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Analysis of combustion pressure in a two-cycle engine, Otto and Diesel

Abstract: Results of investigations performed on a four-stroke internal combustion engine working both with spark ignition and auto-ignition from a fuel dose – according to the patent solution by B. Sendyka – were presented in this paper. The analysis concerned the process of pressure in the cylinder in function of crank angle at work with spark- and auto-ignition initiated from an ignition dose. Traces of changes of pressure rise rate in a function of crank angle and pressure increase in the work space of the engine - which is the result of the combustion process – were given for both ignition and combustion systems. During engine work with combustion initiation from an ignition dose increment of the indicated mean effective pressure as well as caloric and total efficiency was obtained. The results point at improvement of the combustion process in the case of combustion initiation from an ignition dose what influences the considerable increase in total efficiency of an engine as it in the case in engine with auto-ignition.

Key words: *two-cycle internal combustion engine, compression ignition, spark ignition, increase in internal combustion engine total efficiency*

Analiza przebiegu ciśnień spalania w silniku dwuobiegowym, z zapłonem iskrowym i samoczynnym

Streszczenie: W artykule opisano wyniki badań prowadzonych na czterosuwowym silniku spalinowym, który ma możliwość pracy, zarówno z zapłonem iskrowym, jak też i samoczynnym od dawki zapłonowej paliwa według rozwiązania patentowego autorstwa B. Sendyki. Analizie poddano przebiegi ciśnienia w cylindrze w funkcji kąta obrotu wału korbowego dla przypadku pracy z zapłonem iskrowym oraz samoczynnym od dawki zapłonowej. Przedstawiono również przebiegi zmian szybkości przyrostu ciśnienia w funkcji kąta obrotu wału korbowego oraz przedstawiono wzrost ciśnienia w przestrzeni roboczej silnika będący wynikiem procesu spalania dla obywu systemów zapłonu i spalania. Podczas pracy silnika z inicjacją spalania od dawki zapłonowej uzyskano przyrost średniego ciśnienia indykowanego, jak też i sprawności cieplnej oraz ogólnej. Uzyskane wyniki wskazują na poprawę przebiegu procesu spalania w przypadku inicjacji spalania od dawki zapłonowej, co ma wpływ na znaczące podniesienie sprawności ogólnej silnika, jak to ma miejsce w silniku o zapłonie samoczynnym

Słowa kluczowe: *silnik dwuobiegowy, zapłon iskrowy i samoczynny, zwiększenie sprawności ogólnej silnika spalinowego*

1. Introduction

The scientific – research works whose results were given beneath were carried out within the framework of the project N N509 405036 financed by Ministry of Science and Higher Education of the Polish Republic. The project aimed at elaboration of a combustion system of engine working with two systems of fuel injection: basic – indirect, multipoint and ignition, system of direct fuel injection. It was assumed that engine will work with spark ignition and after its switching off the initiation of the combustion process will proceed automatically from the fuel dose injected directly into the cylinder. Starting of the engine and work on low load and lower rotational speed proceeded with action of the ignition system, whereas, at higher and maximal load and higher rotational speed deactivation of spark ignition will take place and activation of direct injection of the pilot dose initializing auto-ignition of the charge will follow. The aim of such activity was gaining increase in total engine efficiency and reduction of toxic component concentration in exhaust gasses. The scientific – research work described in the article aimed at determination of differences in the traces of the combustion process occurring in engine working in the two considered modes.

2. Investigation object

Results of investigations described in the following paper were carried out in the laboratory of the Chair of Internal Combustion Engines at Cracow University of Technology. The test bench was constructed basing on a naturally aspirated, four cylinder spark ignition engine Toyota 2SZ-FE of displacement 1.298 dm^3. The engine in its original version is equipped with a multipoint system of petrol injection into the intake ports. With regard to the needs of the project the cylinder head was modified so as to make it possible to install injectors of the ignition dose injected directly into each of the four combustion chambers. On the dynamometer test bench powertrain unit was connected with an eddy-current brake equipped with an electronic control – measurement block. The control system of the brake can communicate with the computer of the PC-class for visualization and recording of values of the measured quantities . Fig. 1 presents a general view of the test bench.

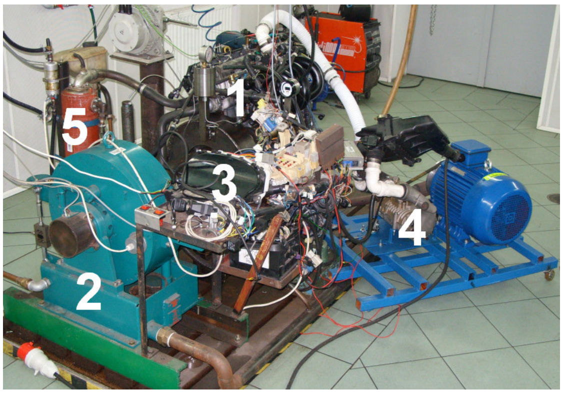

Fig. 1. General view of the test bench;
1 – Engine, 2 – Eddy current brake , 3 – Engine management system, 4 – Supercharging system, 5 – Water to water engine heat exchanger
Rys. 1. Widok ogólny stanowiska badawczego;
1 – Silnik, 2 – Hamulec elektrowirowy, 3 – System sterowania silnika, 4 – System doładowania, 5 – Wymiennik ciepła układu chłodzenia silnika typu woda - woda

In order to make high pressure injectors of direct injection system work properly the injectors were connected to the engine controller by a special electronic driver. This system permits supplying the injectors with increased voltage of about 100V and it has the function of limiting the supply current to the injector after its full opening.

In order to enable the test engine in which the compression ratio did not undergo any modification to work in the mode of combustion initiation from a pilot dose its intake system was equipped with a electric-power driven positive displacement compressor of Eaton-type. This was thought to increase the pressure and temperature in the final phase of the compression stroke. The supercharging system possesses an electronic system of rotational speed control of the compressor driving engine and this permits optional pressure regulation in the intake manifold.

3. Investigation method

Carrying out the analysis of the combustion process of application of spark ignition and during work with fuel mixture ignition from the pilot dose required indication of the work space of the examined engine i.e. measurement and recording of the instantaneous cylinder pressure with concomitant record of momentary angular position of crankshaft.

An electronic converter Optrand type C82255-SP served for measurements of instantaneous pressure in the cylinder. It was mounted in a redesigned spark plug. Record of the crankshaft position was obtained due to an incremental encoder of angular position Omron E6B-CWZ3E of resolution 360 impulses per revolution and a separate TDC-marker channel.

The signals from the pressure converter and from encoder of the angular position were recorded by means of a PC-class computer equipped with a data acquisition card National Instruments USB 6251 co-operating with a application developed in LabView software. Application has sub-module of identification of crankshaft position basing upon the encoder signal and the crank angle may be recalculated into an instantaneous value of the engine work space volume after introduction of its respective structural features. Moreover, this software apart from a record of data, permits also visualization in real time of indicator diagrams of the engine both in $p_c – \alpha$ and $p_c – V$ form. Registration of digital data was realized in a format permitting comfortable elaboration by use of the spreadsheet, especially of the applied MS Excel.

Apart from the process of combustion pressure in the cylinder and crank angle during measurements registration was carried out of standard parameters of work essential from the point of view of engine tests i.e. torque T, rotational speed n and instantaneous fuel consumption G_e, temperature T_{amb} and surrounding pressure p_{amb}.

4. Analysis of indicator diagrams and basic indices of engine work in both modes

Investigations whose results served for analysis of the combustion process were carried out for rotational speed n=2000 RPM at fully opened throttle. Similarly as in former investigations performed in such conditions the ignition advance angle was established at 14 ºCA before TDC, whereas, during work with auto-ignition from the ignition dose it started at 30 ºCA before TDC. The attempted absolute pressure in the intake manifold was 0.13 MPa during work with spark ignition, whereas, in the case of ignition from the pilot dose it equaled 0.14 MPa. As it was mentioned before increase in pressure in the intake manifold from 0.13 to 0.14 MPa caused automatic switch from spark ignition to combustion initiated by injection of a pilot dose. Injection pressure of the ignition dose was 15 MPa. The measured value of torque during work of engine with spark ignition was 106 Nm, whereas, after switching to auto-ignition it increased to 113 Nm. The fuel dose per one cycle was 0.0219g of petrol when the engine worked with spark ignition. During engine work with auto-ignition 0.0198g petrol was added per work cycle to the intake pipe of each cylinder, whereas, the value of pilot dose equaled 0.0015g. Quantities of particular fuel doses per one work cycle were determined on the basis of fuel consumption measured during engine work. Its value being G_e =5.256 kg/h during engine work with spark ignition and 5.112 kg/h jointly when the engine worked with auto-ignition initiated from a ignition dose injection.

P_c-α indicator diagrams recorded during work of engine with spark ignition and in the case of combustion initiation from the ignition dose were presented in Fig. 2.

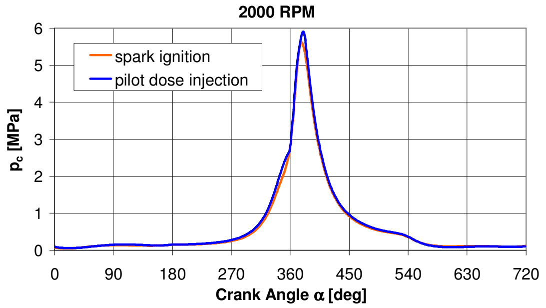

Fig. 2. P_c – α indicator diagrams obtained during work of engine with spark ignition and with combustion process initiation from an ignition dose at rotational speed 2000 RPM

Rys. 2. Otwarte wykresy indykatorowe uzyskane podczas pracy silnika z zapłonem iskrowym oraz z inicjacją procesu spalania od dawki zapłonowej przy prędkości obrotowej 2000 obr/min

On the basis of curves in Fig. 2 p_c – V_c indicator diagrams were made. They were obtained by calculating the instantaneous volume of the work space in function of crank angle with application of formula (1):

$$V_c = V_{cc} + \frac{\pi \cdot d_c^2}{4000}\left[r(1-\cos\alpha)+l-\sqrt{l^2-r^2\sin^2\alpha}\right] \quad (1)$$

The constant 4000 results from recalculation of applied units.

Fig. 3. shows the curves of pressure in the engine cylinder obtained by means of replacing the crank angle with an instantaneous volume of work space applying formula (1).

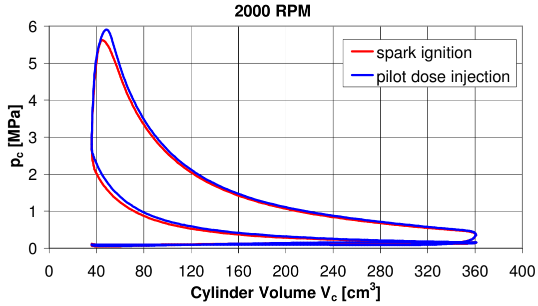

Fig. 3. Comparison of a closed indicator diagram obtained for the case of engine work with spark ignition and during work with ignition initiation from a pilot dose

Rys. 3. Porównanie zamkniętego wykresu indykatorowego uzyskanego dla przypadku pracy silnika z zapłonem iskrowym oraz podczas pracy z inicjacją zapłonu od dawki pilotującej

The above given diagrams were obtained on the way of approximation by use of spline of traces averaged from 10 subsequent cycles obtained during measurements. Without this procedure correct determination of fuel mixture combustion would be impossible with regard to noise overlapping the traces of pressure in the cylinder registered from the gauge. The method of combustion velocity appraisal applied in the next step of investigations requires application of undisturbed traces of indicated pressure. Approximation of real indicator traces was performed by use of the spreadsheet MS Excel. For this purpose the functions in form of polynomials of various degrees were mainly used, whereas, fragments of the indicator diagram proper for compression and expansion were approximated by traces of polytrophic $p_c \cdot (V_c)^k$ = const. Values of polytrophic index were determined from p_c – V_c indicator diagrams of the engine presented in logarithmic polar co-ordinates. These were shown in Table 1.

Tab. 1. Values of polytrophic indexes approximating real traces of process of compression and expansion in both work modes

Tab. 1. Wartości wykładników politrop przybliżających rzeczywisty przebieg ciśnienia podczas procesu sprężania i rozprężania dla obu trybów pracy

Engine work mode	Compression polytrophic index	Expansion polytrophic index
Spark Ignition	1,205	1,295
Pilot dose injection	1,228	1,256

In the two presented above diagrams and specially in Fig. 3 increment of the field of the diagram area representing the positive work of the engine work cycle is noticeable. Maximal pressure during the combustion process reached the value 5.62 MPa for 17 ºCA after TDC at work of engine with spark ignition and 5.90 MPa for 19 ºCa after TDC at ignition from a pilot dose. Top combustion pressure is, thus, higher in the case of work at fuel ignition dose of 0.28 MPa as compared with the result obtained during work with spark ignition.

Fig. 4 presents the traces of cylinder pressure increment δp_c in function of crank angle in the case of work of the engine with ignition from pilot dose in comparison with pressure in the cylinder at spark ignition.

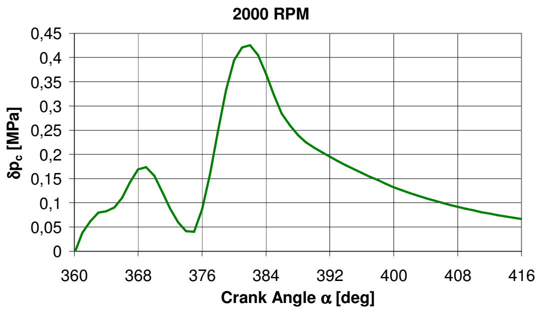

Fig. 4. Traces of indicated pressure increment δp_c in function of crank angle in the case of work of engine with ignition from a pilot dose in comparison with pressure in the cylinder at spark ignition

Rys. 4. Przebieg przyrostu ciśnienia indykowanego δp_c funkcji kąta obrotu wału korbowego w przypadku pracy silnika z zapłonem od dawki pilotującej w porównaniu do ciśnienia w cylindrze przy zapłonie iskrowym

In order to determine more precisely the differences resulting from the traces of the obtained indicator diagrams the indicated mean effective pressures IMEP for both cases respectively were calculated basing upon the recorded data. The method of numerical integration of respective areas of the diagrams in Fig. 3 in the spreadsheet was applied. To ensure a higher accuracy of calculations the rule of trapezium of the height dV_c and basis equaling respective values of the function $p_c = p_c(V_c)$ was used. The calculation results were given in the Table 2.

The brake mean effective pressure BMEP was determined on the basis of formula (2):

$$BMEP = \frac{\pi \cdot \tau \cdot T}{500 V_{ss}} \qquad (2)$$

Engine type τ factor equals 2 in the case of four-stroke engine. Correction factor equaling 500 results from applied measure units.

Whereas, on the basis of relation (3) it was possible to calculate the value of caloric efficiency of the engine in both cases. Basing upon [4] the lower heating value of fuel equaling 43000 kJ/kg was assumed.

$$\eta_c = \frac{N_i}{N_c} = \frac{30 \cdot IMEP \cdot V_{ss} \cdot n}{G_e \cdot LHV} \qquad (3)$$

Constant equaling 30 results from recalculation of measure units applied in the formula.

Results of calculations of the mean effective pressure and caloric efficiency performed by use of equations (2), (3) and total efficiency calculated by use of equation (4) were presented in Table 2.

$$\eta_{tot} = \frac{T \cdot n}{2{,}6528 \cdot G_e \cdot LHV} \qquad (4)$$

In equation (4) the value of the multiplier in the denominator is the result of recalculation of the applied measure units.

Tab. 2. Comparison of engine work indices in both modes

Tab. 2. Porównanie wskaźników pracy silnika w obu trybach

	Spark Ignition (SI)	Pilot Dose Injection (CI)	Increment respect to SI, [%]
BMEP, [MPa]	1,026	1,094	6,63
IMEP, [MPa]	1,194	1,271	6,45
η_c, [-]	0,411	0,450	9,49
η_{tot}, [-]	0,354	0,388	9,60

During engine work with ignition injection a 6.45% increment of mean indicated mean effective pressure and 9.49% increment of caloric efficiency as compared with spark ignition engine work was obtained. This results in improvement of total efficiency of the engine by 9.60% being the most measureable effect of carried out treatments. Basing on the above given results it may be stated that increase in the indicated mean effective pressure and caloric efficiency points at improvement of fuel mixture combustion is initiated by a fuel dose injected directly into the cylinder.

The pressure rise rate $dp_c/d\alpha$ was considered as the last work index in this part of analysis of indicator traces of the engine. The diagram presenting changes of this quantity in a function of crank angle limmited to the most essential part of the indicator diagram was presented in Fig. 5.

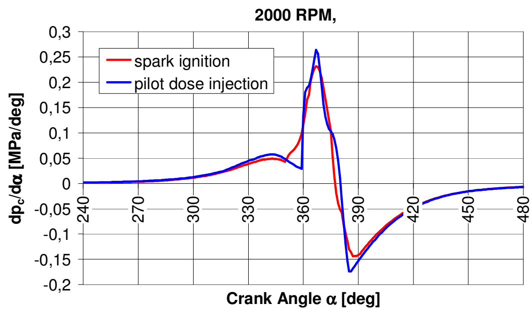

Fig. 5. Traces of pressure rise rate in a function of crank angle obtained for both combustion systems

Rys. 5. Przebiegi zmian szybkości przyrostu ciśnienia w funkcji kąta obrotu wału korbowego uzyskane dla obu rozpatrywanych systemów spalania

The pressure rise rate in the main phase of combustion process is adopted as the most important criterion indicating as the possibility of knock combustion onset.

The obtained results point at some increase in the pressure rise rate in the case of work of engine with ignition from a pilot dose. The highest value of this index was 0.227 MPa/°CA during work with spark ignition and 0.264 MPa/°CA when the engine worked with combustion initiation from pilot dose auto-ignition. Increase in pressure increment rate in not a wanted phenomenon since it causes among others, increased load in the cranktrain, nevertheless value obtained during work with ignition from pilot dose is not high. It should be mentioned that onset of knock combustion is characterized by occurrence of the peak of the pressure rise rate mostly higher than 0.5 MPa/°CA [2].

5. Conclusions

On the basis of the investigation results presented in this paper following conclusions can be drawn:

1) At application of a two-cycle engine Otto and Diesel the pressure traces in the cylinder proceed in a positive way what causes increase in the torque at changing from Otto cycle to Diesel cycle,
2) In result of a positive process of the combustion at ignition initiation from an ignition dose increase in the indicated mean effective pressure takes place what in turn causes increase in caloric and total efficiency,
3) A more positive process of combustion pressure consist in the fact that in the initial phase of the combustion process from an ignition dose increase in combustion velocity takes place, hence, pressure increment rates are more advantageous.

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Nomenclature/*Skróty i oznaczenia*

α Crank Angle [°]/*kąt obrotu wału korbowego*
δp_c cylinder pressure increment [MPa]/ *przyrost ciśnienia w cylindrze*
η_c caloric efficiency [-]/*sprawność cieplna*
η_{tot} total efficiency [-]/*sprawność ogólna*
BMEP Brake Mean Effective Pressure/*średnie ciśnienie efektywne*
CA Crank Angle/*kąt obrotu wału korbowego - OWK,*
CI Compression Ignition/*zapłon samoczynny*
d_c bore [mm]/*średnica cylindra*
$dp_c/d\alpha$ pressure rise rate [MPa/°CA]/*szybkość przyrostu ciśnienia*
G_e instantenous fuel consumption [kg/h]/ *godzinowe zużycie paliwa*
IMEP Indicated Mean Effective Pressure/*średnie ciśnienie indykowane*
k polytrophic index [-]/*wykładnik politropy*
l connecting rod lenght [mm]/*długość korbowodu*
n rotational speed/*prędkość obrotowa*
N_i indicated power [kW]/*moc indykowana*
N_c heat flow from fuel combustion [kJ/s]/ *strumień ciepła dostarczany od spalania paliwa*
p_{amb} ambient pressure [hPa]/*ciśnienie otoczenia*
p_c cylinder pressure [MPa]/*ciśnienie w cylindrze*
r crank radius [mm]/*promień wykorbienia*
SI Spark Ignition/*zapłon iskrowy*
T Torque [Nm]/*moment obrotowy*
T_{amb} ambient temperature [°C]/*temperatura otoczenia*
TDC Top Dead Center/*górne martwe położenie - GMP*
V_c cylinder work space volume [cm^3]/*objętość przestrzeni roboczej cylindra*
V_{cc} combustion chamber volume [cm^3]/*objętość komory spalania*
LHV Lower Heating Value of fuel [kJ/kg]/*dolna wartość opałowa paliwa*

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