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Cycle-by-cycle variability in CAI engine with internal EGR and direct gasoline injection

Abstract: Cycle-by-cycle variability in CAI gasoline engine was studied in terms of indicated mean effective pressure and combustion timing variations. Cyclic variability was analyzed for two different engine loads and different injection timings, applied during negative valve overlap. It was found that fluctuations of combustion course and produced indicated work reveal deterministic oscillations, where engine was operated in close to misfire regime. Moreover, kind of correlation between consecutive cycles was found to be affected by fuel injection strategy.

Keywords: *controlled auto-ignition, cycle-by-cycle variability, injection strategy*

Zmienność cykliczna procesu roboczego silnika CAI z wewnętrzną recyrkulacją spalin i bezpośrednim wtryskiem benzyny

Streszczenie: W pracy przedstawiono analizę zmienności cyklicznej procesu roboczego benzynowego silnika CAI (controlled auto-ignition) przeprowadzoną na podstawie średniego ciśnienia indykowanego oraz przebiegu spalania. badania przeprowadzono dla dwóch wartości obciążenia silnika oraz różnych kątów początku wtrysku paliwa w czasie ujemnego współotwarcia zaworów. Uzyskane wyniki wykazały występowanie zdeterminowanych oscylacji przebiegu kątowego spalania oraz ilości pacy generowanej w cyklu roboczym w warunkach pracy silnika w pobliżu granicy wypadania zapłonów. Ponadto zaobserwowano, że rodzaj korelacji pomiędzy kolejnymi cyklami zależny jest od strategii wtrysku.

Słowa kluczowe: *kontrolowany samozapłon, zmienność cykliczna, strategia wtrysku*

1. Introduction

Each combustion engine reveals some cycle-by-cycle variability, even if is operated in a steady conditions [8]. The cyclic instability can be observed via variations of in-cylinder pressure between separate cycles, especially during combustion. An indicated mean effective pressure (IMEP) is variable as well, thus resulting in torque fluctuations and deterioration of engine efficiency [7].

Cycle-by-cycle fluctuations are usually identified as combustion variations in terms of peak pressure, maximum heat release rate (HRR), amount of heat released and combustion timing [7]. Other parameters, characterizing engine working processes, like in-cylinder fluid motion, amount of re-circulated exhaust, fuel distribution within the cylinder are widely recognized as reasons for operation roughness.

Considering specific principles of controlled auto-ignition (CAI) engine operation, approach to the cyclic variability should be verified as well. Internal exhaust gas re-circulation (EGR) is utilized for increasing in-cylinder internal energy in order to achieve auto-ignition of the mixture. Thus, amount of re-circulated exhaust and its temperature is main factor determining combustion timing [5, 9, 10]. In-cylinder conditions during intake process affect amount of fresh air, which enters the cylinder. As a result, thermal balance between trapped residuals and intake air determines compression temperature histories in individual cycles. However, Koopmans et al. [5] related ignition timing to HC content in the internally re-circulated exhaust.

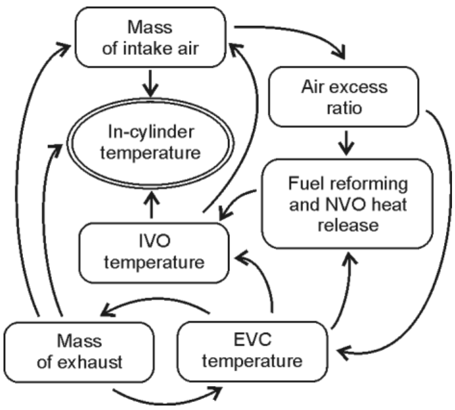

Fig. 1. Graphic presentation of internal relationships between parameters determining in-cylinder compression temperature in CAI engine
Rys. 1. Graficzna prezentacja wewnętrznych zależności pomiędzy wielkościami determinującymi temperaturę sprężania w cylindrze silnika CAI

Application of fuel injection during a negative valve overlap (NVO) period introduces further complexities into the cyclic variability mechanisms, as shown in Fig. 1. Early NVO injection enables fuel reforming, which is an endothermic process and causes drop of in-cylinder temperature. However, NVO injection at lean mixtures and presence of some amount of oxygen in the re-circulated exhaust can lead to heat release during the NVO period, thus influencing in-cylinder temperature.

In general, CAI engines reveal much lower cycle-by-cycle variability in terms of IMEP [6]. However, this improvement results from short combustion period taking place at almost constant volume, close to piston top dead center (TDC). In fact, cyclic variations in start of combustion (SOC) timing are much higher than in Diesel or spark ignition engines where SOC is forced via fuel injection or spark discharge.

In recent years researchers put their attention to combustion stability in CAI engines. The variations between consecutive cycles seem to be a key process which allows for self-regulation of in cylinder mixture composition and its thermodynamic properties.

Daw et al. [1] and Sen et al. [9] demonstrated deterministic patterns of cyclic variations of heat release rate in CAI engine. It was assumed that cyclic variations result mainly from nonlinear internal EGR feedback.

Recently, Ghazimirsaied and Koch [2] developed an algorithm enabling prediction of combustion timing one cycle ahead on the base of peak pressure crank angles from two previous cycles. They have also demonstrated peak pressure patterns at engine operation close to misfire limit.

However, mentioned efforts to describe cyclic variability considered CAI engines with mixture formation outside the cylinder. In the case of direct fuel injection it can be assumed, that complex mechanisms of mixture formation will alternate identified combustion variability relationships.

The aim of this study was identification of cycle-by-cycle variability mechanism in CAI engine with direct fuel injection applied during NVO. Influence of combustion course on IMEP from the point of view of cyclic variations was also investigated.

2. Experimental test stand

The examinations were carried out using a SB 3.5 single cylinder research engine with a fully variable valvetrain mechanism. All valvetrain parameters could be changed independently for the intake and exhaust valves during engine operation. Variable valve lift was achieved with the use of a hydraulic device, described in details in ref. [4]. The research engine had a bowl shaped combustion chamber located in the engine head. The piston face was protruding on its perimeter and approached the cylinder head closely at TDC, which generated some amount of squish. The main engine parameters were as follows: cylinder bore was 84 mm, piston stroke – 90 mm and compression ratio – 11.7. The engine head was fitted with an in-cylinder pressure transducer.

Solenoid swirl-type injector was used for fuel metering. The injector was positioned tangentially to the swirl generated by the shape of the intake port and inclined by 38 ° in relation to the cylinder axis. Detailed description of combustion system can be found in ref. [3].

The research engine was coupled to a DC motor dynamometer, which allowed for motored engine operation.

The engine control system was based on a microprocessor timing module governed by a personal computer with a real-time software. A dedicated injection and ignition timing module was designed in order to allow accurate and repeatable dosing of fuel and spark discharge generation.

3. Experimental conditions and data analysis procedure

Experiments were performed at constant rotational speed of the engine and constant valvetrain settings, specified in Table 1. The engine was fuelled with commercial gasoline (95 RON) from a single batch.

Table 1. Experimental conditions
Tabela 1. Warunki badań

Description	Unit	value
Rotational speed	rev/min	1500
Throttle position	%	50
IVO	°CA	87
IVC	°CA	217
Intake valve lift	mm	3.6
EVO	°CA	515
EVC	°CA	634
Exhaust valve lift	mm	2.9
NVO	°CA	173
SOI after TDC NVO	°CA	-40, 20
Fuel pressure	MPa	10
Air excess ratio	-	1.0, 1.3

Four different operating conditions were taken into consideration. Engine was fuelled with stoichiometric and lean mixture, close to misfire limit. Fuel was injected in the early stage of NVO and in the late stage of NVO. The first injection strategy allowed fuel reforming, while the second strategy provided homogeneous mixture without reforming.

Data analysis was based on in-cylinder pressure traces, acquired with crank angle resolution of 0.1 °CA for 100 consecutive cycles. Net heat release rate traces were calculated for each cycle using the first law of thermodynamics. In order to obtain cycle resolved mass of air and re-circulated exhaust, flow model was applied for pressure data analysis. Combustion timing characteristic values were derived from mass fraction burnt (MFB) courses obtained via integration of HRR traces. Coefficient of variation (CoV) in IMEP was calculated as ratio of IMEP standard deviation divided by mean value from analyzed data series. Overall engine operating parameters, combustion timings and CoV in IMEP at selected conditions are shown in Table 2.

Table 2. Main combustion parameters at analyzed engine operating conditions

Tabela 2. Wskaźniki procesu roboczego w analizowanych warunkach pracy silnika

λ [-]	1.0		1.3	
SOI [°CA]	-40	20	-40	20
η_V [-]	0.31	0.30	0.22	0.29
EGR [-]	0.49	0.49	0.64	0.54
T_{EXH} [K]	791	781	602	701
IMEP [-]	0.29	0.28	0.14	0.20
p_{max} [MPa]	3.63	3.50	2.64	2.44
φ (p_{max}) [°CA]	365.8	366.7	364.5	372.6
φ (5% MFB) [°CA]	359.9	360.4	355.6	364.4
φ (50% MFB) [°CA]	362.4	363.3	360.1	370.4
φ (5-95% MFB)[°CA]	5.4	5.1	8.5	12.2
HRR_{max} [J/°CA]	103.7	98.6	24.8	30.3
φ (HRR_{max}) [°CA]	362.5	363.4	360.0	370.1
CoV (IMEP) [%]	3.40	2.86	4.20	9.81

The applied injection strategy did not affect combustion timing in a high extent at stoichiometric mixture, as shown in Table 2. However, early NVO injection and resulting fuel reforming negligibly improved volumetric efficiency. This phenomenon can be ascribed to cooling effect of reforming and reduction of IVO temperature. In case of lean mixture opposite and much stronger influence of injection timing on amount of intake air was observed. Early NVO injection and presence of oxygen in the re-circulated exhaust resulted in heat release during NVO. Thus, increase of IVO temperature reduced volumetric efficiency. In contrast, heat of fuel evaporation at late injection allowed increase of volumetric efficiency.

It should be noted that at lean mixture late NVO injection considerably deteriorated cycle-by-cycle stability of IMEP.

4. Experimental results

Figure 2 presents in-cylinder pressure curves for 100 consecutive cycles. In case of stoichiometric mixture injection timing did not affect combustion timing or pressure rise rate. However, at lean mixture early injection resulted in smaller dispersion in start of combustion than one obtained for late injection. In case of Late NVO injection (Fig. 2d) early start of combustion produced relatively high pressure rise rate, while late auto-ignition was followed by slow combustion.

Late injection provided better fuel conversion efficiency than early injection for stoichiometric mixture. At SOI equal -40 °CA indicated specific fuel consumption (ISFC) was equal 320 g/kW h, while at SOI equal 20 °CA fuel consumption was 317 g/kW h. This difference is a result of NVO mixture formation phenomena. Early injection and resulting fuel reforming increased indicated work of NVO to 15 J versus 12 J at late injection, while indicated work calculated from IVC to EVO was equal 164 J versus 159 J accordingly.

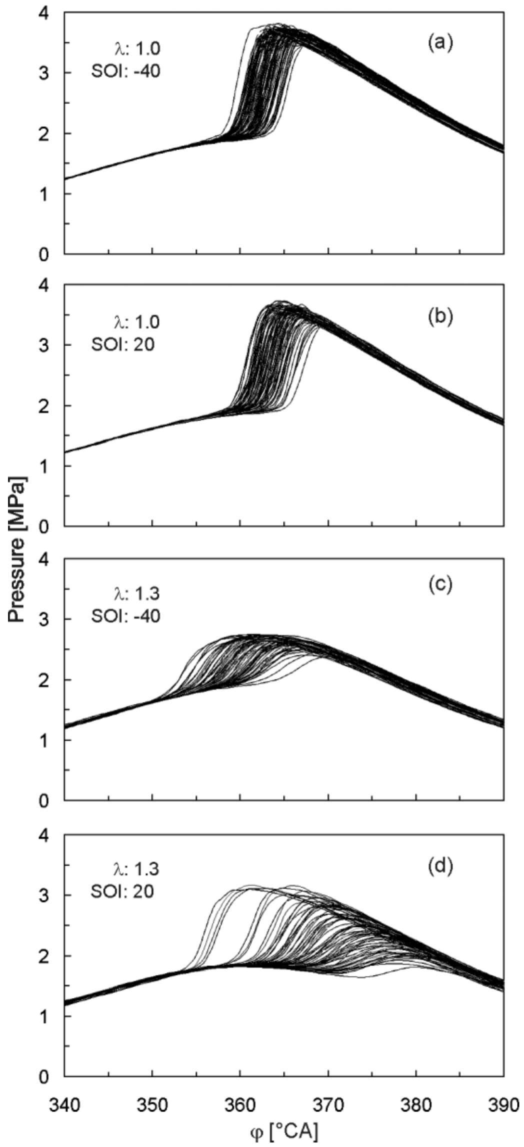

Fig. 2. In-cylinder pressure curves for 100 consecutive cycles

Rys. 2. Krzywe ciśnienia w cylindrze dla 100 kolejnych cykli pracy silnika

At lean mixture early NVO injection deteriorated thermal efficiency in comparison to late injection was observed. Application of fuel injection during exhaust compression resulted in heat release during NVO which reduced amount of heat released during main combustion. ISFC was equal 368 g/kW h for early injection versus 350 g/kW h for late injection. However, early injection provided reduction of volumetric efficiency, thus lower engine load was achieved.

Figure 3. presents return maps of IMEP, where values for one and two cycles ahead were shown versus value for current cycle. Return maps for stoichiometric mixture show stochastic character of IMEP cyclic variability (Fig. 3a,b). At lean mixture and early injection small scale deterministic oscillations were observed. Most of cycles provided IMEP around mean value and lack of correlation was noted. However, so called excursions from mean value were observed (Fig. 3c). This effect can be clearly seen for late fuel injection (Fig. 3d) and excessive CoV of IMEP. Some number of cycles which produced IMEP close to mean value were followed by low IMEP, slow combustion cycles. The lower amount of heat released in the current cycle, the higher work was produced by one or even two cycles ahead. This phenomenon can be interpreted as circulation of combustible exhaust compounds. If slow cycles appear, temperature drop causes combustion termination and result in high CO content in the exhaust gas. Re-circulated CO is utilized in the following cycle, thus producing additional energy.

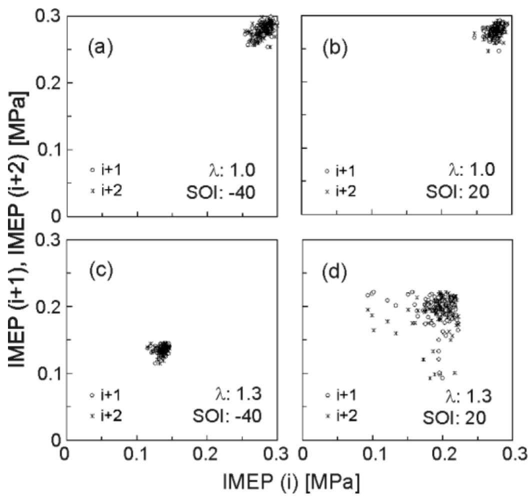

Fig. 3. Return maps of IMEP (one and two engine cycles ahead current cycle)
Rys. 3. Średnie ciśnienie indykowane w kolejnych dwóch cyklach jako funkcja średniego ciśnienia w bieżącym cyklu

Cycle-by-cycle variability in IMEP, apart from amount of chemical energy in the cylinder, is related to combustion timing as well, as shown in Fig. 4. At stoichiometric mixture, where short combustion duration was observed, maximum fuel economy was achieved where 50% of MFB appeared few CA degrees after TDC. Earlier combustion timings resulted in deterioration of engine efficiency, as some amount of heat was released during compression (Fig. 4a,b). However, at lean mixture, early combustion did not have negative effect on fuel conversion efficiency, as shown in Fig. 4c. It could be ascribed to influence of in-cylinder temperature, which was extremely low due to lean mixture and high EGR rate. Thus, relatively slow combustion could be terminated by dropping in-cylinder temperature during expansion. This effect can be seen for late injection (Fig. 4d), where slow cycles occurred. The later combustion took place, the lower fuel utilization was observed, as a result of flame quenching.

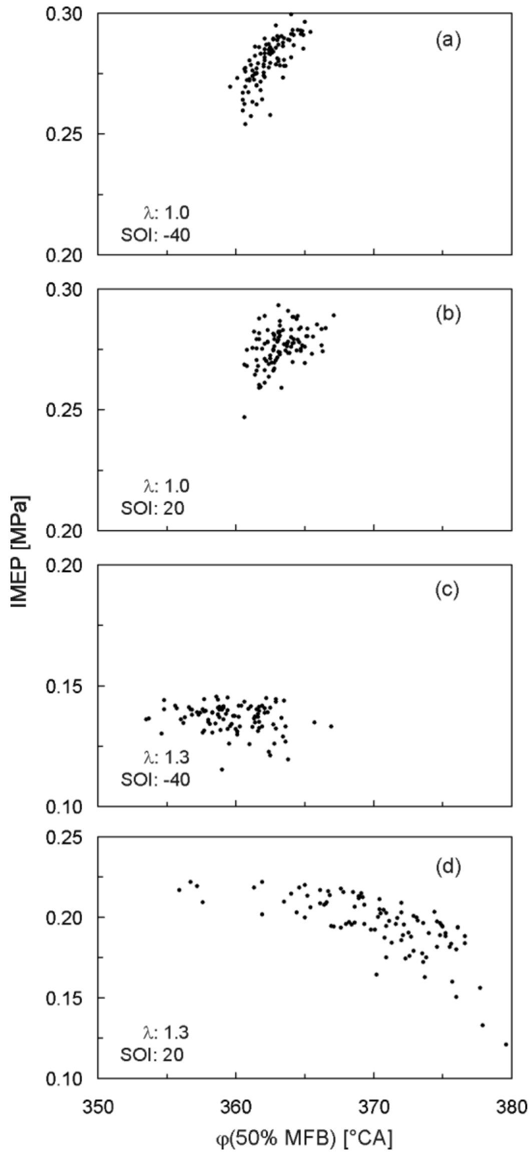

Fig. 4. IMEP versus 50% of MFB for 100 engine cycles
Rys. 4. Średnie ciśnienie indykowane jako funkcja kąta wypalenia 50% dawki paliwa dla 100 cykli pracy silnika

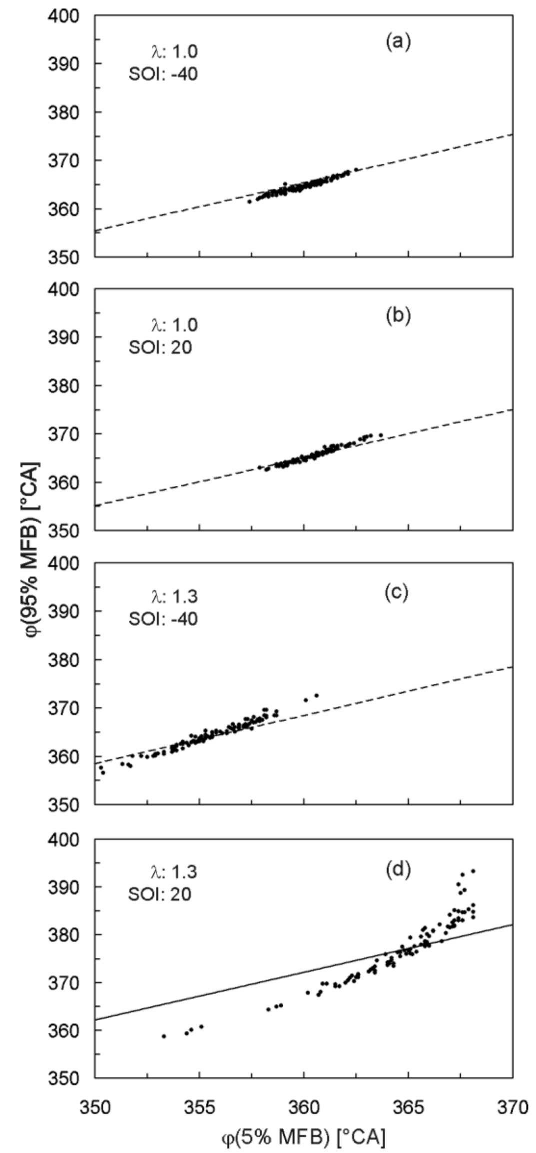

Fig. 5. 95% of MFB versus 5% of MFB for 100 engine cycles. Lines show 95% of MFB for mean combustion duration

Rys. 5. Kąt wypalenia 95% dawki paliwa jako funkcja kata wypalenia 5% dawki paliwa dla 100 cykli pracy silnika. Linie przedstawiają kąt wypalenia 95% dawki wynikający ze średniego kąta spalania

Figure 5 presents cycle-by-cycle relationships between combustion duration and auto-ignition timing. It can be seen that at low CoV in IMEP combustion duration is almost constant, independent of auto-ignition timing. Late auto-ignition causes slow combustion as shown in Fig. 5d. Thus too late end of combustion at low temperature deteriorates fuel conversion efficiency via flame quenching.

Variations in IMEP and its relation to combustion timing is directly related to start of combustion variability. If slow cycles do not take place, relatively low variations in IMEP are observed, even at higher variability of start of combustion, as shown in Fig. 6c. Slow cycles at low in-cylinder temperatures, which are characterized by excessive CO emission are always followed by early auto-ignition cycles. This can be associated with higher oxygen content in re-circulated exhaust and heat release during NVO period. NVO heat release increases compression temperature and reduces volumetric efficiency, thus influences air excess ratio as well. As combustion duration is determined by air excess in the combustible mixture, cycle-by-cycle variations are additionally affected by load exchange phenomena. The effect of late auto-ignition was observed also two cycles ahead the slow combustion cycle, as shown in Fig. 6d.

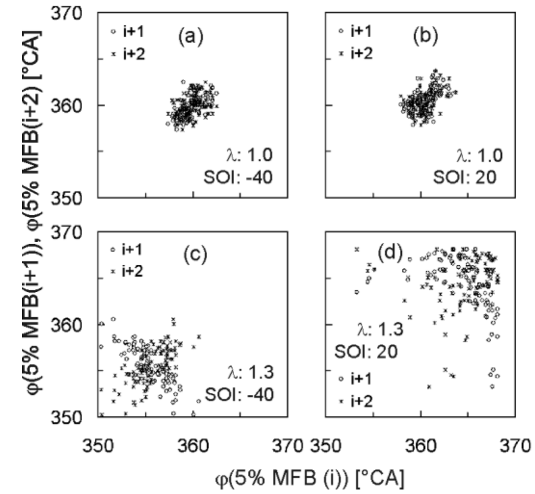

Fig. 6. Return maps of 5% of MFB (one and two engine cycles ahead current cycle)

Rys. 6. Kąt wypalenia 5% dawki paliwa w kolejnych dwóch cyklach jako funkcja jego bieżącej wartości

5. Conclusions

Cycle-by-cycle variability in CAI engine with internal EGR and direct gasoline injection was studied. The analysis was based on IMEP and combustion timing. The findings of the study are summarized below:

1. Cycle-by-cycle variations in IMEP reveal deterministic character at low loads and lean mixture.
2. Variations in IMEP at lean mixture are related to combustion timing. At low loads, the later combustion the lower IMEP values were observed. It was ascribed to combustion termination due to dropping in-cylinder temperature.
3. So called late combustion cycle is always followed by one or two high IMEP cycles due to CO utilization from the previous cycle.
4. CO content in trapped residuals and resulting oxygen presence influences amount of heat released during NVO, which determines intake temperature, volumetric efficiency, air excess ratio, and consequently – combustion timing.

Acknowledgements

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Nomenclature/*Skróty i oznaczenia*

CA Crank Angle/*kąt obrotu wału korbowego*
CAI Controlled Auto-Ignition/*kontrolowany samozapłon*
CoV Coefficient of Variation/*współczynnik zmienności*
EVC Exhaust Valve Closing/*zamknięcie zaworu wylotowego*
EVO Exhaust Valve Opening/*otwarcie zaworu wylotowego*
EGR Exhaust Gas Re-circulation/*recyrkulacja spalin*
HC Hydrocarbons/*węglowodory*
HRR Heat Release Rate/*szbkość wywiązywania się ciepła*
IMEP Indicated Mean Effective Pressure/*średnie ciśnienie indykowane*
IVC Intake Valve Closing/*zamknięcie zaworu dolotowego*
IVO Intake Valve Opening/*otwarcie zaworu dolotowego*
MFB Mass Fraction Burnt/*stopien wypalenia dawki paliwa*
NVO Negative Valve Overlap/*ujemne współotwarcie zaworów*
p In-cylinder Pressure/*ciśnienie w cylindrze*
SOI Start of Injection (angle)/*kąt początku wtrysku*
T_{EXH} Exhaust Temperature/*temperatura spalin*
TDC Piston Top Dead Center/*górne zwrotne położenie tłoka*
φ Crankshaft Position/*położenie wału korbowego*
η_V Volumetric Efficiency/*współczynnik napełnienia cylindra*
λ Air Excess Ratio/*współczynnik nadmiaru powietrza*

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