

The Simple Method for IC Engine Heat Release Evaluation from Cylinder Pressure Record

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Abstract— This paper presents the quick, simplified method for the evaluation of heat release rate and combustion analysis on the bases of in cylinder pressure record in IC engine. The method is based on the approach proposed by Hohenberg and Killman, and it is extended by dint of the evaluation of heat transferred to the cylinder walls omitted in original approach. Therefore, the complete heat released by fuel combustion was considered. The accuracy of the method was also analyzed and it is shown that errors caused by the simplifications in the model were very small. In order to demonstrate the method it was applied for the heat release analysis of tested diesel engine fuelled by standard mineral diesel fuel and pure biodiesel fuel. Combustion characteristics of two applied fuels were analysed and compared.

Keywords— Internal combustion engine, combustion, pressure diagram, rate of heat release, biodiesel.

I. INTRODUCTION

Engine performance parameters (power output, torque, fuel consumption) as well as environmental characteristics (exhaust emissions, noise) are normally measured during engine testing so that the global engine efficiency can be recognized. However, more sophisticated informations about in-cylinder processes are necessary in order to have insight in the sources and distribution of energy losses, exhaust gas pollutants formation etc. The combustion is the key process that crucially influences both engine performances as well as exhaust and noise emissions. The informations about combustion dynamics is of prime importance for engine working process improvement and proper control.

One of the most frequent way to obtain necessary information as regards working process is recording of the in-cylinder pressure trace, or so-called “indicator diagram”. Even without any calculation the pressure record provides some important informations about combustion, for example: peak

pressure and its position, the rate of pressure rise (influencing basically combustion noise) etc. Computation of mean indicated pressure of high 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, more sophisticated thermodynamic analysis of the pressure data is required. The rate of heat release due to combustion or simply “the rate of combustion” and the mean gas temperature are the major results of such analysis.

With the introduction of digital computers and their application in engine working cycle modelling, the computer-aided methods for the analysis of engine indicator diagram have been developed. The works of Vibe [1] and Lange and Woschni [2] are well known, but the most comprehensive analysis was reported by Krieger and Borman [3] and their method has been regarded as a reference for IC engine indicator diagram analysis. Another remarkably in-depth work, introducing the number of influences, was reported by Jankov [4]. A good survey over numerous approaches and methods as well as appropriate discussion was given by Heywood [5]. The development and application of digital data acquisition systems in engine testing have further gained the possibilities of measurement of high speed engine variables and significantly improved the accuracy of collected data.

All the methods for heat release evaluation from recorded indicator diagram are based on the first law of thermodynamic and strive to take into account the reality of physical processes that occur in the cylinder during combustion. Thus, real thermodynamic properties of cylinder content depending on temperature, pressure and gas composition are considered, mass flow and losses through crevices are included etc. Krieger and Borman [3] 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, but, in general, models remain fairly complex for routine engineering practice. The method explained in the text below is very simple and easy for programming, but still accurate enough for practical purposes. Its practical application is very simple and low cost, and it could be even incorporated in engine testing control system to provide “on line” information about combustion.

II. SIMPLIFIED METHOD OF RATE OF HEAT RELEASE EVALUATION

In reality, engine combustion chamber is an open thermodynamic system, even during high pressure part of the cycle (when intake and exhaust valves are closed) due to the mass flow and loss through crevices between piston rings and cylinder liner. This is schematically illustrated in Fig. 1. The basic equation of the first law of thermodynamic applied to an open system is:

$$dQ = dU + pdV \quad (1)$$

Here, dQ is the change of elementary energy entering/leaving the system (except kinetic energy which is omitted), dU is the elementary change of cylinder charge internal energy and $p \cdot dV$ is elementary mechanical work delivered to the piston.

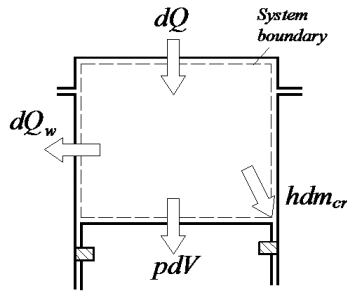


Fig.1 Engine cylinder as an open thermodynamic system

The derivative dQ consists of the energy released by fuel combustion dQ_f , the energy transferred to the walls dQ_w and energies taken into/out the system by means of the mass flows through the boundaries $\sum h_i dm_i$, where h_i and dm_i are the enthalpies and elementary mass flows respectively. As it is already stated, mass flow through crevices during high pressure part of the cycle provided that proper cylinder sealing is encountered, is very small and can be neglected. Thus, it can be written:

$$dQ = dQ_f - dQ_w \quad (2)$$

and, after simple evaluation:

$$dQ_f = mdu + pdV + dQ_w \quad (3)$$

where m and du is the mass and elementary change of specific internal energy of cylinder content.

The proposed simplification is based on the fact that the 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 encoders enable very fine angular increment ($\Delta\alpha$), the changes in pressure and volume are small. Therefore, 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. 2), the amount of fuel chemical energy ΔQ_f is released by fuel combustion. If we neglect the effect of crevice flow, this heat is partly transferred to the gas ΔQ_g and partly transferred to the walls ΔQ_w . Then, we can write:

$$\Delta Q_f = \Delta Q_g + \Delta Q_w \quad (4)$$

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

A. Heat transferred to gas

Hohenberg and Killman [6] proposed a simplified method for the evaluation of the amount of heat transferred to gas ΔQ_g . The real process (1 to 2) is virtually divided into two steps (Fig. 2).

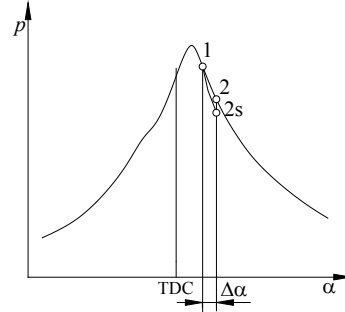


Fig.2 Recorded p-V diagram; gas state change from point 1 to point 2

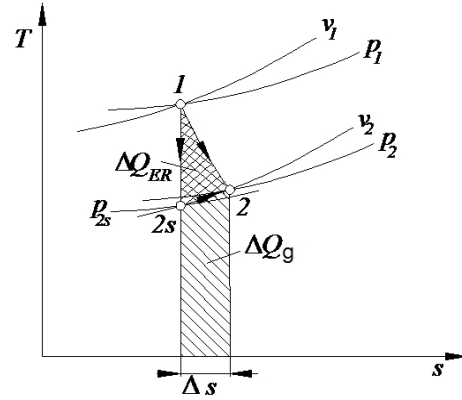


Fig.3 Gas state change from point 1 to point 2 in T-s diagram

First step is the adiabatic and isentropic expansion (or compression) from point 1 to point 2s (from volume V_1 to volume V_2) with no heat transfer to the gas. In the second step, the heat added to the gas due to combustion is considered as at constant volume ($V_2 = \text{const.}$). The required parameters in characteristic points can be obtained using the equation of state for ideal gas and the equation for isentropic state change. If gas behaviour is considered as ideal (which is, actually, not too far from real process) we can write:

$$p_{2s} = p_1 \left(\frac{V_1}{V_2} \right)^\kappa ; \quad \kappa = \frac{c_p}{c_v} ; \quad c_p = c_v + R \quad (5)$$

$$T_1 = \frac{p_1 \cdot V_1}{m \cdot R} ; \quad T_2 = \frac{p_2 \cdot V_2}{m \cdot R} \quad (6)$$

$$T_{2s} = \frac{p_{2s} \cdot V_2}{m \cdot R} = \frac{p_1 \cdot V_1^\kappa \cdot V_2^{1-\kappa}}{m \cdot R} \quad (7)$$

$$\Delta Q_g = m \cdot c_v (T_2 - T_{2s}) = \frac{c_v}{R} \cdot V_2 (p_2 - p_{2s}) \quad (8)$$

Here R is gas constant and c_v is gas specific heat at constant volume.

Finally, after appropriate operation, we obtain the amount of heat released due to combustion that is transferred to the gas between points 1 and 2 as:

$$\Delta Q_g = \frac{c_v}{R} \cdot V_2 \left[p_2 - p_1 \left(\frac{V_1}{V_2} \right)^\kappa \right] \quad (9)$$

The calculation of the heat transferred to the gas ΔQ_g according to equation (9), is an approximation which can be clearly seen in entropy-temperature diagram shown in Fig. 3. The difference between real heat transferred to the gas ΔQ_{gr} (proportional to the area under the curve 1-2) and calculated value ΔQ_g (proportional to the area under the curve 2s-2) is being designated as ΔQ_{ER} (error). This difference can be estimated using the entropy rise Δs (Fig. 3) which is the same for real process (1-2) and approximation considered (2s-2) [7]:

$$\Delta s \approx \frac{2 \cdot \Delta Q_{gr}}{T_1 + T_2} \quad (10)$$

$$\Delta Q_{ER} \approx \frac{T_1 - T_{2s}}{2} \cdot \Delta s = \Delta Q_{gr} \frac{T_1 - T_{2s}}{T_1 + T_2} \quad (11)$$

After substituting (6) and (7) in (11) the relative error can be expressed as:

$$\frac{\Delta Q_{ER}}{\Delta Q_{gr}} \approx \frac{1 - \left(\frac{V_2}{V_1} \right)^{1-\kappa}}{1 + \frac{p_2}{p_1} \frac{V_2}{V_1}} \quad (12)$$

The error is negative before TDC (calculated ΔQ_g is greater than the real heat transferred to gas ΔQ_{gr}) and positive after TDC (calculated ΔQ_g is less than the real heat transferred to gas ΔQ_{gr}). It is obvious that the outmost factor is the volume change between the points 1 and 2. The less is the volume change, the less is the calculation error. Fig. 4 and 5 show the recorded pressure and volume changes as well as relative error expressed in % ($ERROR = 100 \cdot \Delta Q_{ER} / \Delta Q_{gr}$), for two angular increments $\Delta\alpha = 1$ CA deg. and $\Delta\alpha = 0.2$ CA deg.

Obviously, for sufficiently small angular increment the relative error is getting negligible. For angular increment

resolution of $\Delta\alpha = 0.2$, which was used in the examples given in section III, relative error does not exceed $\pm 0.15\%$.

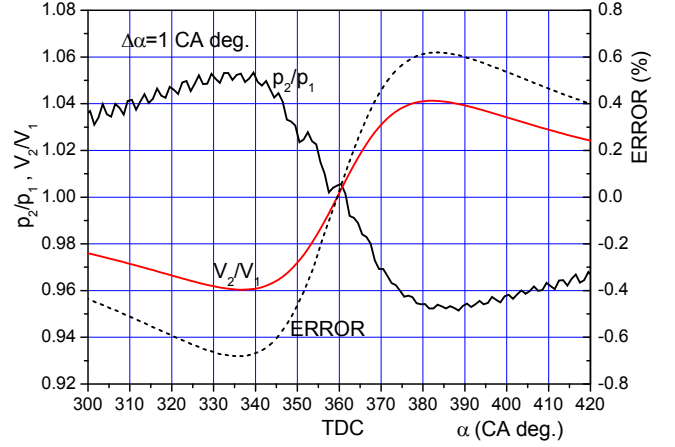


Fig.4 Pressure, volume and relative error changes for pressure recording angular increment $\Delta\alpha = 1$ CA deg.

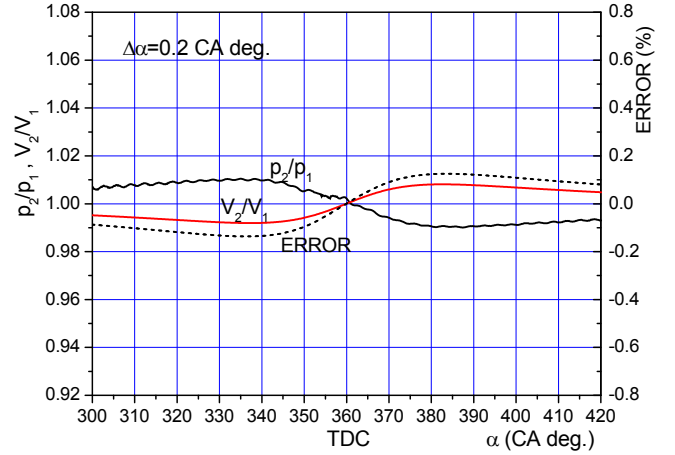


Fig.5 Pressure, volume and relative error changes for pressure recording angular increment $\Delta\alpha = 0.2$ CA deg.

For the evaluation of equation (9), the thermodynamic properties for the gases in engine cylinder are required. The approximate expressions for gas constant R and gas specific heat at constant volume c_v and discussion about their accuracy can be found in [7]. Expressions valid for lean mixture operation, used for diesel engine heat release evaluation presented in the example within section III are:

$$c_v = 692 + 0.15 \cdot T + \frac{1}{\lambda} (17.5 + 0.1094 \cdot T) \quad [Jkg^{-1}K^{-1}] \quad (13)$$

where λ is air excess ratio.

Gas constant R can be used as for ideal gas with a very small error [4]. In that case a gas constant depends only on gas composition. For diesel engines, before the start of combustion, pure air has been compressed and gas constant has the value $287 \text{ [Jkg}^{-1}\text{K}^{-1}\text{]}$. Gas constant of combustion products depends on fuel composition and mixture strength. In the case of usual diesel fuels considered the differences due to fuel composition are very small and can be neglected. With sufficient accuracy gas constant of combustion products R_{cp} can be calculated using the expression [4,11]:

$$R_{cp} = 290.65 - 0.5 \cdot \lambda \quad [Jkg^{-1}K^{-1}] \quad (14)$$

B. Heat transferred to the cylinder walls

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

$$\frac{dQ_w}{dt} = \alpha_w \cdot A_w (T - T_w) \quad (15)$$

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 bounding 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 due to lack of accurate data for different parts. Thus we have:

$$\frac{dQ_w}{dt} = \alpha_w \sum_i A_{wi} (T - T_{wi}) \quad (16)$$

Usually, piston crown, cylinder head and cylinder liner are considered as the elements in expression (16), the last having variable surface according to piston motion and variable temperature from the top (at TDC) to the bottom (at BDC).

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

$$\Delta Q_w = \left[\alpha_w \sum_i A_{wi} (T - T_{wi}) \right] \Delta t = \left[\alpha_w \sum_i A_{wi} (T - T_{wi}) \right] \frac{\Delta \alpha}{6 \cdot n} \quad (17)$$

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 [8] which is relatively simple and convenient for use:

$$\alpha_w = 0.013 \cdot V^{-0.06} \cdot p^{0.8} \cdot T^{-0.4} (c_m + 1.4)^{0.8} [\text{Wm}^{-2}\text{K}^{-1}] \quad (18)$$

where V is instantaneous cylinder volume, p and T cylinder pressure and temperature and c_m mean piston speed. Alternatively, some other models, for example Woschni's or Annand's can be used.

III. SOME EXAMPLES OF METHOD APPLICATION

In order to elucidate the method described some results of the research of combustion process in diesel engine fuelled with standard diesel fuel and rape methyl ester (RME) are presented [10]. All experiments were carried out in the laboratory of IC engine department, Faculty of Mech. Eng. in Belgrade. In addition to standard engine testing equipment (dynamometer with engine torque and speed measuring system, intake air and fuel flow measuring systems, inlet air, exhaust gases, engine coolant and oil temperatures measuring system), the engine was equipped with the system for in-cylinder pressure recording. This equipment includes pressure transducer Kistler type 7031 mounted directly to combustion chamber, charge amplifier Kistler type 5001, angular encoder installed on engine crankshaft type COM1 and data acquisition system ADS 2000 developed in the IC engines department.

Angular encoder provides angular increment of 1 deg. (360 marks per revolution) which is by dint of software divided into five parts, so that the angular resolution of

pressure sampling was 0.2 CA deg. 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 crank angle synchronization, since these problems could significantly influence the results of heat release analysis. For this purpose the methodology reported in [6] and [9] was applied.

More details about testing equipment and procedure can be found in [10]. Main engine specifications are given in Table I and the characteristics of fuels used in Table II.

TABLE I
MAIN SPECIFICATIONS OF TEST ENGINE

| | |
|---------------------|--|
| Engine type | LDA 450 Diesel, 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 jets |

TABLE III
CHARACTERISTICS OF FUELS USED

| | DIESEL 100 | RME 100 |
|---|------------|---------|
| Density [kg l^{-1}] | 0.828 | 0.880 |
| Lower caloric value [kJ kg^{-1}] | 41494 | 37631 |
| Kin. viscosity [$\text{mm}^2 \text{s}^{-1}$] | 3.16 | 4.59 |
| Stoichiometric air mass [kg kg^{-1}] | 15.08 | 12.65 |
| O ₂ content [kg kg^{-1}] | 0 | 0.120 |
| C content [kg kg^{-1}] | 0.8496 | 0.772 |
| H ₂ content [kg kg^{-1}] | 0.1504 | 0.120 |

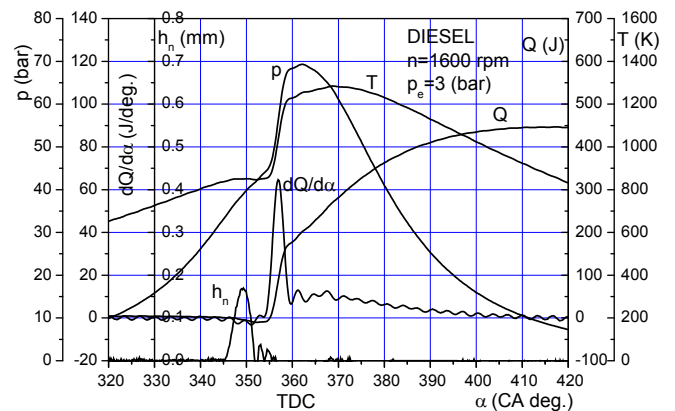


Fig. 6 Recorded pressure p , injector needle lift h_n , rate of heat release $dQ/d\alpha$, cumulative heat released Q and mean gas temperature T ; fuel standard diesel

Fig. 6 shows the results of pressure diagram analysis using described method for one of recorded engine operating conditions using diesel fuel: the rate of heat release $dQ_f/d\alpha$, cumulative heat released Q_f and the mean gas temperature T . The mean gas temperature is calculated using the equation of state for ideal gases.

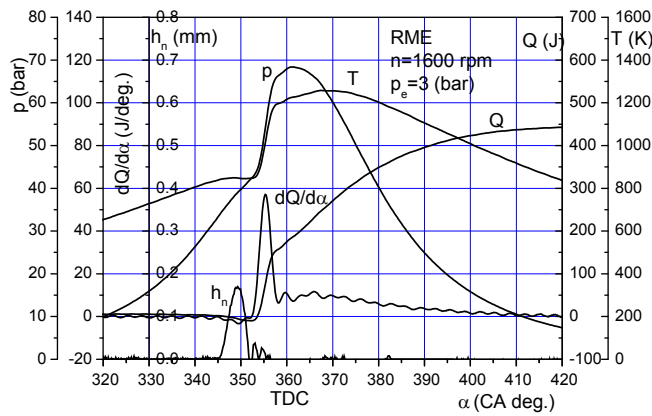


Fig. 7 Recorded pressure p , injector needle lift h_n , rate of heat release $dQ/d\alpha$, cumulative heat released Q and mean gas temperature T , fuel pure RME

Fig. 7 shows the same data in the case of engine operation with pure RME. In both cases engine speed and load were the same, 1600 rpm and mean effective pressure $p_e=3$ bars.

It can be noticed that for both fuels the rate of heat release has the form that is typical for diesel engines with direct injection. High peak in the phase of premixed combustion, which reaches app. 60 J/deg, is followed by relatively low intensity and lengthy combustion in diffusion phase. The duration of premixed phase is app. 5-6 CA deg. while total combustion duration is about 60 CA deg.

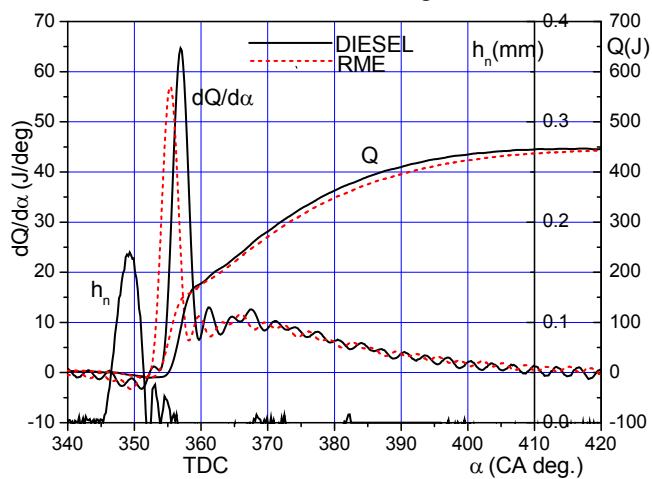


Fig. 8 The comparison of combustion dynamics of standard diesel (full line) and pure RME (dashed lines); h_n needle lift, $dQ/d\alpha$ the rate of heat release, Q cumulative heat released

It is interesting to compare combustion dynamics of standard diesel fuel and pure RME. This is shown in Fig. 8. The curves for needle lift are nearly coincident for both fuels, i.e. the differences in fuel densities and viscosity does not affect significantly the dynamics of fuel injections. The dynamic injection timing is approx. 15 CA deg. before TDC. However, the start of combustion differs significantly for two fuels. In the case of RME fuel combustion commences app. 2 CA deg. earlier, i.e. ignition delay period is app. 2 CA deg. shorter. Ignition delay periods for RME and diesel fuel are app. 7 and 9 CA deg. respectively. Taking into account that engine speed is 1600 rpm, the duration of ignition delay is 0.73 ms for RME, and app. 0.93 ms for diesel fuel. Shorter

ignition delay means higher cetane number of RME with a positive effect on premixed combustion yielding lower peak of the rate of heat release in premixed phase of combustion thereafter. However, in general the combustion of RME is slower and is finished in a few degrees later then in the case of diesel fuel, despite the fact that RME contains a considerable amount of oxygen, which should accelerate combustion. The reason is probably slower process of mixture formation due to poorer atomisation of the fuel spray caused by higher density and viscosity of RME.

IV. CONCLUSIONS

Described method for the evaluation of rate of heat release on the bases of pressure history in IC engine enables quick and simple combustion analysis. It considers in a simplified way the heat transferred to the gas, uses simple expressions for gas thermodynamic properties and calculates heat transfer to the walls using a phenomenological model.

The error analysis has shown that the error due to introduced simplification is less in the case of pressure angular increment record decrease. Sufficiently fine angular resolution yields the error negligible. The errors due to simplified gas thermodynamic properties persist also at the low level.

Taking into account that the method described is simple, and "easy to use", with satisfactory accuracy, it can be stated that it is very convenient for combustion analysis practice. It could be even incorporated in engine testing control system to provide "on line" information about combustion.

The comparison of combustion for standard mineral diesel fuel and RME, presented as an example of model application, provides very useful information of combustion dynamics of both fuels. It clearly shows that RME has higher cetane number and consequently shorter ignition delay period, but in general diesel fuel burns faster.

NOMENCLATURE

| | |
|----------|--|
| A | Area [m ²] |
| A_w | Surface area of the wall [m ²] |
| c_m | Piston mean speed [m s ⁻¹] |
| c_p | Specific heat at constant pressure [J kg ⁻¹ K ⁻¹] |
| c_v | Specific heat at constant volume [J kg ⁻¹ K ⁻¹] |
| H_f | Lower fuel calorific value [J kg ⁻¹] |
| h_n | Injector needle lift [mm] |
| L_0 | Stoichiometric amount of air [-] |
| m | Mass [kg] |
| n | Engine speed [rpm] |
| p | Pressure [Pa] |
| p_e | Mean effective pressure [bars] |
| Q | Heat, [J] |
| Q_{ER} | Heat calculation error [J] |
| Q_f | Heat released by combustion [J] |
| Q_g | Heat transferred to the gas [J] |
| Q_{gr} | Real heat transferred to the gas [J] |
| Q_w | Heat transferred to the walls [J] |
| R | Special gas constant [J kg ⁻¹ K ⁻¹] |
| R_{cp} | Gas constant of combust. products [J kg ⁻¹ K ⁻¹] |
| S | Entropy [JK ⁻¹] |
| s | Specific entropy [J kg ⁻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^3kg^{-1}] |

Greek letters

| | |
|------------|---|
| α | Crank angle, [deg CA] |
| α_w | Heat transfer coefficient [$\text{Wm}^{-2}\text{K}^{-1}$] |
| κ | Ratio of specific heats at const. pressure and at const. volume [-] |
| λ | Air excess ratio [-] |

Abbreviations

| | |
|-----|---------------------|
| IC | Internal Combustion |
| RME | Rape Methyl Ester |
| TDC | Top Dead Centre |
| BDC | Bottom Dead Centre |

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