

Determining selected diesel engine combustion descriptors based on the analysis of the coefficient of variation of in-chamber pressure

A. BAŁKOWSKI* and L. RADZISZEWSKI

Department of Mechatronics and Machine Design, Kielce University of Technology,
7 Tysiąclecia Państwa Polskiego Ave., 25-314 Kielce, Poland

Abstract. This paper presents a method that uses the coefficient of variation (COV) of pressure in a diesel engine combustion chamber to determine the crank angle degree (CAD) for which the heat release rate (HRR) reaches the maximum value. The COV was proposed for determining the point corresponding to the angle of start of combustion (SoC). Regression models were fit with these descriptors for the engine powered by diesel, biodiesel or a combination of both, operating under full- or part- load conditions. The uncertainty parameter in these models was determined. Good agreement between the experimental results and the literature data shows the validity of the analysis.

Key words: diesel engine, coefficient of variation, measurement uncertainty, heat release rate, start of combustion.

1. Introduction

For social reasons mainly, advances in internal combustion engine design technology rely on a potential to reduce the amounts of toxic exhaust components and to lower fuel consumption and noise levels [1, 2]. Primary development effort is focusing on designing engines and fuel systems that can help optimize combustion processes [3, 4] and additional effort is put toward introducing new fuel types and fitting engines with catalytic converters and filters. The application of selected simple descriptors of pressure signal in the combustion chamber [5, 6] helps conform to social demand and expectations. The measurement and analysis of pressure dynamics may be used to control the fuel supply of individual cylinders [7] thus improving their work with respect to performance and toxic emissions [8]. Also, by measuring the CAD and the pressure in the combustion chamber (p_c), real-time values can be calculated of parameters such as mean indicated pressure, maximum pressure, CAD at the maximum pressure and CAD for which 50% of the fuel feed has been burnt. These parameters are used for controlling the work of each cylinder in each working cycle and for diagnostic purposes. The calculated parameter values are passed to the system, which implements the combustion control algorithms and controls the actuators [8]. Striving to maintain the same optimal values of the combustion parameters in each cylinder results in difficulties in the injection system control. Individual, varying attributes and parameters of injectors as well as different properties of fuels used to power diesel engines make this control even more complicated.

That is why the values of the signals used for controlling should be known to the least uncertainty. Stochastic in most cases, these signals are usually considered to have normal distributions [9]. The study conducted by the authors of this

paper suggests that these assumptions do not always hold [10]. Random noise that occurs in combustion engines operating in cycles is a major factor in cycle-to-cycle variability, even at steady state operation.

The engines fitted with mechanically controlled fuel injection pump are still used in off-road vehicles designed to work under extreme conditions. The benefit of these engines lies in their capacity to run on cheaper fuels, including bio-fuels [11]. The disadvantage is in their hindered cooperation with the engine exhaust after-treatment systems. The high reliability of such fuel injection systems makes warranted the modernization that allows multiphase fuel injection at a high pressure and the use of elements of traditional fuel injection systems.

Charge cycle fluctuations resulting from the instability of fuel, air and residual gas in the cylinder cause the so-called cyclic unrepeatability [12]. The parameters that characterize this unrepeatability of the combustion engine working cycle fall into four groups related to pressure distribution in the cylinder, heat release, flame front development and exhaust fumes removal.

The parameters associated with in-cylinder pressure are used most often mainly due to the simplicity of conducting experiments. The maximum pressure shows the highest unrepeatability. The peak pressure angle is the most adequate cyclic unrepeatability indicator in relation to the flame front development. The unrepeatability of the mean indicated pressure is the most effective way of describing variation in the entire range of the combustion process. The combustion values are determined following the analysis of pressure charts and the use of a thermodynamic model of heat release. Uncertainty of these values is dependent, to a high degree, on the measurement accuracy of in-cylinder pressure and the CAD lines [13].

*e-mail: abakowski@tu.kielce.pl

According to [14], the combustion engine working cycle can be evaluated using different signal descriptors. The variation of the standard deviation of the pressure signal in the combustion chamber is one of the descriptors [15, 16]. This paper presents the analysis of this descriptor as applied to determining selected characteristic points in the work of the combustion engine fueled with diesel and biofuels.

2. Experimental facilities

The tests were carried out on the Perkins engine AD3.152 UR [17]. Table 1 summarizes the selected technical parameters of this engine [18].

Table 1
Technical parameters of Perkins engine AD3.152 UR

Type of injection	Direct
Number of cylinder	3
Compression ratio	16.5
Cylinder bore	91.44 mm
Displaced volume	2.502 dm ³
Maximum engine power	34.6 kW
Maximum power rotational speed	2250 rpm
Maximum engine torque	165.4 Nm
Maximum torque rotational speed	1300–1400 rpm
Static angle of injection advance	17°

During the tests, pressure p_c was measured using the piezoelectric sensor AVL QC34D. The engine was coupled with a 132 kW dynamometer [17]. The parameters measured were recorded as a function of the crank angle, with a resolution of 1.4°, which gave 512 measurement points for one working cycle of the engine. The values obtained from 50 full engine cycles were recorded for all the working conditions. The scope of the tests covered the work of the engine under full- and part- load characteristics for loads from 4 to 20 kW, at speeds from $n = 1000$ to 2000 rpm. In both cases the engine was fueled with diesel, biofuel FAME (methyl esters of fatty acids), and a combination of both (B20).

3. Analysis of the variation of the standard deviation of the pressure signal

The results of the literature review indicate that determining the position of points that characterize the charge combustion process (in-chamber pressure variation) using simple and accurate methods has a considerable influence on the correct control of engine working cycle. The point at which the pressure with combustion line separates from the compression line is often considered to mark the start of combustion. The point of inflexion of these curves may be apparent at maximum load of the engine and hardly apparent under low loads [19]. The methods of determining the position of this point by analyzing the change in the first derivative value of the pressure curve relative to the CAD are not simple and do not guarantee the required accuracy. An additional complication is the fact that the equation for combustion pressure is not known. Thus, other and simpler methods should be found for

determining the position of points such as a point of start of combustion or a point at which the HRR reaches its maximum value.

The experimental study results show some irregularity in the recorded variable values (both maximum values and those in the function of the CAD) over one and many engine working cycles. To evaluate the irregular graphs, the analysis of the COV is used [20]. The COV for the in-chamber pressure can be defined using the following relationships

$$COV_{p_c}(\alpha) = \frac{\sigma_{p_c}(\alpha)}{\overline{p_c}(\alpha)}, \quad (1)$$

when it is calculated for the specified value α of the CAD, or

$$COV_{F_{p_c}}(\Delta \alpha) = \frac{\sigma_{F_{p_c}}(\Delta \alpha)}{\overline{F_{p_c}}(\Delta \alpha)} \quad (2)$$

when its value is determined within the adopted range of CAD. The COV is then a relative, dimensionless measure of the variable dispersion. This coefficient can also be used when there are no grounds for rejecting the hypothesis about the agreement between the distribution of the investigated variable and the normal distribution. In [10], the authors studied the stationarity of the in-chamber pressure, the injection pressure and the maximum needle lift, and the consistency of their maximum values distribution with the normal distribution. It was proved that the maximum values of all the measured variables could be considered stationary signals. Statistical tests helped recognize the distributions of the maximum values of in-chamber and injection pressures as consistent with normal distribution.

These tests (in some cases) provided grounds for rejecting the null hypothesis about the consistency of the needle lift maximum value distribution with their normal distribution. Coefficient COV_{p_c} was used in the analysis aimed to evaluate the fluctuations of the measured variables. On the basis of the recorded cycles, the coefficient described by relationship (1) can be determined for every CAD, for which the curve variability we find interesting.

Relationship (2) is a measure of p_c variability over the adopted range of the CAD on the basis of the surface area (between variable p_c and axis of abscissa) obtained from subsequent engine working cycles. For example, the surface area for the working cycle j can be determined as $F_{p_c}(j) = \int_{\alpha_1}^{\alpha_2} p_c(j) d\alpha$, where α_1 – the angle corresponding to the inlet valve closing, α_2 – the angle corresponding to the exhaust valve opening, are the examples of range boundaries over which, the variation of p_c is being investigated.

Other denotations: $\overline{F_{p_c}} = \frac{1}{m} \sum_{j=1}^m F_{p_c}(j)$ – the average value of the surface area under the diagram $p_c = f(\alpha)$, $\sigma_{F_{p_c}}$ – the standard deviation of the surface area value under the diagram, m – the number of measurement cycles.

A change in the crank speed affects the values of $COV_{F_{p_c}}(\Delta \alpha)$. Over the crank angle range from 223° to 494°, the coefficient decreases with an increase of the crank

angle speed. These changes were within the range from 0.034 for $n = 1200$ rpm to 0.012 for $n = 2000$ rpm. It has to be noted that the values of COV for maximum pressures in the combustion chamber were approximately 0.012 regardless of the crank angle speed. In both cases, the engine was fueled with diesel and worked under full-load conditions.

Examples of coefficients COV of in-chamber pressure are presented in Fig. 1. Graph 1a summarizes the changes in the values of COV_{p_c} associated with instantaneous parameters dependent on the crank angle (continuous line – A), and coefficients COV_{Fp_c} , which depend on the selected crank angle range (dashed line F).

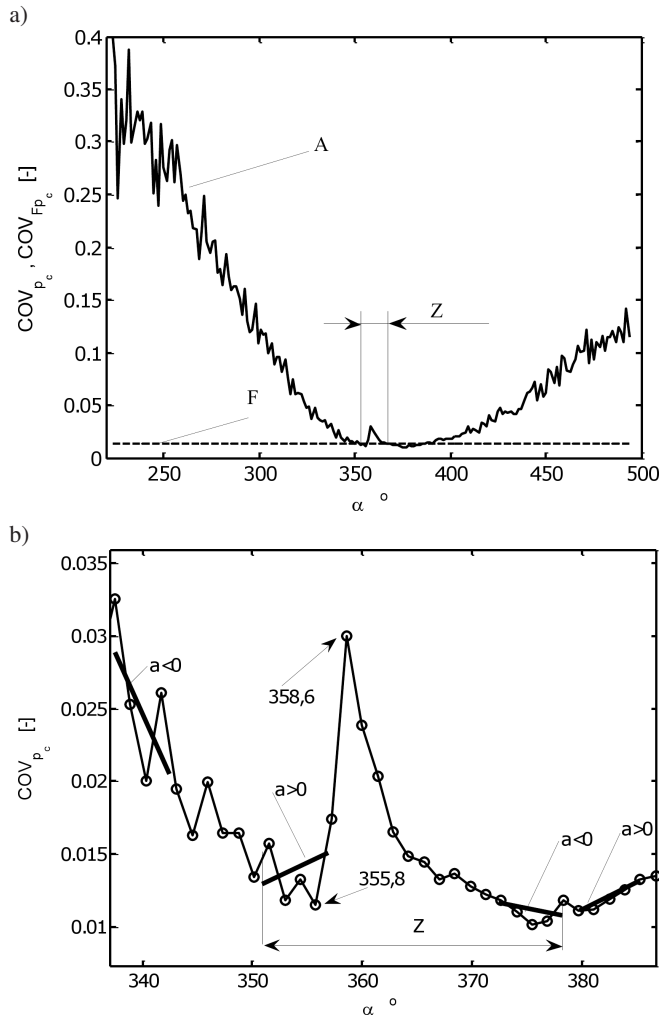


Fig. 1. COV of pressure in combustion chamber as a function of CAD; diesel fueled engine, operating under full-load condition $n = 1800$ rpm, a) within the crank angle range from 223 to 494°, b) determination of CAD corresponding to the points: start of combustion and maximum value of HRR. A – plotted based on dependence (1), F – plotted based on dependence (2)

The plot of COV_{p_c} in Fig. 1 gives rise to a presumption that this parameter can also be used to determine CAD associated with the combustion process. Attention should be paid to large changes in COV_{p_c} within the crank angle range from 223° to 494° (at closed inlet and exhaust valves). These

changes are irregular and significant, in particular within the crank angle range from 223° to 300°. Occurring in a closed chamber before the injection process starts, the changes may result from a variety of small disturbances in the gaseous media composed of air and residual exhaust fumes, caused by the piston movement recorded by a sensitive piezoelectric transducer. In the crank angle range from 350° to 380° the COV_{p_c} lies in the range from 0.010 to 0.030. The authors assumed arbitrarily that the angle from which the COV_{p_c} rises dramatically leading to the local extreme (under the studied engine operation conditions, the angle was $\alpha_{ps} = 355.8^\circ$ – Fig. 1b) represents the SoC point. The value of COV_{p_c} increases due to a rapid pressure increase resulting from the start of the charge combustion process. Compared with other methods such as analysis of the trajectory of the pressure first derivative relative to the CAD [21] or an acoustic method [14], this method of determining the position of the charge combustion starting point is easier. The angle for which the coefficient reaches its local peak ($\alpha_{\dot{Q}_{MAX}} = 358.6^\circ$ – Fig. 1b) corresponds, in the authors' view, to the point at which the HRR reaches its peak.

The procedure for determining $\alpha_{\dot{Q}_{MAX}}$, involves establishing the CAD, for which the $COV_{p_c}(\alpha)$ determined in the Z range defined by the common portion of straight line F and curve A (Fig. 1a) reaches the maximum value. The duration of the calculations run through MATLAB, from the moment of recording the experimental results to the determination of the sought value of $\alpha_{\dot{Q}_{MAX}}$ was approximately 0.09 seconds. Analysis of the results indicated, however, that this procedure is not a universal method. For some operating conditions (33% for diesel fueled and 16% for FAME fueled engine) there is no common portion defined by straight line F and the curve A.

Therefore the authors developed a novel procedure that allows determining the angle interval, Z (Fig. 1b), with local maximum $COV_{p_c}(\alpha)$, and calculating the CAD corresponding to this extreme. Linearization of $COV_{p_c}(\alpha)$ is used in subsequent subintervals of the crank angle. The length of the subintervals is dependent on the value of m (Fig. 1b). In the Z interval containing local extreme $COV_{p_c}(\alpha)$, the slope a takes a reversed sign. To determine the beginning of the Z interval, the slopes in the linearized subintervals have to be calculated for the points from $n = 1$ to $n + m$, where n is the control variable. For the descending part of the $COV_{p_c}(\alpha)$ plot, these coefficients are negative (Fig. 1b). The value of the control variable n , for which slope a is positive, defines the index of the crank angle which is the beginning of the sought interval Z.

The end of interval Z is found in the analogous way. The angle that corresponds to the highest value of $COV_{p_c}(\alpha)$ in the determined interval Z is the angle for which HRR reaches its maximum. Part of the algorithm that shows the method of determining the beginning of the interval Z (Fig. 1b) is shown in Fig. 2. The value of m – the length of the linearized intervals- should be selected according to the crank angle sampling time. For purposes of the calculations the authors assumed that $m = 9$. Too high or too low value of m

may lead to flaws in the procedure. Individual analysis should be conducted to solve this problem, according to the sampling time of the measured signals. It has to be noted that the implementation of $COV_{pc}(\alpha)$ based descriptor algorithms is quite simple and there is no need for supercomputing.

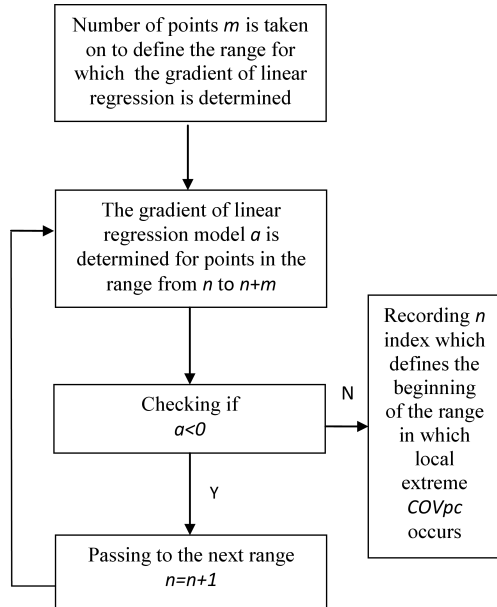


Fig. 2. Fragment of the algorithm showing the method of determining the Z interval from Fig. 1b

Computational time on the order of 0.25 seconds is needed to determine $\propto \dot{Q}_{MAX}$ using the procedure above. This procedure helps measure $COV_{pc}(\alpha)$ within the crank angle range from 223° to 494° and does not require user's interference. In order to reduce the computational time, the calculations can be carried out for a narrower range of the crank angle values. An example procedure time for the range from 300° to 400° was 0.15 s.

The crank angle values, corresponding to the moments at which combustion started and the HRR reached its maximum value, were compared with the values found in the literature [17]. In [17], the points were found with the spline method and complex thermodynamic models, which, while accurate, require making lots of time-consuming calculations.

The values of $\propto \dot{Q}_{MAX}$ determined based on the analysis of changes in COV_{pc} were compared to the theoretical values. For that purpose the authors developed a model of combustion HRR described by equations 3 to 13.

Instantaneous in-cylinder gas volume can be expressed by the following dependence [22]:

$$V = V_S - V_{SX} + V_\varepsilon + V_Z + V_{Px} \quad (3)$$

where V_S – displaced volume, V_{SX} – cylinder volume corresponding to the distance traveled by the piston from the bottom dead centre, V_ε – combustion chamber geometrical volume, V_Z – cylinder volume change resulting from the wear of parts and the assembly, V_{Px} – apparent change in cylinder volume due to blow-by.

Dividing expression (3) by volume V_S we obtain the following dimensionless form:

$$\vartheta = 1 - \vartheta_{SX} + \vartheta_\varepsilon + \vartheta_Z + \vartheta_{Px}, \quad (4)$$

where

$$\vartheta_{SX} = \frac{V_{SX}}{V_S} = \frac{\pi r^2 x}{\pi r^2 2R} = \frac{x}{2R} = S_x,$$

R – crank radius, r – cylinder radius, x – distance traveled by the piston, S_x – dimensionless distance traveled by the piston.

Assuming that: $\vartheta_{XS} = 1 - \vartheta_{SX}$, and $\vartheta_{CX} = \vartheta_\varepsilon + \vartheta_Z + \vartheta_{Px}$ equation (3) can be written:

$$\vartheta = \vartheta_{XS} + \vartheta_{CX}$$

compression curve model is assumed to have the form:

$$p_c \vartheta_x^m = const = A, \quad (5)$$

where the compression curve exponent can be described using the polynomial:

$$m_x = m_0 + m_1 S_x + \dots + m_a S_x^a.$$

Coefficients: $m_0, m_1 \dots m_a$ are determined through the approximation of pressure in the range for which compression occurs. For degree $a = 0$, the model becomes a polytrophic model. The distance traveled by the piston can be represented as:

$$x = R \left[1 - \cos \alpha + \frac{\lambda}{2} \sin^2 \alpha \right],$$

where $\lambda = \frac{R}{L}$, L – connecting rod length.

Let us assume that $\lambda^2 \ll 1$ and:

$$S_x = \frac{x}{2R} = \frac{1}{2} - \frac{1}{2} \cos \alpha + \frac{\lambda}{4} \sin^2 \alpha. \quad (6)$$

The equation for the first law of thermodynamics can be written:

$$dU = dQ - dW - \sum_i dm_i h_i, \quad (7)$$

dU – change in internal energy of charge, dQ – heat transferred to the system, dW – work done by the system, dm_i – amount of substance exchanged by the system boundary, h_i – specific enthalpy of the substance.

Mass exchange with the surrounds may occur due to the inflow and outflow of the agent through the open valves, due to the injection of fuel into the combustion chamber, the flow from the combustion chamber and back and the charge loss through the closed rings (blow-bys) and valves. Equation (7) differentiated in the time domain can be written as:

$$\frac{dU}{dt} = \frac{dQ}{dt} - p_c \frac{dV}{dt} + \sum_i \frac{dm_i}{dt} h_i. \quad (8)$$

The heat sum resulting from the mass exchange is positive in the equation above, but it is negative when it leaves the system. The heat supplied to the system can be expressed as:

$$dQ = dQ_{SP} - dQ_{Ch},$$

where dQ_{SP} – heat released as a result of fuel combustion, dQ_{Ch} – heat from cooling.

Let us transform equation (7) into the following relation:

$$dQ - \sum_i dm_i h_i = dU + dW$$

Determining selected diesel engine combustion descriptors based on the analysis of the coefficient of variation...

and introduce the following notation:

$$dQ_r = \sum_i dm_i h_i \quad \text{and} \quad dW = p_c dV.$$

Then we can write:

$$dQ_{SP} - dQ_{Ch} - dQ_r = dQ_n.$$

The expression above defines the net heat released, which can be written as:

$$dQ_n = dU + p_c dV = d(mu) + p_c dV, \quad (9)$$

where m – charge mass, u – specific internal energy.

Differentiation of equation (9) gives:

$$dQ_n = u dm + m du + p_c dV.$$

Because charge losses were taken into account in the balance defining the net released heat, then charge losses du result from the process of fuel injection and combustion. To determine Q_n , two factors are needed, fuel injection characteristics and the dynamic model of the fuel combustion. Internal energy of the charge is calculated by assuming that it is a linear function of the charge relative temperature. The composition of gases, dependent on the processes of fuel injection and combustion, also affects the internal energy. Taking into account the effect of these factors on the heat release process leads to complex models. For the ideal gas, the net released heat equation (from the first law of thermodynamics) has the form [23]:

$$dQ_n = \kappa(\kappa - 1)^{-1} p_c dV + (\kappa - 1)^{-1} V dp_c, \quad (10)$$

κ – isentropic exponent, V – momentary in-cylinder gas volume.

In fact, the value of the isentropic exponent κ is not constant but depends on temperature (in the range of 300–2800 K linear approximation) and on the processes of injection and combustion of fuel. To determine the net released heat in the first approximation we can assume κ to be a constant. Let us divide the value of the net released heat by the cylinder volume displaced V_s , and then differentiate this expression with respect to the crank angle α and finally write it in the form:

$$\frac{dq}{d\alpha} = (\kappa - 1)^{-1} \left[\kappa p_c \left(\frac{d\vartheta_{XS}}{d\alpha} + \frac{d\vartheta_{CX}}{d\alpha} \right) + (\vartheta_{XS} + \vartheta_{CX}) \frac{dp_c}{d\alpha} \right]. \quad (11)$$

Maximum net heat release rate occurs when:

$$\begin{aligned} \frac{d^2q}{d\alpha^2} &= \frac{d}{d\alpha} \left\{ (\kappa - 1)^{-1} \left[\kappa p_c \left(\frac{d\vartheta_{XS}}{d\alpha} + \frac{d\vartheta_{CX}}{d\alpha} \right) + (\vartheta_{XS} + \vartheta_{CX}) \frac{dp_c}{d\alpha} \right] \right\} \\ &= (\kappa - 1)^{-1} \left\{ \kappa \frac{dp_c}{d\alpha} \left(\frac{d\vartheta_{XS}}{d\alpha} + \frac{d\vartheta_{CX}}{d\alpha} \right) + \kappa p_c \left(\frac{d^2\vartheta_{XS}}{d\alpha^2} + \frac{d^2\vartheta_{CX}}{d\alpha^2} \right) + \left(\frac{d\vartheta_{XS}}{d\alpha} + \frac{d\vartheta_{CX}}{d\alpha} \right) \frac{dp_c}{d\alpha} + (\vartheta_{XS} + \vartheta_{CX}) \frac{d^2p_c}{d\alpha^2} \right\} = 0. \end{aligned}$$

Taking into account that $\vartheta = 1 - S_x + \vartheta_{CX}$ and that ϑ_{CX} and its first and second derivatives relative to α are close to zero, simple transformation lead to:

$$\begin{aligned} \frac{d^2q}{d\alpha^2} &= -\frac{dp_c}{d\alpha} \left[\kappa \frac{dS_x}{d\alpha} + \frac{dS_x}{d\alpha} \right] \\ &+ \frac{d^2p_c}{d\alpha^2} (1 - S_x) - \kappa p_c \frac{d^2S_x}{d\alpha^2} = 0. \end{aligned} \quad (12)$$

Taking into account relation (6), we obtain:

$$\begin{aligned} \frac{d^2q}{d\alpha^2} = 0 &= -\frac{dp_c}{d\alpha} \left[\kappa \frac{1}{2} \sin \alpha + \kappa \frac{\lambda}{2} \sin \alpha \cos \alpha + \frac{1}{2} \sin \alpha + \frac{\lambda}{2} \sin \alpha \cos \alpha \right] \\ &+ \frac{d^2p_c}{d\alpha^2} \left[\frac{1}{2} + \frac{1}{2} \cos \alpha - \frac{\lambda}{4} \sin^2 \alpha \right] \\ &- \kappa p_c \left[\frac{1}{2} \cos \alpha + \frac{\lambda}{2} \cos^2 \alpha - \frac{\lambda}{2} \sin^2 \alpha \right]. \end{aligned} \quad (13)$$

Expression (13) was used here to compute a theoretical angle for which HRR reaches its maximum. The equation for determining $\frac{d^2q}{d\alpha^2} = f(\alpha)$ was solved numerically. The first and the second derivatives of p_c with respect to angle α were determined based on the averaged pressure p_c calculated from 50 measurement cycles. MATLAB was used to make the calculations. Examples of $\frac{d^2q}{d\alpha^2} = f(\alpha)$ for a diesel fueled engine operating under full-load conditions ($n = 1400$ rpm or $n = 1800$ rpm) are shown in Fig. 3a and b.

Angles for which the HRR reaches its maximum, determined based on the curves in Figs. 3a and 3b are 358.32° for $n = 1400$ rpm and 360.78° for $n = 1800$ rpm.

Figure 4 graphically summarizes the values of angles corresponding to the combustion start or the maximum HRR for a diesel engine operating under full-load conditions. The following notation is used in Fig. 4a for the combustion start angle: α_{ps} – the angle calculated based on the coefficient of variation $COV_{p_c}(\alpha)_{Lit}$ – the angle reported in [17], determined using the spline method. In Fig. 4b, the position of the point at which the maximum HRR occurs is denoted as follows: $\alpha_{\dot{Q}_{MAX}}$ – the angle determined based on the coordinate of the local maximum coefficient of variation $COV_{p_c}(\alpha)$, $\frac{d^2q}{d\alpha^2}$ – the angle calculated after solving equation (13), Lit. – the angle reported in [17]. The results presented in Fig. 4 show a satisfactory agreement between the values of angles determined based on coefficients COV_{p_c} , and the results calculated theoretically and reported in the literature.

Figure 4 also presents the models of linear regression $\alpha = f(n) = a \cdot n + b$ analyzed descriptors. Table 2 summarizes the parameters of these models and the standard uncertainties (parameters u_a and u_b) of determination. For each model, standard deviation was calculated as a measure of the regression model error:

$$\sigma_e = \sqrt{\frac{\sum_{i=1}^n (\alpha_i - \widehat{\alpha}_i)^2}{n-2}}, \quad (14)$$

where α_i – values obtained based on the experimental study results, $\widehat{\alpha}_i$ – values calculated based on the regression model.

From this data we can see that the values of the angles increase with increasing rotational speed. The values of the combustion start angles, determined based on the analysis of COV_{pc} are slightly higher (Fig. 4a) than those reported in [17]. It has to be noted, however, that σ_e and standard uncertainty u_a and u_b are higher in the regression model of the angles determined based on COV than those in the model referring to the literature data.

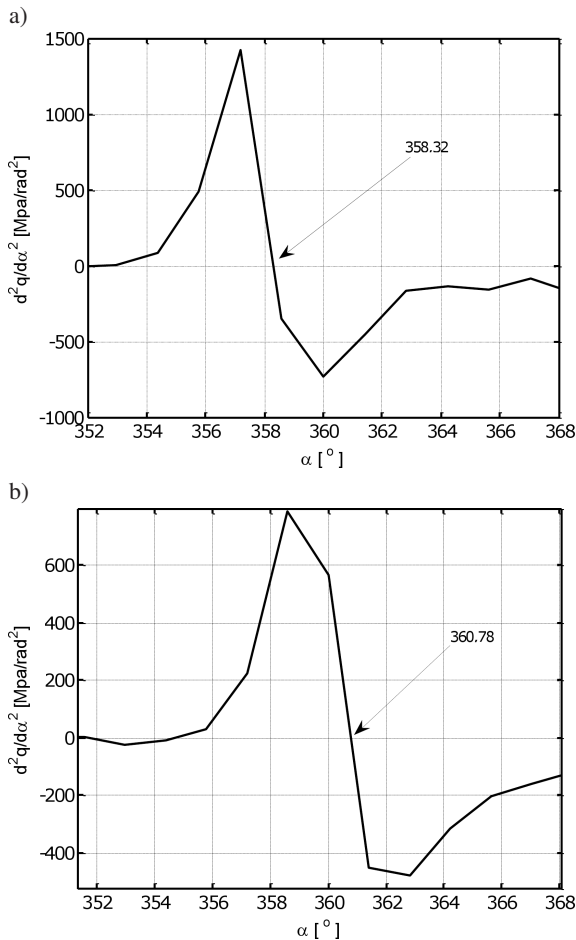


Fig. 3. Plot of $\frac{d^2q}{d\alpha^2} = f(\alpha)$ for diesel fueled engine operating under full-load condition a) for $n = 1400$ rpm, b) for $n = 1800$ rpm

From the graph in Fig. 4b, it is apparent that the values of angles $\alpha_{\dot{Q}_{MAX}}$ determined based on coefficients COV_{pc} and after solving equation (13) vary slightly from the data reported in the literature. The values of angles $\alpha_{\dot{Q}_{MAX}}$ increase with increasing rotational speed. The slopes in Table 2 provide evidence of this fact. The values of standard deviation σ_e and standard uncertainty u_b are similar for all the cases analyzed and considerably lower than those in the regression models of angle α_{ps} . Note that for angle $\alpha_{\dot{Q}_{MAX}}$ the linear regression curve has the better fit to the data from the analy-

sis of COV than to the literature data. The values of linear correlation coefficient $r_{\alpha n}$ confirm this conclusion.

A comparison of the values of standard deviation σ_e in all dependences shows a good convergence of the $COV_{pc}(\alpha)$, analysis results from the proposed analytical model with those reported in the literature.

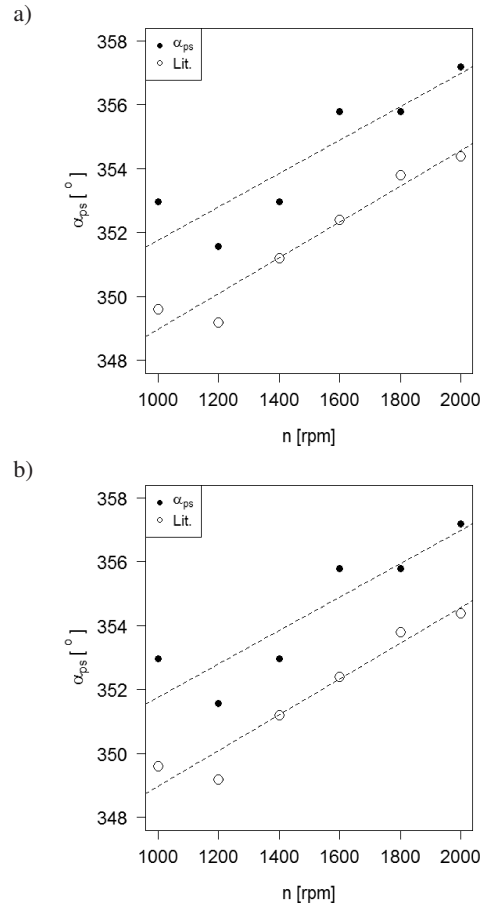


Fig. 4. Comparison of linear regression models and the values of angles: a) α_{ps} determined based on the analysis of COV_{pc} and on the values reported in the literature, b) $\alpha_{\dot{Q}_{MAX}}$ based on the analysis of COV_{pc} , the theoretical model and the values reported in the literature – diesel fueled engine operating under full-load condition in the rotational speed range of $n = 1000 - 2000$ rpm

The angle for which the HRR reaches the maximum value can be determined by analyzing the pressure signal p_c using other methods, for example, fast Fourier Transform (FFT). The FFT is more time-consuming, requires large computing capacities, and the value of σ_e may be higher (for the diesel engine operating by full load characteristics σ_e was 0.91° crank angle).

These methods of calculating angles α_{ps} and $\alpha_{\dot{Q}_{MAX}}$ based on the analysis of coefficients COV_{pc} were used for the results obtained when the engine was fueled with FAME or B20. Similarly to the case of the diesel fueled engine, a satisfactory agreement was found between the values of the angles calculated based on the coefficients $COV_{pc}(\alpha)$ and those reported in the literature and obtained from the theoretical calculations. The parameters of the regression models for the determined angles are shown in Table 3.

Table 2
 Parameters of regression models, described by function $\alpha = f(n) = a \cdot n + b$, for angles α_{ps} and $\alpha_{\dot{Q}_{MAX}}$, presented in Fig. 4a and b

	experiment		literature		theory
	α_{ps}	$\alpha_{\dot{Q}_{MAX}}$	α_{ps}	$\alpha_{\dot{Q}_{MAX}}$	$\alpha_{\dot{Q}_{MAX}}$
$a \cdot 10^{-3}$ [°CAD/rpm]	5.22	5.32	5.57	4.39	4.98
b [°CAD]	346.5	349.1	343.4	351.5	351.6
$r_{\alpha n}$ [-]	0.90	0.99	0.97	0.97	0.99
$u_a \cdot 10^{-4}$ [°CAD/rpm]	12.9	3.51	6.93	5.61	2.55
u_b [°CAD]	1.98	0.54	1.07	0.86	0.39
σ_e [°CAD]	1.08	0.29	0.58	0.47	0.21

Table 3
 Parameters of regression models, described by function $\alpha = f(n) = a \cdot n + b$, for angles α_{ps} and $\alpha_{\dot{Q}_{MAX}}$ – FAME fueled engine operating under full-load conditions

	experiment		literature		theory
	α_{ps}	$\alpha_{\dot{Q}_{MAX}}$	α_{ps}	$\alpha_{\dot{Q}_{MAX}}$	$\alpha_{\dot{Q}_{MAX}}$
$a \cdot 10^{-3}$ [°CAD/rpm]	5.43	6.03	5.43	5.24	5.25
b [°CAD]	345.1	347.2	345.1	347.9	350.2
$r_{\alpha n}$ [-]	0.98	0.98	0.98	0.98	1.00
$u_a \cdot 10^{-4}$ [°CAD/rpm]	5.42	5.82	5.43	4.89	2.27
u_b [°CAD]	0.83	0.90	0.84	0.75	0.35
σ_e [°CAD]	0.45	0.49	0.45	0.41	0.19

When the engine was fueled with FAME, the values of angles α_{ps} calculated based on the analysis of coefficients $COV_{(pc)}$ agreed with the values of angles reported in the literature. Therefore in both cases the same parameters of regression models were obtained. Comparing the values of the computed coefficient of correlation $r_{\alpha n}$, standard deviation σ_e and standard uncertainties u_a and u_b for all the dependences from Table 3, the convergence between the results is apparent. Similar regression models were obtained for engines fueled with B20, both for angle α_{ps} and for $\alpha_{\dot{Q}_{MAX}}$.

Parameters of each of the determined models depend on the type of fuel. This relationship has already been reported in the literature [24]. Further research has to be devoted to quantitative dependencies. From what has been found so far by the authors of this paper follows that the fuel type has a stronger influence on parameter b than on parameter a . Also, the impact of the fuel type on uncertainties u_a , u_b and the value of σ_e has been established to be small.

A potential of the proposed method of in-combustion chamber pressure signal analysis was also studied in engines operating by the load characteristics. The application of coefficient COV_{pc} to determining the values of angles α_{ps} or $\alpha_{\dot{Q}_{MAX}}$ with satisfactory accuracy was confirmed for all the fuels tested. Figure 5 shows the analysis results for $\alpha_{\dot{Q}_{MAX}}$ when the engine was fueled with diesel or FAME.

A comparison of Figs. 5a and 5b shows clearly that the values of angle $\alpha_{\dot{Q}_{MAX}}$ for FAME are lower than those for the diesel fuel. This provides evidence for the statement that the fuel type affects the value of coefficient b in the linear regression model. The slope a in this model takes on negative values for load characteristics. No impact of the fuel type on the value of this coefficient is observed. The fuel type has some influence on the values of uncertainties u_a and u_b as well as on σ_e .

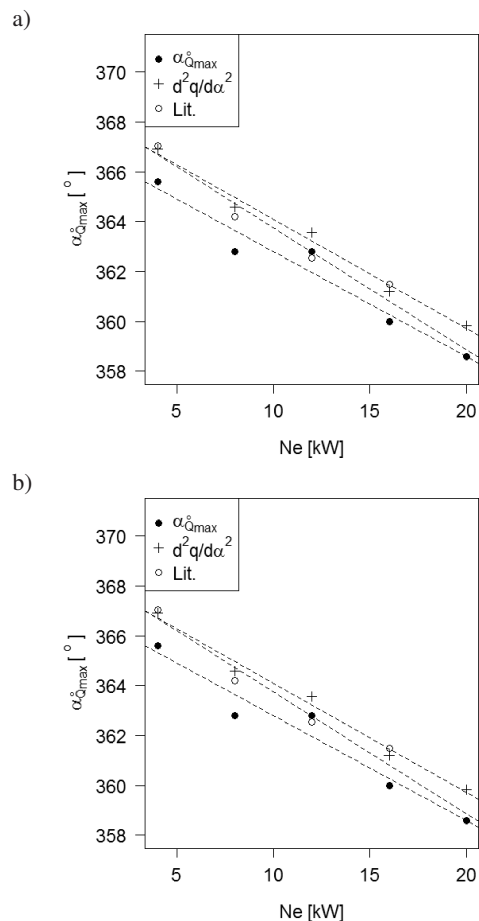


Fig. 5. Comparison of values of angles $\alpha_{\dot{Q}_{MAX}}$ obtained from the analysis of coefficients COV_{pc} , theoretical model and literature data for the engine operating by the load conditions in the range of Ne 4–20 kW, at the speed of $n = 1400$ rpm fueled with: a) diesel, b) FAME

4. Conclusions

The proposed method of analyzing the dependence of pressure in the combustion chamber on the crank angle, based on coefficients COV_{pc} , allows determining the location of the point at which the maximum HRR occurs. The values of crank angles obtained with this method show a satisfactory agreement with the theoretical calculations and the literature data. The proposed method was also used to determine the angle of the start of combustion. A satisfactory agreement was found with the literature data. The method for calculating angles α_{ps} and $\alpha_{Q_{MAX}}$ based on the analysis of coefficients COV_{pc} was used for the results of measurements obtained when the engine was fueled with the diesel, FAME or B20, providing a satisfactory accuracy in each case.

The analysis results of the proposed linear regression models $\alpha = f(n) = a \cdot n + b$ indicate that the sign of slope a depends on the conditions under which the experiment is conducted. In the investigated dependencies, the slope is positive for the full-load conditions and negative for the load characteristics. The fuel type affects the value of parameter b more than the value of a . A slight impact of the fuel type on uncertainties u_a and u_b as well as on σ_e was also observed.

The proposed algorithms of determining combustion descriptors based on coefficients of variation are quite simple to implement, do not require large capacity processors and last short. These properties make them suitable for controlling various systems in real time.

REFERENCES

- [1] L. Feliong, A. Gehan, J. Amaratunga, N. Collings, and A. Soliman, "An experimental study on engine dynamics model based in cylinder pressure estimation", *SAE 2012-01-0896* (2012).
- [2] B. Oger, "Soot characterisation in diesel engines using laser induced incandescence", *PhD Thesis*, University Brighton, Brighton, 2012.
- [3] M. Noga and B. Sendyka, "Increase of efficiency of SI engine through the implementation of thermodynamic cycle with additional expansion", *Bull. Pol. Ac.: Tech.* 62 (2), 349–356 (2014), DOI: 10.2478/bpasts-2014-0034.
- [4] J. Jeschke, "Konzeption und Erprobung eines zylinderdruckbasierten motormanagements für PKW-dieselmotoren", *PhD Thesis*, University Magdeburg, Magdeburg, 2002.
- [5] K. Schten, G. Ripley, A. Punater, and C. Erickson, "Design of an automotive grade controller for in cylinder pressure based engine control development", *SAE 2007-01-0774* (2007).
- [6] M. Sellnau, F. Matekunas, P. Battiston, C. Chang, and D. Lancaster, "Cylinder pressure based engine control using pressure ratio management and low cost non intrusive cylinder pressure sensors", *SAE 2000-01-0932* (2000).
- [7] S. Neumann, "Integrated cylinder pressure measurement for gas engine control", *Combustion Engines* 3, 16–23 (2011).
- [8] G. Fraidl and P. Herzog, "Multiple-benefit technology development for gasoline-diesel-hybrid powertrains", *Combustion Engines* 1, 3–19 (2007).
- [9] U. Libal and J. Płaskonka, "Noise sensitivity of selected kinematic path following controllers for a unicycle", *Bull. Pol. Ac.: Tech.* 62 (1), 3–14 (2014), DOI: 10.2478/bpasts-2014-0001.
- [10] A. Bąkowski, "Metrological analyses of diesel engine indicated signals", *PhD Thesis*, Kielce University of Technology, Kielce, 2013, (in Polish).
- [11] M.D. Redel-Macías, C. Hervás-Martínez, S. Pinzic, P.A. Gutiérrezb, A.J. Cubero-Atienzaa, and M.P. Dorado, "Noise prediction of a diesel engine fueled with olive pomace oil methyl ester blended with diesel fuel", *Fuel* 98, 280–287 (2012).
- [12] Z. Chłopek and P. Stasiak, "The analysis of an unrepeatability of cylinder pressure signal in internal combustion engines", *Combustion Engines* 1, 31–39 (2005).
- [13] R.K. Maurya and A.K. Agarwal, "Investigations on the effect of measurement errors on estimated combustion and performance parameters in HCCI combustion engine", *Measurement* 46, 80–88 (2013).
- [14] A. Różycki, "Identification method of knock combustion in two fuel combustion ignition engine", *KONES* 15, 479–487 (2008).
- [15] K.D. Neroorkar, "Simulations and analysis of fuel flow in an injector including transient needle effects", *ILASS-Americas 24th Annual Conf Liquid Atomization and Spray Systems TX*, CD-ROM (2012).
- [16] M.Y.E. Selim, "Effect of engine parameters and gaseous fuel type on the cyclic variability of dual fuel engines", *Fuel* 84, 961–971 (2005).
- [17] P. Łagowski, "Methodology of determining and evaluation of diagnostics parameters of indicated plot for an internal combustion engine", *PhD Thesis*, Kielce University of Technology, Kielce, 2010, (in Polish).
- [18] A. Ambrozik, "Glued functions – based assessment of interpolation accuracy of self ignition engine real indicator diagram", *Solid State Phenomena, Mechatronic Systems, Mechanics and Materials* 180, 261–268 (2012).
- [19] Z. Stelmasiak, *Gas Combustion Process in a Dual-Fuel Diesel Engine Supplied with Natural Gas And Diesel*, Academy of Technology and Humanities in Bielsko Biała, Bielsko Biała, 2003, (in Polish).
- [20] T. Bodisco and R.J. Brown, "Inter-cycle variability of in-cylinder pressure parameters in an ethanol fumigated common rail diesel engine", *Energy* 52, 55–65 (2013).
- [21] J.M. Luján, V. Bermudez, C. Guardiola, and A. Abbad, "A methodology for combustion detection in diesel engines through in-cylinder pressure derivative signal", *Mechanical Systems and Signal Processing* 24, 2261–2275 (2010).
- [22] S. Polanowski, "A review of methods for analyzing indicated plots with regard to marine engine diagnostics", *Scientific Works of the Polish Naval Academy* 169, CD-ROM (2007), (in Polish).
- [23] R.K. Maurya and A.K. Agarwal, "Statistical analysis of the cyclic variations of heat release parameters in HCCI combustion of methanol and gasoline", *Applied Energy* 89, 228–236 (2012).
- [24] P.X. Pham, T.A. Bodisco, Z.D. Ristovski, R.J. Brown, and A.R. Masri, "The influence of fatty acid methyl ester profiles on inter-cycle variability in a heavy duty compression ignition engine", *Fuel* 116, 140–150 (2014).