 80 kW is shown in Fig. 10.

$$\eta_v = \left(\frac{p_1}{p_0} - \frac{1}{\varepsilon - 1} \cdot \frac{p_{rs} - p_1}{p_0} \right) \cdot \frac{T_0}{T_0 + \Delta T}, \tag{6}$$

where:

ΔT – temperature increment of the charge supplied to the cylinder in consequence of hot walls effect, was adopted 40 [K],
p_r – pressure of exhaust gases rests was adopted 0,104 [MPa],
T_o – surrounding temperature during investigations 293 [K],
p_1 – filling pressure [MPa],
p_o – investigations surrounding pressure 0,0993 [MPa],
ε – compression ratio for the examined engine is 10.

Introducing particular values from investigations of an engine with Atkinson cycle and Otto engine at 50% load (Fig. 10) increment of total efficiency is obtained as shown in Fig. 11.

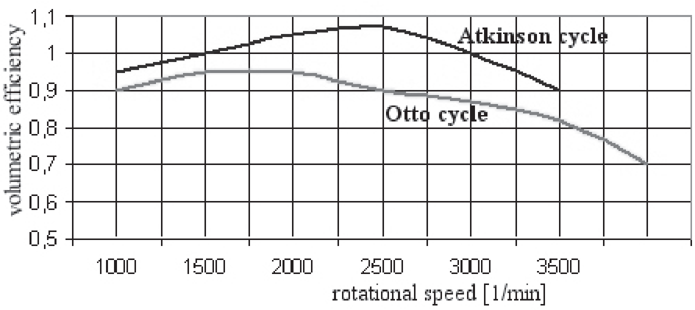

Fig. 10. Comparison of volumetric efficiency of engine with Atkinson cycle and Otto engine as function of rotational speed (at about 50% load).
Rys. 10. Porównanie sprawności wolumetrycznej silnika z obiegiem Atkinsona do silnika Otto jako funkcji prędkości obrotowej (przy ok. 50 % obciążeniu).

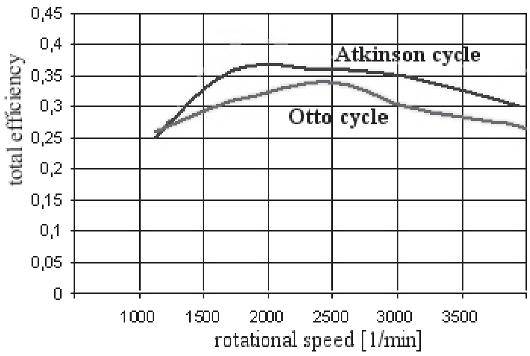

Fig.11. Increment of general efficiency of Atkinson cycle engine compared with Otto engine.
Rys. 11. Przyrost sprawności ogólnej silnika z obiegiem Atkinsona do silnika Otto.

6. Summary

1. Application of variable timing gear phases conditioning occurrence of Atkinson cycle permits to obtain some increase in filling efficiency within the range of rotational speeds from 1500 to 3500 rpm of statistically highest load density in time.
2. In an analogous way application of variable timing gear phases conditioning occurrence of Atkinson cycle makes it possible to gain a total efficiency increase in the rotational speed range from 1500 to 3500 rpm of statistically highest load density in time.
3. It was shown in this elaboration that the obtained performances of the engine permit to realize the idea of downsizing i.e. to obtain higher engine performances at unchanged main dimensions.

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Wpływ ciśnienia napełniania na kształtowanie się sprawności ogólnej w silniku 2SZ-FE

Streszczenie

W pracy przedstawiono wyniki badań analizy wpływu ciśnienia napełniania na sprawność ogólną przy zastosowaniu zmiennych faz rozrządu. Wykorzystując odpowiedni układ dolotowy oraz dobór ustawienia rozrządu w stosunku do obciążenia i prędkości obrotowej silnika 2SZ-FE (Toyota Prius) uzyskano znaczne przyrosty sprawności wolumetrycznej (napełniania). Wykorzystując pomiary ciśnień napełniania optoelektronicznym czujnikiem ciśnienia przy stałych obciążeniach i prędkościach obrotowych, wydłużając procesy ładowania wykazano przyrost sprawności ogólnej. Wykorzystując przesuwnik fazowy wykazano bezpośredni wpływ na pole pracy całego obiegu silnika 2SZ-FE. Wykazano dalszą możliwość wzrostu sprawności napełniania poprzez zmianę układu dolotowego przy wyższych prędkościach obrotowych silnika.

Tabela 1. Tabela równań transmitancji.
Table 1. Table with LaPlace equations.

Ilość okresów	Model ARX					Model OE	Model ARMAX	
	1 1 1	**1 1 0**	**2 1 1**	**2 2 1**	**1 2 0**	**2 2 1**	**2 2 1 1**	**2 2 2 1**
1 okres	0.1584 ----------- s - 0.03529 3,01%	0.8152 s + 0.7494 ----------------- s - 0.1658 22,02%	-0.5177 s - 0.9835 ---------------------- s^2 - 0.2018 s + 0.216 93,71% ω_0=0,464 c=----	7.162 s - 0.1584 ----------------------- s^2 + 1.454 s + 0.03524 93,86% ω_0=0,187 c=3,872	5.918 s + 1.131 --------------- s - 0.2485 92,13%	0.5603 s^2 + 22.93 s + 0.5865 ------------------------------------ s^3 + 0.4043 s^2 + 9.908 s - 0.09744 40,03% ω_0=---- c=----	0.6617 s - 0.8852 ------------------------ s^2 + 0.01319 s + 0.1947 93,21% ω_0=0,441 c=0,014	0.209 s^2 + 44.25 s + 7.288 --------------------------------- s^3 - 0.7155 s^2 + 10.04 s - 1.602 91,12% ω_0=---- c=----
2 okresy	0.1709 ----------- s - 0.03792 2,99%	0.7909 s + 0.729 ---------------- s - 0.1609 19,32%	-0.5617 s - 1.058 ---------------------- s^2 - 0.2483 s + 0.232 92,18% ω_0=0,481 c=----	6.78 s - 0.07571 ----------------------- s^2 + 1.315 s + 0.01685 90,54% ω_0=0,129 c=5,065	6.169 s + 1.252 --------------- s - 0.2755 91,21%	0.5828 s - 0.875 ------------------------ s^2 - 0.01989 s + 0.1926 92,25% ω_0=0,438 c=----	2.157 s - 0.6652 ----------------------- s^2 + 0.3022 s + 0.1465 90,53% ω_0=0,382 c=0,394	2.219 s - 0.6288 ----------------------- s^2 + 0.2865 s + 0.1385 88,15% ω_0=0,372 c=0,384
3 okresy	0.2227 ----------- s - 0.04913 3,75%	0.8661 s + 0.7926 ----------------- s - 0.1748 21,04%	-0.6045 s - 1.129 --------------------- s^2 - 0.288 s + 0.249 84,02% ω_0=0,498 c=----	3.347 s - 0.4416 ------------------------ s^2 + 0.5244 s + 0.09763 80,08% ω_0=0,312 c=0,839	6.297 s + 1.409 --------------- s - 0.3103 83,94%	1.493 s - 0.7161 ---------------------- s^2 + 0.148 s + 0.1577 86,51% ω_0=0,397 c=1,86	3.895 s - 0.2893 ------------------------ s^2 + 0.6698 s + 0.06503 51,41% ω_0=0,255 c=1,313	2.326 s - 0.5566 ---------------------- s^2 + 0.2961 s + 0.123 79,89% ω_0=0,351 c=0,422
4 okresy	0.238 ----------- s - 0.05242 3,98%	0.8864 s + 0.8097 ----------------- s - 0.1783 21,71%	-0.5747 s - 1.077 ---------------------- s^2 - 0.272 s + 0.2372 79,34% ω_0=0,487 c=----	3.731 s - 0.2853 ------------------------ s^2 + 0.6615 s + 0.06275 82,91% ω_0=0,251 c=1,321	6.049 s + 1.405 --------------- s - 0.3091 83,11%	-7.825 s^2 - 53.75 s - 52.2 --------------------------------- s^3 + 5.412 s^2 + 18.8 s + 11.48 -85,46% ω_0=---- c=----	1.702 s - 0.661 ----------------------- s^2 + 0.1826 s + 0.1456 82,76% ω_0=0,381 c=0,239	2.197 s - 0.555 ----------------------- s^2 + 0.3079 s + 0.1222 83,61% ω_0=0,349 c=0,441
5 okresów	0.2236 ----------- s - 0.04926 3,63%	0.8569 s + 0.785 ---------------- s - 0.1729 20,65%	-0.5669 s - 1.063 ---------------------- s^2 - 0.2707 s + 0.234 80,81% ω_0=0,483 c=----	3.659 s - 0.3153 ------------------------ s^2 + 0.6524 s + 0.06935 83,72% ω_0=0,263 c=1,238	5.967 s + 1.364 --------------- s - 0.3003 83,64%	0.221 s - 0.181 ------------------------- s^2 + 0.04641 s - 0.07613 -91,11% ω_0=---- c=----	3.333 s - 0.3641 ------------------------ s^2 + 0.5728 s + 0.08081 70,32% ω_0=0,284 c=1,007	2.674 s - 0.4849 ----------------------- s^2 + 0.4196 s + 0.1067 83,91% ω_0=0,326 c=0,642
6 okresów	0.2184 ----------- s - 0.04804 3,42%	0.8476 s + 0.7771 ----------------- s - 0.171 20,28%	-0.5704 s - 1.07 ---------------------- s^2 - 0.2698 s + 0.235 81,68% ω_0=0,484 c=----	3.912 s - 0.2816 ------------------------ s^2 + 0.6967 s + 0.06206 84,41% ω_0=0,249 c=1,398	6.104 s + 1.384 --------------- s - 0.3043 84,37%	-4.878 s^2 - 0.8773 s - 47.21 --------------------------------- s^3 + 2.005 s^2 + 11.11 s + 10.39 -83,43% ω_0=---- c=----	2.162 s – 0.5682 ---------------------- s^2 + 0.2974 s + 0.125 85,01% ω_0=0,353 c=0,421	3.161 s - 0.3855 ------------------------ s^2 + 0.5248 s + 0.08483 84,98% ω_0=0,291 c=0,901
7 okresów	0.2346 ---------- s - 0.0516 3,82%	0.8713 s + 0.7971 ----------------- s - 0.1754 20,91%	-0.5783 s - 1.082 ----------------------- s^2 - 0.2829 s + 0.238 82,26% ω_0=0,487 c=----	3.71 s - 0.2947 ----------------------- s^2 + 0.6445 s + 0.0649 84,87% ω_0=0,254 c=1,264	6.121 s + 1.417 --------------- s - 0.3116 84,58%	3.705 s - 0.2801 ------------------------ s^2 + 0.6447 s + 0.06164 85,03% ω_0=0,248 c=1,298	2.158 s - 0.5667 ----------------------- s^2 + 0.2894 s + 0.1251 80,66% ω_0=0,353 c=0,409	2.104 s - 0.5745 ----------------------- s^2 + 0.2745 s + 0.1265 83,91% ω_0=0,355 c=0,385