A QUICK, SIMPLIFIED APPROACH TO THE EVALUATION OF COMBUSTION RATE FROM AN INTERNAL COMBUSTION ENGINE INDICATOR DIAGRAM

by

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In this paper a simplified procedure of an internal combustion engine in-cylinder pressure record analysis has been presented. The method is very easy for programming and provides quick evaluation of the gas temperature and the rate of combustion. It is based on the consideration proposed by Hohenberg and Killman, but enhances the approach by involving the rate of heat transferred to the walls that was omitted in the original approach. It enables the evaluation of the complete rate of heat released by combustion (often designated as "gross heat release rate" or "fuel chemical energy release rate"), not only the rate of heat transferred to the gas (which is often designated as "net heat release rate"). The accuracy of the method has been also analyzed and it is shown that the errors caused by the simplifications in the model are very small, particularly if the crank angle step is also small. A several practical applications on recorded pressure diagrams taken from both spark ignition and compression ignition engine are presented as well.

Key words: *internal combustion engine, indicator diagram, rate of heat release*

Introduction

Classical demands: high power output, shape of torque curve, reliability, production costs, *etc*., are still important but not sufficient in modern internal combustion (IC) engine development. Fuel economy, exhaust emission and engine noise have become important parameters not only for engine competitiveness, but also are subjected to legislation becoming more severe every few years. To satisfy and fulfill all these demands, a very comprehensive knowledge on complex chemical and physical processes occurring during energy transformation in the engine are required in order to characterize complex influences and find out the ways and directions how to improve in-cylinder processes and, if possible, avoid undesirable effects.

The combustion is the key process crucially influencing engine performances (power output, torque, specific fuel consumption) as well as engine environmental characteristics *i*. *e*. exhaust emissions, noise, *etc*. Normally, power output, torque and fuel consumption are measured during engine testing, and also, if necessary, exhaust gas composition, noise, *etc*., so that the global engine efficiency can be recognized. However, more sophisticated information about in-cylinder invents is necessary in order to have insight in the sources and distribution of energy losses, exhaust gas pollutants formation, *etc*.

One of the most frequently used ways to obtain necessary information about the working process is recording of the in-cylinder pressure trace, or so-called "indicator diagram". Even without any calculation the pressure record provides some information about combustion in engine cylinder, for example: peak pressure and its position, the rate of pressure rise (influencing basically combustion noise), *etc*. Research engineer can obtain more detailed information by the analysis of pressure diagram. Computation of mean indicated pressure of high pressure and low pressure parts of the cycle enables the determination of engine mechanical losses and gas exchange losses. The combustion process and its losses are, however, more complex, and therefore, far more sophisticated thermodynamic analysis of the pressure data is required. The rate of heat released by combustion or simply "the rate of combustion" and the mean gas temperature appear as major results of that analysis.

The importance of the information that can be obtained from an engine in-cylinder pressure trace has been recognized since early days of engine development. Scientists and engineers have been paying great attention to indicator diagram analysis even when test instrumentation and the calculation possibilities where rather limited before the days of digital computers and data acquisition systems. The early publications on these topics, such as Neumann [1], Zinner [2], and List [3], take into account rather realistic thermodynamic properties of cylinder gas. However, the interest and possibilities for refining the methods increased as digital computers had been introduced in modeling and computation of engine working cycle. The development and application of digital data acquisition systems in engine testing have further gained the possibilities of measurement high speed engine variables and significantly improved the accuracy of acquired data.

With the introduction of digital computers and ever increasing application in engine working cycle modeling, the computer-aided methods for indicator diagram analysis have been simultaneously developed and reported. The works of Vibe [4] and Lange and Woschni [5] are well known, but the most comprehensive analysis was reported by Krieger and Borman [6] and their method has been regarded as a reference for indicator diagram analysis. Another remarkably in-depth work, introducing the number of influences, was given by Jankov [7], although this method, as well as [6], is primarily intended for compresion ignition (CI) engines. A good survey over numerous approaches and methods and appropriate discussion was given by Heywood [8].

However, even with modern test equipment and calculation possibilities, the recording of indicator diagram and its thermodynamic analysis remain very sensitive and demanding task subjected to number of possible errors. Some of these problems will be discussed in text as well.

Theoretical background

Engine combustion chamber as open thermodynamic system is presented schematically in fig. 1. The basic equation for all known methods is the first law of thermodynamic applied to an open system [6-8]:

$$\mathrm{d}Q \quad \mathrm{d}U \quad p\mathrm{d}V \tag{1}$$

Here, $\mathrm{d}Q$ is the change of elementary energy entering/leaving the system (except kinetic energy which is omitted), $\mathrm{d}U$ is the elementary change of cylinder charge internal energy and $p\mathrm{d}V$ is elementary mechanical work delivered to the piston.

The derivative dQ consists of the energy released by the combustion dQ_f, the energy transferred to the walls dQ_w and the energies taken into/out the system by means of the mass flows through the boundaries $\Sigma h_i \mathrm{d}m_i$, where h_i and dm_i are the enthalpies and elementary mass flows, respectively. Thus, we can write:

$$\mathrm{d}Q \quad \mathrm{d}Q_\mathrm{f} \quad \mathrm{d}Q_\mathrm{w} \quad \sum_i h_i \mathrm{d}m_i \tag{2}$$

The change of internal energy dU is further evaluated as:

$$\mathrm{d}U \quad \mathrm{d}(mu) \quad m\mathrm{d}u \quad u\mathrm{d}m \tag{3}$$

Substituting the eqs. (2) and (3) in eq. (1), the elementary energy released by combustion can be expressed as follows:

$$\mathrm{d}Q_\mathrm{f} \quad m\mathrm{d}u \quad p\mathrm{d}V \quad u\mathrm{d}m \quad \sum_i h_i \mathrm{d}m_i \quad \mathrm{d}Q_\mathrm{w} \tag{4}$$

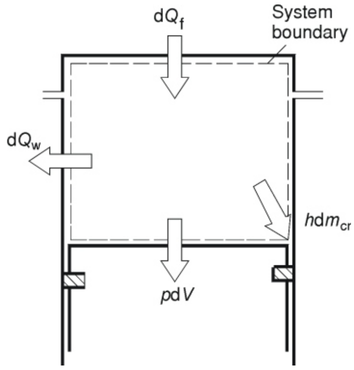

Figure 1. Engine cylinder as an open thermodynamic system

Since the high pressure part of the cycle is considered, intake and exhaust valves are closed and there are no flows through inlet and exhaust ports. In the absence of fuel injection (the case of spark ignition engine) only flow into and out of crevice region dm_{cr} is present. In that case the terms representing the elementary change of the system mass and the energy entering/leaving the system by mass flows can be specified as:

$$\mathrm{d}m \quad \mathrm{d}m_\mathrm{cr}; \quad \sum_i h_i \mathrm{d}m_i \quad h\mathrm{d}m_\mathrm{cr} \tag{5}$$

Regarding CI engine, the portion of fuel is injected near the end of compression and the considered terms become:

$$\mathrm{d}m \quad \mathrm{d}m_\mathrm{f} \quad \mathrm{d}m_\mathrm{cr}; \quad \sum_i h_i \mathrm{d}m_i \quad h_\mathrm{f}\mathrm{d}m_\mathrm{f} \quad h\mathrm{d}m_\mathrm{cr} \tag{6}$$

Thermal enthalpy of the fuel injected is very small ($h_f \sim 0$) compared to its heating value H_f, and can be neglected, while the increase of cylinder charge can be a few percent at full load. Introducing terms from eq. (6), the eq. (4) becomes as follows:

$$\mathrm{d}Q_\mathrm{f} \quad m\mathrm{d}u \quad p\mathrm{d}V \quad u\mathrm{d}m_\mathrm{f} \quad (h \quad u)\mathrm{d}m_\mathrm{cr} \quad \mathrm{d}Q_\mathrm{w} \tag{7}$$

or expressed per crank shaft angle:

$$\frac{\mathrm{d}Q_\mathrm{f}}{\mathrm{d}\alpha} \quad m\frac{\mathrm{d}u}{\mathrm{d}\alpha} \quad p\frac{\mathrm{d}V}{\mathrm{d}\alpha} \quad u\frac{\mathrm{d}m_\mathrm{f}}{\mathrm{d}\alpha} \quad (h \quad u)\frac{\mathrm{d}m_\mathrm{cr}}{\mathrm{d}\alpha} \quad \frac{\mathrm{d}Q_\mathrm{w}}{\mathrm{d}\alpha} \tag{8}$$

In further evaluations, it is necessary to include the ideal gas law in differential form:

$$pV = m\mathrm{R}T$$

$$\frac{1}{p}\frac{\mathrm{d}p}{\mathrm{d}\alpha} \quad \frac{1}{V}\frac{\mathrm{d}V}{\mathrm{d}\alpha} \quad \frac{1}{m}\frac{\mathrm{d}m}{\mathrm{d}\alpha} \quad \frac{1}{\mathrm{R}}\frac{\mathrm{d}\mathrm{R}}{\mathrm{d}\alpha} \quad \frac{1}{T}\frac{\mathrm{d}T}{\mathrm{d}\alpha} \tag{9}$$

In general, the thermodynamic properties of the cylinder gas are the functions of pressure, temperature, and gas composition. The composition both prior the combustion (fuel-air mixture or pure air plus residual gas) and after combustion (combustion products), depends on fuel composition and mixture strength, usually represented by air excess ratio – $\lambda = m_a/m_f L_0$. Then, in general, we have:

$$u \quad u(p, T, \lambda) \quad \text{and} \quad \mathrm{R} \quad \mathrm{R}(p, T, \lambda)$$

$$\frac{\mathrm{d}u}{\mathrm{d}\alpha} \quad \frac{\delta u}{\delta p}\frac{\mathrm{d}p}{\mathrm{d}\alpha} \quad \frac{\delta u}{\delta T}\frac{\mathrm{d}T}{\mathrm{d}\alpha} \quad \frac{\delta u}{\delta \lambda}\frac{\delta \lambda}{\mathrm{d}\alpha} \quad \text{and} \quad \frac{\mathrm{dR}}{\mathrm{d}\alpha} \quad \frac{\delta \mathrm{R}}{\delta p}\frac{\mathrm{d}p}{\mathrm{d}\alpha} \quad \frac{\delta \mathrm{R}}{\delta T}\frac{\mathrm{d}T}{\mathrm{d}\alpha} \quad \frac{\delta \mathrm{R}}{\mathrm{d}\lambda}\frac{\mathrm{d}\lambda}{\mathrm{d}\alpha} \tag{10}$$

After substituting eqs. (9) and (10) in eq. (8), the eq. (8) can be solved numerically and the rate of the heat released by combustion can be evaluated. The recorded cylinder pressure yields $p(\alpha)$ and $\mathrm{d}p/\mathrm{d}\alpha$, while initial mass of the system m_0 and mixture strength λ must be known. The appropriate models for the thermodynamic properties of the working fluid and heat transfer term $\mathrm{d}Q_w/\mathrm{d}\alpha$ are also required.

In the case of CI engine, an alternative approach yielding the fuel mass burning rate $\mathrm{d}m_{fb}/\mathrm{d}\alpha$ is often applied – eqs. (6) and (7). The connection is simple:

$$\frac{\mathrm{d}Q_f}{\mathrm{d}\alpha} \quad H_f \frac{\mathrm{d}m_{fb}}{\mathrm{d}\alpha} \tag{11}$$

where H_f is the fuel lower heating value.

In the expressions (10) for thermodynamic properties of the working fluid the gas composition is expressed as the function of air excess ratio λ. If a simple model is applied, the composition of combustion products is evaluated by means of stoichiometric equations for fuel-air mixture combustion. More comprehensive models consider chemical equilibrium or chemical reactions kinetics, enabling dissociation of combustion products to be taken into account. The heat transfer to combustion chamber walls can be modeled using well known Newton's convective heat transfer equation and this term will be evaluated and discussed later on. Since crevice effects are usually small, a sufficiently accurate, yet simple model for their overall effect is to consider a single aggregate crevice volume where the gas is at the same pressure as the combustion chamber but at different temperature. Since these crevice regions are very narrow, next assumption regards the crevice gas at the wall temperature.

Figure 2 shows a general distribution of cumulative heat release. All traces are normalized, *i. e*. divided by the total fuel chemical energy taken into cylinder $Q_{f0} = m_f H_f$. Dashed line represents the net heat release Q_n *i. e*. heat delivered to the gas. The addition of heat transfer to the walls Q_w and energy due to crevice flow, yields the fuel chemical energy release Q_f or gross heat release The straight line at the top represents the total fuel chemical energy taken into cylinder (normalized). The difference between this line and final value of Q_f should be equal to the combustion inefficiency. The combustion inefficiency can be determined from the exhaust gas composition. Inaccuracies in the cylinder pressure data and the heat release calculation will also contribute to this difference.

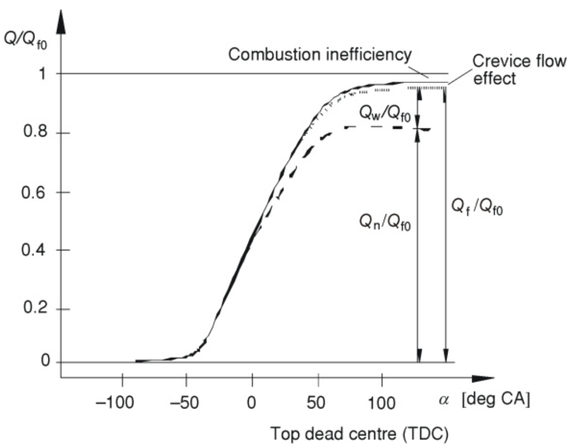

Figure 2. Cumulative heat release distribution *vs.* crank angle (CA)

The approach yielding the eq. (8), in combination with comprehensive models for gas properties (9) and (10), provides a considerable level of sophistication. However, it still contains the assumptions and uncertainties making the results only approximate. For example, the uniformities of temperature and gas composition are assumed. Also, fuel-air mixture formation (in the case of CI engine) is omitted and all available models for wall heat transfer and crevice flow are rather inaccurate. The term "apparent" is often used to describe these uncertainties, *i. e*. the obtained results are frequently named as "apparent rate of heat release" or "apparent fuel mass burning rate".

Krieger and Borman [6] carried out the sensitivity analysis for critical assumptions and variables. They found that the effect of dissociation was negligible. They also neglected the effect of crevice flow. All this permits the substantial simplification in eqs. (8), (9), and (10), but, in general, the model remains rather complex and inconvenient for engineering practice. The method explained in the text below is very simple and easy for programming, but still accurate enough for practical purposes. Namely, its practical application is extremely simple and low cost, and it could be even incorporated in engine testing control system to provide "on line" information about combustion.

Simplified method for the evaluation of heat release rate

Pressure diagram recorded using data acquisition system is the series of pressure values taken in discrete crank shaft positions. The series of pressure-crank angle data can be easily transformed into pressure-volume data. Since modern crank angle (CA) encoders have a very fine angular increment, the changes in pressure and volume are small. So, if we consider the changes between two recorded points as elementary, the error will be within reasonable limits.

During the gas state change between two consecutive points 1 to 2 (fig. 3), the amount of fuel chemical energy ΔQ_f is released by combustion. If we neglect the effect of crevice flow, this heat is partly transferred to the gas ΔQ_n (formerly designated as "net heat release"), and partly transferred to the walls ΔQ_w. Then, we can write:

$$\Delta Q_{\mathrm{f}} \quad \Delta Q_{\mathrm{f}} \quad \Delta Q_{\mathrm{f}} \tag{12}$$

The evaluation of ΔQ_n and ΔQ_w can be performed in a simplified way, convenient for engine laboratory testing, but still sufficiently accurate.

Heat transferred to the gas

Hohenberg and Killman [9] proposed a simplified method for the evaluation of the amount of heat transferred to the gas ΔQ_n. The real process (1 to 2) is virtually divided into two steps (fig. 3). First step is the adiabatic isentropic expansion (or compression) from point 1 to point 2s (from volume V_1 to volume V_2) with no heat transfer to gas. In the second step, the heat added to gas by combustion is being considered at constant volume (V_2 = = const.). The required parameters in characteristic points can be obtained using the ideal gas law and the equation for isentropic state change. If gas behavior is considered as ideal

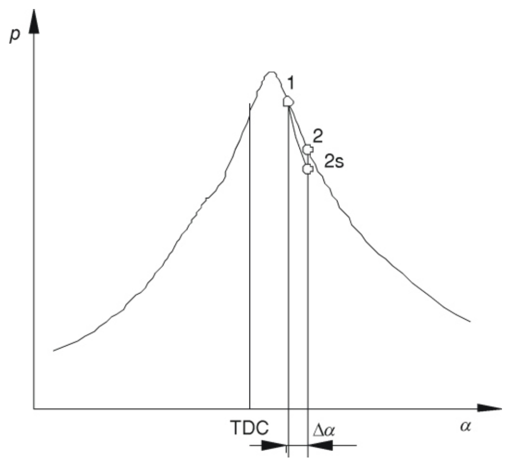

Figure 3. Recorded p-α diagram; gas state change from point 1 to point 2

(which is, actually, not to far from real process) we can write:

$$p_{2\mathrm{s}} = p_1 \left(\frac{V_1}{V_2}\right)^{\kappa}; \quad \kappa = \frac{c_\mathrm{p}}{c_\mathrm{v}} \tag{13}$$

$$T_1 = \frac{p_1 V_1}{m\mathrm{R}}; \quad T_2 = \frac{p_2 V_2}{m\mathrm{R}} \tag{14}$$

$$T_{2\mathrm{s}} = \frac{p_{2\mathrm{s}} V_2}{m\mathrm{R}} = \frac{p_1 V_1^{\kappa} V_2^{1-\kappa}}{m\mathrm{R}} \tag{15}$$

$$\Delta Q_\mathrm{n} = \Delta Q\Big|_1^2 = m c_\mathrm{v} (T_2 - T_{2\mathrm{s}}) = \frac{c_\mathrm{v}}{\mathrm{R}} V_2 (p_2 - p_{2\mathrm{s}}) \tag{16}$$

Finally, after substituting, we obtain the amount of heat released by combustion and transferred to the gas between points 1 and 2 in the form:

$$\Delta Q_\mathrm{n} = \frac{c_v}{\mathrm{R}} V_2 \left[p_2 - p_1 \left(\frac{V_1}{V_2}\right)^{\kappa} \right] \tag{17}$$

Described approximation is obvious in *T-s* diagram shown in fig. 4 where the amount of heat calculated using eq. (17), ΔQ_n, is shaded.

For the evaluation of eq. (17) the thermodynamic properties of the gas in combustion chamber are required. Several approaches, appropriate simplified expressions and their influence on heat release rate evaluation accuracy are laid down and discussed in Appendix.

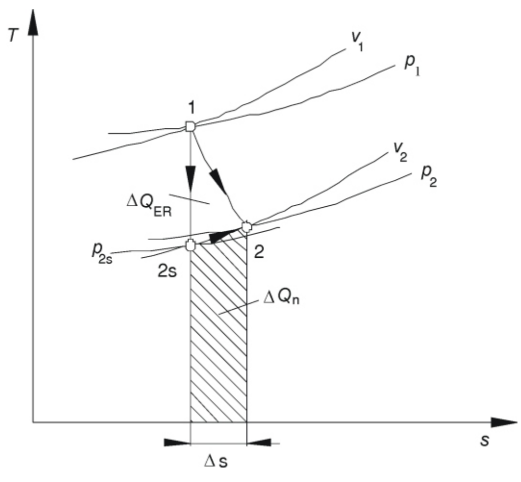

Figure 4. Gas state change from point 1 to 2 in *T-s* diagram

Heat transferred to the walls

The convective heat transfer rate to the combustion chamber walls can be calculated from the general relation:

$$\frac{\mathrm{d}Q_\mathrm{w}}{\mathrm{d}t} = \alpha_\mathrm{w} A_\mathrm{w} (T - T_\mathrm{w}) \tag{18}$$

where α_w is the heat transfer coefficient (averaged over the chamber surface area), A_w is combustion chamber surface area, T_w is the mean wall temperature of combustion chamber surface area, and T is the mean gas temperature. Heat transfer to the walls surrounding combustion chamber is usually considered separately for characteristic parts of combustion chamber surface area, since they have significantly different temperature. Heat transfer coefficient is mainly taken as average for the whole combustion chamber because of lack of accurate data for different parts. Than we have:

$$\frac{\mathrm{d}Q_\mathrm{w}}{\mathrm{d}t} = \alpha_\mathrm{w} \sum_i A_{\mathrm{w}i} (T - T_{\mathrm{w}i}) \tag{19}$$

Usually, piston crown, cylinder head and cylinder liner are considered as the elements in expression (19), the last having variable surface according to piston motion and variable temperature from the top (at top dead centre – TDC) to the bottom (at bottom dead centre – BDC).

Therefore, the amount of heat transferred to the walls between two consecutive points 1 and 2 can be calculated as:

$$\Delta Q_{\mathrm{w}} \quad [\alpha_{\mathrm{w}} \sum_i A_{\mathrm{w}i}(T \quad T_{\mathrm{w}i})]\Delta t \quad [\alpha_{\mathrm{w}} \sum_i A_{\mathrm{w}i}(T \quad T_{\mathrm{w}i})]\frac{\Delta\alpha}{6n} \tag{20}$$

where $\Delta\alpha$ is angular increment and n engine speed in rpm.

For the heat transfer coefficient several models are widely used. One of the recent is Hohenberg's expression which is relatively simple and convenient for use [10]:

$$\alpha_{\mathrm{w}} \quad 0.013 V^{\quad 0.06} p^{0.8} T^{\quad 0.4} (c_{\mathrm{m}} \quad 1.4)^{0.8} \ \mathrm{J/m^2K} \tag{21}$$

where V is instantaneous cylinder volume, p and T cylinder pressure and temperature, and c_{m} mean piston velocity. Alternatively, some other models, for example Woschni's or Annand's can be used [7].

Calculated heat transfer to the walls should be considered as an approximation rather than quite accurate result. Firstly, whichever model for heat transfer coefficient is used, it is based on limited experimental data and its validity can hardly be universal for all engine categories. Secondly, accurate temperatures of combustion chamber walls are unknown. Direct measuring of walls temperatures is very demanding and complicated task, hardly feasible during normal engine testing. These temperatures are usually estimated on the base of literature data but such estimation can be rather inaccurate for concrete case. Literature data mainly correspond to walls maximum temperatures for appropriate engines categories, but wall temperature changes with engine speed and load can be hardly found. Having all this in mind, calculated heat transfer to the walls is often corrected in order to obtain logical result – for example, at the end of combustion process, to reach reasonable agreement between calculated heat released by combustion and fuel chemical energy taken into engine cylinder per cycle.

Evaluating the accuracy of the methodology

The calculation of the heat transferred to the gas ΔQ_{n} according to eq. (17), is an approximation which can be clearly seen in entropy-temperature diagram shown in fig. 4. The difference between real heat transferred to the gas (proportional to the area under the curve 1-2) and calculated value (proportional to the area 1-2s-2-1) is being designated as ΔQ_{ER} (error). This difference can be estimated using the entropy rise (Δs, fig. 4) which is the same for real process (1-2) and considered approximation (1-2s-2):

$$\Delta s \quad \frac{2\Delta Q_{\mathrm{n}}}{T_{2\mathrm{s}} \quad T_2} \tag{22}$$

$$\Delta Q_{\mathrm{ER}} \quad \frac{T_1 \quad T_{2\mathrm{s}}}{2}\Delta s \quad \Delta Q_{\mathrm{n}} \frac{T_1 \quad T_{2\mathrm{s}}}{T_2 \quad T_{2\mathrm{s}}} \tag{23}$$

After substituting eqs. (14) and (15) in eq. (23) the error can be expressed as:

$$\Delta Q_{\mathrm{ER}} \quad \Delta Q_{\mathrm{n}} \frac{\left(\frac{V_2}{V_1}\right)^{\kappa \;\; 1} \quad 1}{\frac{p_2}{p_1}\left(\frac{V_2}{V_1}\right)^{\kappa} \quad 1} \tag{24}$$

The error is negative before TDC (calculated ΔQ_{n} is greater then the real heat transferred to gas) and positive after TDC (calculated ΔQ_{n} is less then the real heat transferred to gas). It is obvious that the most influencing factor is the volume change between the points 1 and 2. The less is the volume change, the less is calculation error. For small angular increment $\Delta\alpha$, the error practically diminishes and become negligible. Figure 5 shows the real example according to data given in section *Some results of the method's practical application*. With angular increment 0.35156 deg CA the absolute value of relative error does not exceed 0.15%.

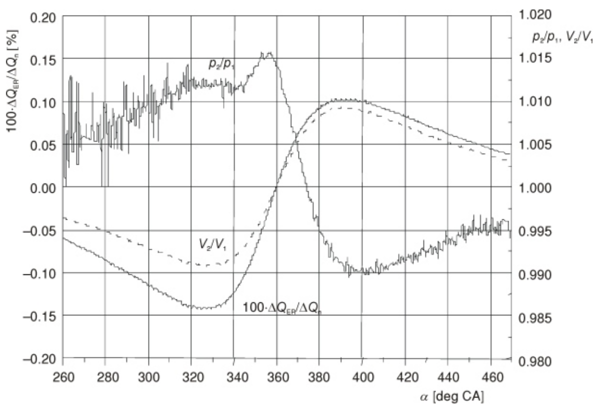

Figure 5. Relative calculating error due to the approximation of real heat transferred to the gas between two consecutive points by eq. (17)

Other errors could be caused by using approximate, simple models for the gas thermodynamic properties (expressions given in Appendix for constant volume specific heat and gas constant). Certain error is also produced due to simple summing portions of heat released between two discrete points (1 and 2), instead of numerical evaluation of differential equation. This kind of error is also minimized by CA interval decrease, and for small angular increment becomes negligible.

The overall result is often influenced by other types of errors, for example possible measuring inaccuracies: cylinder pressure record and its synchronization with CA, measured fuel and air mass and the estimation of residual gas mass. A model describing heat transfer to the walls is also inaccurate. In order to estimate the total error of the method (but free of other kinds of possible errors), a numerical experiment was curried out using a sophisticated commercial software for engine cycle modeling – AVL Boost [11].

Engine system simulation BOOST uses exact models for real gas thermodynamic properties (as the functions of pressure, temperature and composition), takes into account crevice flow, uses accurate differential equations solver, *etc*. With appropriate data and defined rate of heat release, engine cycle was modeled using BOOST to obtain cylinder pressure diagram. This diagram was subsequently analyzed using described method for simplified evaluation of the rate of heat release and results were compared. Figure 6 shows the comparison of the rate of heat release and cumulative heat release defined in engine cycle modeling (solid lines) and the results obtained by pressure diagram analysis using described method (dashed lines) applied to a 1.4l SI engine at full load and 5500 rpm. As can be seen the agreement is quite acceptable. The maximum difference in heat release rate, appearing around the peak, is less than 5%, while the difference in final cumulative heat release value is 1.8%. The difference between the results obtained using the expressions (a1) and (a2) for specific heats given in Appendix are to small to be visible in the graph.

Some results of the method's practical application

Spark ignition engine

The described method for the evaluation of the rate of heat release is demonstrated on several practical examples. In addition to standard engine testing equipment (dynamometer with engine torque and engine speed measuring system, intake air and fuel flow measuring systems, inlet air, exhaust gases, engine coolant, and oil temperatures measuring system), the SI engine was equipped with the system for in-cylinder pressure recording. The water cooled, temperature compensated piezoelectric pressure transducer was installed in cylinder head yielding direct access to the first cylinder combustion chamber (top of the transducer coincides with the combustion chamber wall). Signal conditioning was accomplished through charge amplifier and led to digital data acquisition and control system. The main characteristics of this system are given in tab. 1.

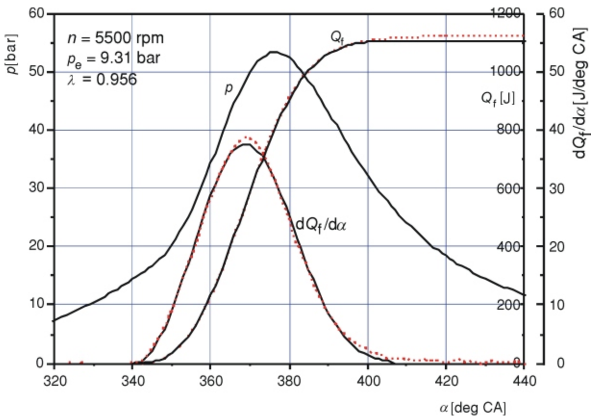

Figure 6. The comparison of modeled rate of heat release (solid line) and the rate of heat released obtained by the analysis of modeled pressure diagram usin desctibed model (dashed line)

Table 1. Measurement system – technical specification

Pressure transducer	Water cooled, drift-compensated AVL 8QP500C
Charge amplifier	KISTLER 5001
Acquisition system	National Instruments PXI
Acquisition board	NI-PXI 6123S No. of chanels: 8 Sampling rate: 8 500 kilo sampling per second
Angle sensor	DAAM CSAS-10 Angular resolution: 1024 incriments per rpm

The optical angular encoder with 1024 marks per revolution was fitted to the engine crank shaft. Pressure signal was recorded for 100 consecutive cycles and mean cycle was evaluated for subsequent analysis.

A special attention was paid to exact determination of absolute pressure level and pressure trace to CA synchronization, since these problems could significantly influence the results of heat release analysis. The piezoelectric pressure transducers, because of their physical characteristics show only pressure differences, which means that the absolute value (zero line) has to be measured or evaluated separately. Moreover, this must be done for every recorded cycle since signal has tendency to float (zero drift). This problem was solved by application of thermodynamic determination of absolute pressure level [9]. The procedure is based on the comparison of recorded pressure trace and theoretical compression line (at the characteristic part of compression stroke), calculated with polytrophic coefficient 1.32 for SI engines and 1.38 for CI engines. Method has been widely used in practice and proved to be very effective [9, 13].

The precise synchronization between pressure signal and CA position is actually the problem of exact determination the TDC position. Since piston motion near TDC is very small, mechanical registration of TDC position is inaccurate. Besides, in operating condition, under load, TDC position can vary a little bit from its stationary position due to engine parts thermal and mechanical deformation. Actual TDC position was determined by recording pressure diagram of the cycles without combustion (pure compression and expansion). It was achieved by switching off the ignition in the cylinder fitted with pressure transducer. The position of maximum pressure is advanced compared to the TDC due to energy losses, as it is shown in fig. 7. This angle named "the thermodynamic loss angle", α_{thloss}, depends on heat transfer to the walls and crevice flow [12, 13]. In general, it can vary in the range of 0.4-1 deg CA [13] and in this case it was taken to be 0.5.

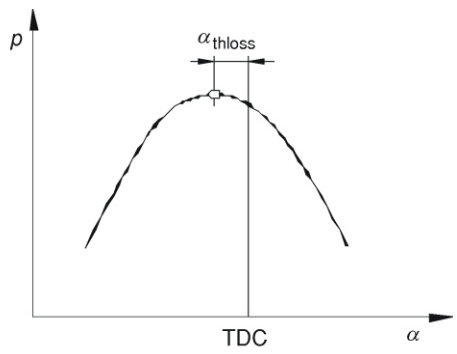

Figure 7. Thermodynamic loss angle

Basic technical data for the engine used in this experiment are given in tab. 2. The engine is fueled by carburetor directly fitted on cylinder head, without intake manifold. Inlet runners, very short and of different length are cast within the cylinder head. Due to layout of inlet pipes significant cylinder to cylinder mixture strength variation can be expected. It was not possible to determine mixture strength distribution; hence, the air excess ratio marked on each graph is the overall for the engine and the actual value for recorded cylinder can differ from this value.

Table 2. Main engine data

Producer	DMB Belgrade
Engine type	SI, 100 GL
Bore/Stroke	65/68 mm/mm
Number of cylinders	4
Compression ratio	9
Max. power output	30 kW/5600 rpm
Cooling system	water
Fueling system	carburetor

Figure 8 shows the results of pressure diagram analysis for one of recorded engine operating conditions: rate of heat release $\mathrm{d}Q_\mathrm{f}/\mathrm{d}\alpha$, cumulative heat released Q_f and the mean gas temperature T. As mentioned before, the mean value of 100 consecutive cycles was analyzed and no curve smoothing procedure was applied. Nevertheless, the obtained curve $\mathrm{d}Q/\mathrm{d}\alpha$ is sufficiently smooth although it is very sensitive to measuring noise that can be superposed to pressure record. As it can be seen the combustion process starts at approx. 345 deg CA and ends at approx. 415 deg CA. If we consider 5 to 95% of fuel burned mass, what is usual in combustion analysis, it occurs between 350 and 400 deg CA.

In fig. 9, the same results are shown in normalized form, *i. e.* $x = Q_\mathrm{f}/Q_\mathrm{f0}$ and $\mathrm{d}x/\mathrm{d}\alpha =$ $= (1/Q_\mathrm{f0})(\mathrm{d}Q_\mathrm{f}/\mathrm{d}\alpha)$, where $Q_\mathrm{f0} = m_\mathrm{f}H_\mathrm{f}$. At the end of combustion x reaches the value of approx. 0.94. Considering that crevice effects are omitted and engine operates with slightly rich mixture ($\lambda = 0.97$), this value could be quite real. Since the rich mixture's lower heat value is actually decreased, Q_f0 could be corrected for the carbon monoxide concentration in exhaust gas (calculated theoretically). In that case the final value of x would be approx. 0.976. However, determined air excess ratio of 0.97 is the mean value for the engine, and its slight variation in the cylinder instrumented with the pressure transducer has to be considered in a final assessment of the result.

It is interesting to examine the sensitivity of heat release analysis to the most frequent possible errors. Figure 10 shows the results of analysis where heat transfer coefficient has been increased and decreased by factor 1.5. Pressure-crank angle synchronization has been also changed for ±1 and ±2 deg CA, and such analysis is presented in fig. 11. All results are given in normalized form.

With relatively large change of heat transfer calculation (±50%), the change in the rate of heat release ($dx/d\alpha$) is not so obvious, but cumulative maximum reached value of heat released (x) differs approx. ±6%. Besides, if heat transfer is over estimated, line x continue to increase after combustion has been ended and decreases (this is often called droop) in the case of heat transfer under estimation. The influence of pressure measurement – CA synchronization is much greater. As it can be seen, the error of only 1 deg CA, which is quite possible if special care was not paid to TDC position determination, produces the significant error in calculated results. It changes not only maximum reached values of x and $dx/d\alpha$, but also their shapes, *i. e.* the combustion is falsely interpreted to be faster or slower than real process occurred. Of course, analyzed influences on accuracy of calculation are not characteristic only for described method and have a universal character.

As mentioned in the introduction, the rate of heat release is very convenient parameter for engine combustion effectiveness analyses. The influence of engine load on combustion is illustrated in fig. 12, where the rate of heat release is plotted for three different loads at approx. similar engine

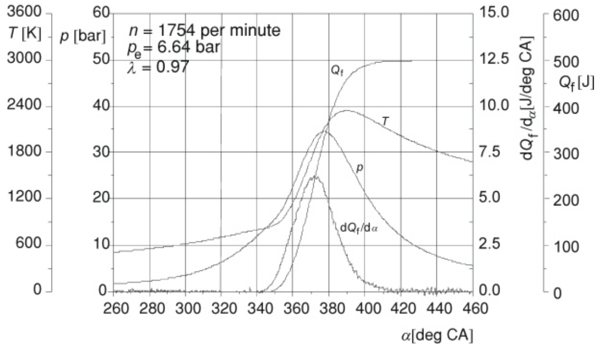

Figure 8. The results of pressure diagram analysis: rate of heat release $dQ_f/d\alpha$, cumulative heat released Q_f, and mean gas temperature T

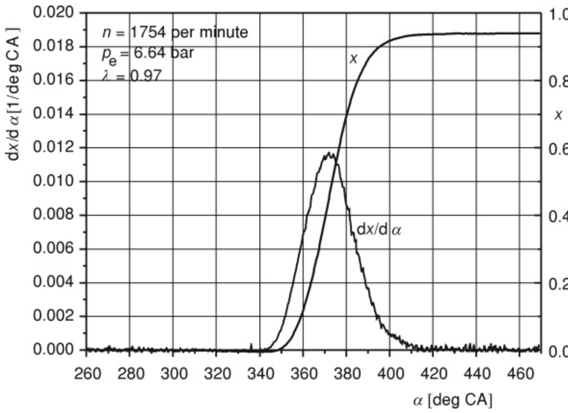

Figure 9. The normalized results of pressure diagram analysis: normalized rate of heat release $dx/d\alpha$, and cumulative normalized heat released x

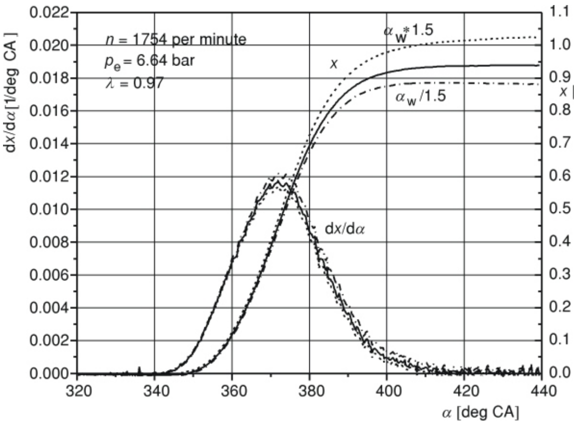

Figure 10. The influence of heat transfer coefficient on the heat release evaluation

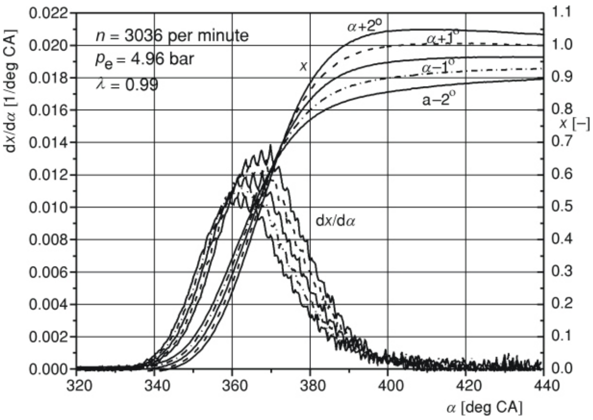

Figure 11. The influence of pressure – crank angle synchronization on the heat release evaluation

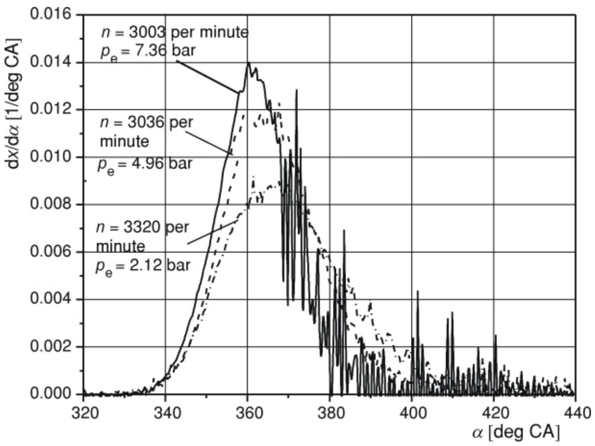

Figure 12. The influence of engine load on the rate heat release

speed. As it can be seen, engine load significantly influences the burning velocity. At all three loads the combustion starts at the same CA position (approx. 335 deg CA), but in the case of smallest mean effective pressure (p_e = 2.12 bars) combustion is substantially slower. The main reason is the increased residual gas concentration, which slows down the combustion process.

Figure 12 also shows that some analyzed pressure records contain a substantial noise, producing the oscillations in calculated rate of heat release. This could be eliminated by applying some curve smoothing technique on pressure record before the heat release evaluation.

Compression ignition engine

The experimental installation was similar as in the case of SI engine. The main difference was that the pressure transducer Kistler type 7507 SN79399, angular encoder with 180 marks per revolution and data acquisition and control system ADS 2000 [14] were used. The encoder angular increment of 2 deg CA was divided by factor 8 using the data acquisition system hardware, so the actual pressure recording increment was 0.25 deg CA. For heat release analysis the mean value of 50 consecutive cycles was used without any curve smoothing technique prior the calculation. The main engine data are given in tab. 3.

Figure 13 shows the results of pressure diagram analysis for one of recorded engine operating conditions, namely rate of heat release $dQ_f/d\alpha$, cumulative heat released Q_f, and the mean gas temperature T. It can be noticed that analyzed pressure record contains a substantial measuring noise producing the oscillations in the evaluated rate of heat release. In spite of these oscillations, a characteristic shape of direct injection CI engine rate of heat release can be recognized. The negative slope at approx. 345-350 deg CA caused by injected fuel evaporation, is followed by the expressive peak due to premixed combustion typical of part load operation (p_e = = 2.67 bars). The diffusion phase is relatively small and represents a small portion of the whole event.

The recorded injector needle lift h_n is also shown in fig. 13 (dashed line). It can be seen that the dynamic injection timing was approx. 18 deg CA before TDC. As the combustion process started at approx. 351 deg CA the ignition delay period was ~9 deg CA. The long delay facilitates evaporating and mixing and explains large premixed burning spike. Combustion process ends at approx. 410 deg CA, so the total combustion duration of ~60 deg CA is relatively short for CI engine, but reasonable considering part load operation.

In fig. 14 the same results are shown in normalized form. The maximum normalized heat release rate dx/dα reaches the value 0.05 what is very high, several times higher than in previous example of SI engine. The final value of the normalized cumulative heat release x of approx. 0.99 shows the high combustion efficiency, and indicates satisfactory measuring and calculating accuracy.

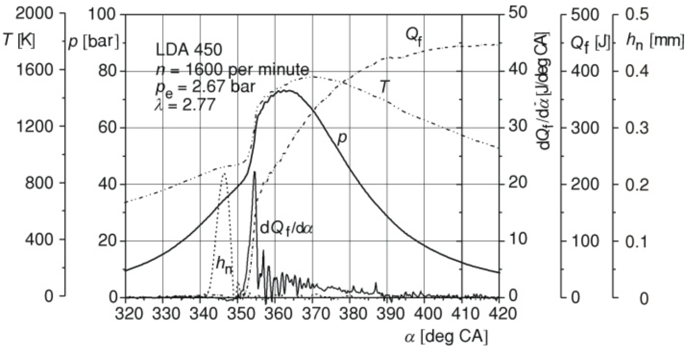

Figure 13. The results of pressure trace analysis: rate of heat release dQ_f/dα, cumulative heat released Q_f, and mean gas temperature T; h_n injector needle lift

Table 3. The main CI engine technical data

Engine type	LDA 450 CI with direct injection
Producer	DMB Belgrade, FMM
Bore	85 mm
Stroke	80 mm
Number of cylinders	1
Compression ratio	17.5
Max. power output	7.3 kW/3600 rpm
Cooling system	air
Fueling system	high pressure pump, injector with 4 fuel jets

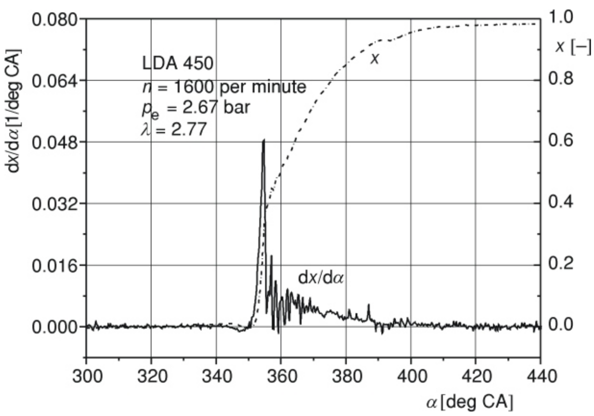

Figure 14. Normalized rate (dx/dα) and cumulative (x) heat release

Conclusions

Described method for evaluation of the rate of heat release from IC engine indicator diagram enables quick and simple combustion analysis. It considers in simplified way the heat transferred to the gas between two consecutive recorded points, uses simple expressions for gas thermodynamic properties, and calculates heat transfer to the walls using a phenomenological model. This provides an extremely computationally efficient technique.

The error analysis has shown that the error is reduced by decreasing pressure recording angular increment, and that with fine enough increment it becomes negligible. The errors due to simplified gas thermodynamic properties also remain at the level of few percents. In comparison, the experimental inaccuracies such as the errors in pressure measurements, measured fuel and air mass, and CA synchronization can produce much grater errors than simplifying assumptions in the model. As an example, special care should be paid to pressure to CA synchronization (TDC position error). The error of only 1 deg CA in TDC position produces the error of approx. 5% in final calculated heat release value and changes significantly the shape of the calculated rate of heat release, giving false information about burning velocity.

The analysis of the sensitivity to heat transfer calculation indicates that the slope of the heat release curve at the end of combustion can be used for qualitative assessment of the validity of the heat transfer coefficient estimation.

The noise superposed on recorded pressure can produce oscillations in calculated rate of heat release. This problem can appear even when mean value of many engine cycles is used and can be probably reduced by applying some of curve smoothing techniques.

Considering the described method simplicity, which makes it to be "easy to use", and, on the other hand, its satisfactory accuracy, it can be claimed very convenient for combustion analysis in engine testing practice. It cloud be even incorporated in engine testing control system to provide "on line" information about combustion.

Acknowledgment

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Nomenclature

A – area, surface area, [m^2]
A_w – surface area of the wall, [m^2]
c_m – piston mean velocity, [ms^{-1}]
c_p – specific heat at constant pressure, [Jkg^{-1}K^{-1}]
c_v – specific heat at constant volume, [Jkg^{-1}K^{-1}]
H – enthalpy, [J]
h – specific enthalpy, [Jkg^{-1}]
h_f – thermal enthalpy of fuel, [Jkg^{-1}]
H_f – lower fuel caloric value, [Jkg^{-1}]
h_n – injector needle lift, [mm]
L_0 – stoichiometric amount of air, [–]
m – mass, [kg]
m_a – air mass, [kg]
m_{cr} – mass flowing into crevice, [kg]
m_f – fuel mass, [kg]
m_{fb} – fuel mass burned, [kg]
n – engine speed, [rpm]
p – pressure, [Pa]
p_e – mean effective pressure, [bar]
Q – heat, [J]
Q_{ER} – heat calculation error, [J]
Q_f – heat released by combustion, [J]
Q_{f0} – amount of heat brought by fuel into cylinder per cycle, [J]
Q_n – net heat released (heat transferred to the gas), [J]
Q_w – heat transferred to the walls, [J]
R – special gas constant, [Jkg^{-1}K^{-1}]
R_{cp} – gas constant of combustion products, [Jkg^{-1}K^{-1}]
R_{mix} – gas constant of fuel-air mixture, [Jkg^{-1}K^{-1}]
R_{unb} – gas constant of unburned gas, [Jkg^{-1}K^{-1}]
S – entropy [JK^{-1}]
s – specific entropy, [Jkg^{-1}K^{-1}]
t – time, [s]
T – temperature, [K]
T_w – wall temperature, [K]
U – internal energy, [J]
u – specific internal energy, [Jkg^{-1}]
V – volume, [m^3]
v – specific volume, [m^{3}kg^{-1}]
x – normalized heat released, [–]

Greek letters

α – crank angle, [deg CA]
α_w – heat transfer coefficient [$Jm^{-2}K^{-1}$]
α_{thloss} – thermodynamic loss angle, [deg CA]
κ – ratio of specific heats at constant pressure and at constant volume, [–]
λ – air access ratio, [–]

Abbreviations

CI – compression ignition
IC – internal combustion
SI – spark ignition
TDC – top dead centre

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APPENDIX – Thermodynamic properties of the working fluid

For the evaluation of eq. (17) the thermodynamic properties of the gas in combustion chamber are required. Since one of the main benefit of the proposed method is to be simple to use, but yet enough accurate, several approaches, appropriate expressions and its effect on the calculation results has been analyzed.

The simple expression for instantaneous specific heat at constant volume c_v has been proposed in [9]:

$$c_{\mathrm{v}} \quad 0.7 \quad T \;\; 10^{\;\,3}(0.155 \quad A)\,[\mathrm{kJ/kg\,K}] \tag{a1}$$

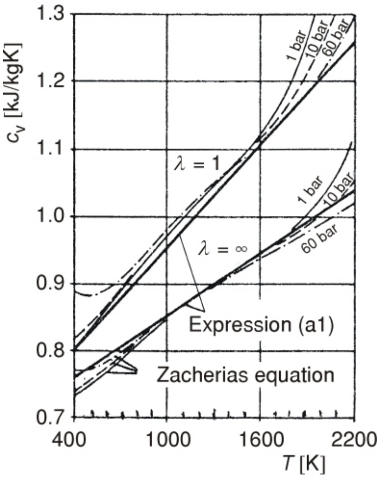

Figure A1. Comparison of specific heat according to Zacharias and expression (a1) [9]

For SI engines parameter A takes a constant value 0.1, and for CI engines A depends on mixture strength, A = = $1/10\lambda$, where λ is air excess ratio. The linear expression (a1) is quite simple; nevertheless the authors show that accuracy is sufficient for most cases. Figure A1 compares the results obtained using the expression (a1) and values resulting from Zacharias' equations.

However, expression (a1) in the case of SI engine doesn't take under account mixture strength which can normally vary in the range λ = 0.8-1.2 (in some cases, for example stratified charge or HCCI engine even much more). The expression originating from linear regression of data obtained using Zacharias' model for combustion products thermodynamic properties is given in [15]. It is also very simple and takes under account the influence of air excess ratio in the case of SI engine by varying coefficients for reach and lean mixture:

$$c_{\mathrm{v}} \quad 692 \quad 0.15T \quad \frac{1}{\lambda}(17.5 \quad 0.1094T)\,[\mathrm{J/kgK}] \text{ for CI engines and SI engines for } \lambda \quad 1$$

$$c_{\mathrm{v}} \quad 558.62 \quad 0.3669T \quad \frac{1}{\lambda}(150.88 \quad 0.10747T)\,[\mathrm{J/kgK}] \text{ for SI engines } \lambda \quad 1 \tag{a2}$$

The results obtained using expression (a2) and the real values taking under account gas reality and combustion products dissociation (according to Zacharias) are compared in fig. A2 [7]. According to [7] the maximum differences are under 2.5%.

The results obtained using expressions (a1) and (a2) are compared in fig. A3. As it can be seen the differences are small and don't exceed 2%. Figure A1 and A2 show that both expressions enable good approximation of the values resulting from Zacharias' model (mostly considered as accurate) in the range relevant for heat release rate evaluation. The significant disagreement appears only for high temperature and low pressure, but this situation is not possible in normal engine process. Small differences between expression (a1) and (a2) can not produce visible difference in calculating result (se also fig. 6). However, the expression (a2) has been adopted for further work since it provides slightly better approximation in the case of SI engine.

For most cases a gas constant R can be taken as for ideal gas. The influence of gas reality is for normal pressure/temperature range at the level of max. 2.5% (for example for p = 100 bar and T = 2500 K the difference is ~2.1%) [7]. In the case of ideal gas, a gas constant depends only on gas composition. For CI engines, before the start of combustion, pure air has been compressed and gas constant has the value 287 J/kgK. In the case of SI engines, the fuel-air mixture has been compressed and, with the assumption that fuel is octane (C_8H_{18}), gas constant of the mixture can be calculated as:

$$R_{mix} = \frac{8314}{28.96 + \dfrac{85.04}{1 + 59.5\lambda}}\ [\mathrm{J/kgK}] \quad \text{(a3)}$$

Gas constant of combustion products depends on fuel composition and mixture strength. If usual gasoline and diesel fuels are considered the differences due to fuel composition are very small and can be neglected. With sufficient accuracy combustion products gas constant R_{cp} can be calculated using the equations:

$$R_{cp} = 372.6 - 82.7\lambda\ [\mathrm{J/kgK}]$$

for SI engines with rich mixture opearion ($\lambda < 1$) (a4)

$$R_{cp} = 290.65 - 0.5\lambda\ [\mathrm{J/kgK}]$$

for CI and SI engines with stoichiometric and (a4)

The expressions (a4) are the linear fit of the values calculated using theoretical exhaust gas composition (evaluated by means of stoichiometric equations for C8H18 and C16H34 fuel-air mixture combustion). The values obtained theoretically as well as the fit curves are shown in fig. A4.

During the combustion process the gas in combustion chamber can be considered as the mixture of unburned gas and combustion products, and gas constant can be calculated from the relation:

$$R = \left(1 - \frac{Q_f}{Q_{f0}}\right) R_{unb} + \frac{Q_f}{Q_{f0}} R_{cp} \quad \text{(a5)}$$

where gas constant of unburned gas R_{unb} has the value 287 for CI and R_{mix} for SI engines.

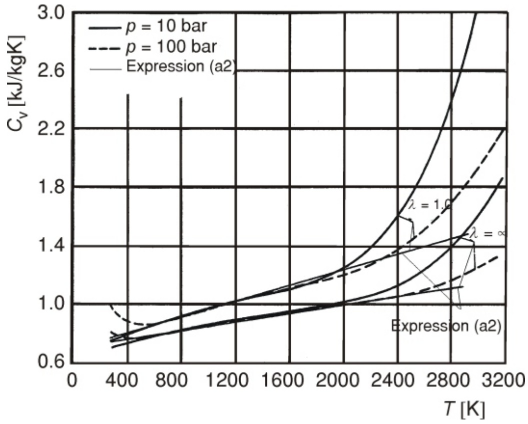

Figure A2. Comparison of the real specific heat (taking under account gas reality and combustion products dissociation according to Zacharias) and the results obtained using expression (a2) [7]

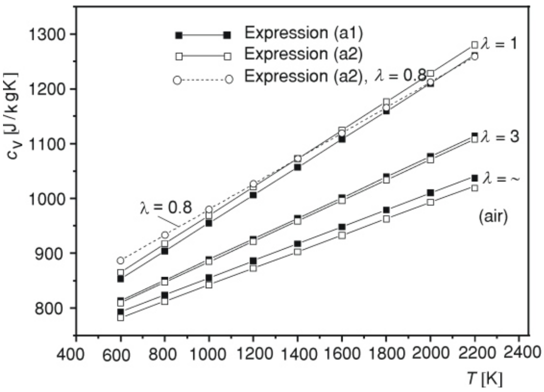

Figure A3. Comparison of the results obtained using expressions (a1) and (a2)

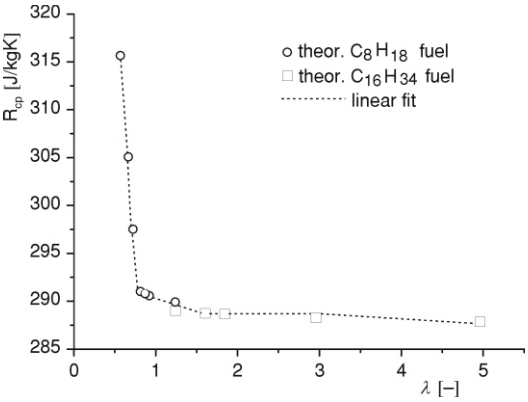

Figure A4. Gas constant of combustion products R_{cp}

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