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Determination of specific heat ratio and error analysis for engine heat release calculations

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HIGHLIGHTS

- A new method is proposed to calculate specific heat capacities ratio (γ).
- Diesel fuel, Rapeseed Methyl Ester (RME) and Jatropha Methyl Ester (JME) are compared.
- Error calculation is done in this work for main combustion parameters.
- The parameter called "combustion burn factor (C_i)" is introduced.
- The benefits of using C_i in combustion analysis are presented.

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ABSTRACT

The burnt fraction *f* of Wiebe equation has been shown to be dependent only on the newly defined parameter 'combustion burn factor (C_i)'; and the benefits of expressing heat release rate with respect to C_i have been presented. The errors associated with the determination of apparent heat release rate (*Ahrr*) and the cumulative heat release (*Cum.Hrr*) from the measured cylinder pressure data and the assumed specific heat ratio (γ) was determined and compared. The γ affected the calculated *Ahrr* more than the cylinder pressure. Overestimation of γ resulted in an underestimation of the peak value of the *Ahrr* and vice versa, this occurred without any shift in the combustion phasing. A new methodology has been proposed to determine the instantaneously and mean value of γ for a given combustion. This new methodology has been applied to determine γ for a wide range of engine operating conditions and for different fuels.

1. Introduction

Diesel engines are widely used in the field of transportation, agricultural and heavy industries sectors owing to their high efficiency, low fuel consumption, durability as well as low CO_2 emissions [1]. Control of emissions from diesel engines has been one of the most important challenges for the engine manufacturers. Stringent legislations have been proposed to reduce the harmful soot and NOx emissions from diesel engines which cause air pollution, affecting the human health and environment [2–9]. To address these challenges several advances have been made through the development of high pressure common rail fuel injection system, to precisely control the injection events and also for adopting different injection strategies [10–15]. In addition to this exhaust gas recirculation, boosting, and exhaust gas after treatment are other promising strategies that are widely used to develop cleaner diesel engines [16–21]. Fuel type, quality and its composition also play a vital role in reducing most of the engine out emissions [22–26]. The engine exhaust emission characteristics are strongly correlated to the in-cylinder combustion processes. The combustion characteristics in engines are mainly understood through the apparent heat release rate (*Ahrr*) that was determined from the first law of thermodynamics [1]. The *Ahrr* model without heat exchange to cylinder walls is shown in Eq. (1).

$$\left(\frac{dQ}{d\theta}\right) = \frac{\gamma}{\gamma-1} p \frac{dV}{d\theta} + \frac{1}{\gamma-1} V \frac{dp}{d\theta} \quad (1)$$

Where Q is the total released heat in J and θ is the instantaneous crank angle in degree, γ is the ratio of specific heat (C_p/C_v), p is the measured cylinder pressure in Pa and V is the cylinder volume in m^3.

The *Ahrr* is strongly related to engine operating conditions, engine specifications as well as physical and chemical properties of

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Nomenclature

Ahrr	apparent heat release rate
C_i	combustion burn factor
C_P	special heat capacity constant pressure
Cum.Hrr	cumulative heat release rate
C_v	special heat capacity constant volume
f	burn fraction
JME	Jatropha Methyl Ester
p	in-cylinder pressure
Q	total released heat
RF	Residual Function
RME	Rapeseed Methyl Ester
V	volume
V_c	clearance volume
α	relative apparent heat release rate error
β	absolute error on apparent heat release rate
$\Delta\theta$	combustion duration
γ	specific heat capacities ratio
θ	instantaneous crank angle
θ_{50}	the crank angle at centre of combustion
θ_{max}	the crank angle where Ahrr reaches to its maximum value
θ_0	the crank angle where Ahrr reaches to positive value

fuel. In addition to this the *Ahrr* provides information about ignition delay (the time interval between the start of injection and the start of combustion, the start of combustion is determined as the time instant when the *Ahrr* data crosses the time-axis after the start of injection), level of premixed and diffusion burn characteristics of the combustion process, which are useful for the understanding of exhaust soot and NOx emissions.

The burn fraction or burn rates of a combustion processes are normally characterised using Wiebe equation [27]. The Wiebe equation presents the relationship between the burnt fraction (*f*) and the three main combustion parameters, *viz*., (i) the instant at which heat release rate becomes positive, θ_0; (ii) instantaneous crank angle, θ; and (iii) the duration of combustion, $\Delta\theta$.

$$f = 1 - \exp\left[-a\left(\frac{\theta-\theta_0}{\Delta\theta}\right)^{m+1}\right] \quad (2)$$

In our recent work [28] the Wiebe equation has been presented in a modified form and it is expressed as:

$$f = 1 - \exp\left[(\ln 0.5)\left(\frac{\theta-\theta_0}{\theta_{50}-\theta_0}\right)^{m+1}\right] \quad (3)$$

The modified version of Wiebe equation has only one constant compared to the original Wiebe Eq. (2) with two constants. The constant '*m*' in Eq. (3) can be determined from the experimentally measured cylinder pressure data which has been demonstrated in [28], and the deduction of Eq. (3) has been given in Appendix [1]. The exponential term of the modified Wiebe equation contains parameters such as θ, θ_0 and θ_{50} and these parameters are combined to be expressed in the form of a non-dimensional parameter $\frac{\theta-\theta_o}{\theta_{50}-\theta_o}$. This non-dimensional parameter is called the *combustion burn factor* and it represents the combined effects of θ_0, θ_{50} and θ of each combustion condition and it is denoted by the notation C_i. Substitution of combustion burn factor in the modified Wiebe equation results in Eq. (4). It can be seen that C_i is the only parameter that affects the burnt fraction and it will be a valuable parameter to study the heat release rate analysis.

$$f = 1 - \exp[(\ln 0.5)C_i^{m+1}] \quad (4)$$

Many investigations have been reported to study the effects of combustion and emission process with respect to θ_0 and θ_{50} independently. It is for the first time the combined effect of these parameters on engine combustion performance was considered [28]. The benefits of using C_i for heat release analysis have been shown in the following discussion.

Fig. 1a and b shows the variation of *Ahrr* plotted against instantaneous crank angle for different engine operating conditions A2, A4, A6 and A7 (at a load of 2.7 bar BMEP) and A8, A11, A14 and A15 (at a load of 5 bar BMEP) for diesel fuel, details of the operating conditions are provided in Table 1. Since these conditions have different injection timing, the difference between θ_0 and θ_{50} for each condition are different, and the corresponding start of combustion and the end of combustion are different for each of these conditions. Consequently it makes difficult to compare the heat release data for different conditions. By using the non-dimensional parameter C_i to reconstruct *Ahrr* instead of instantaneous crank angle (θ) can produce a form of apparent heat release chart (Fig. 2a and b), which provides more information about combustion and it is easier to compare the data from different operating conditions.

In Fig. 1a and b, it is not clear where the centre of combustion is and the exact location where the combustion is taking place with respect to centre of combustion. These aspects can be viewed in Fig. 2a and b at any instant on the *Ahrr-Ci* chart. It is possible to

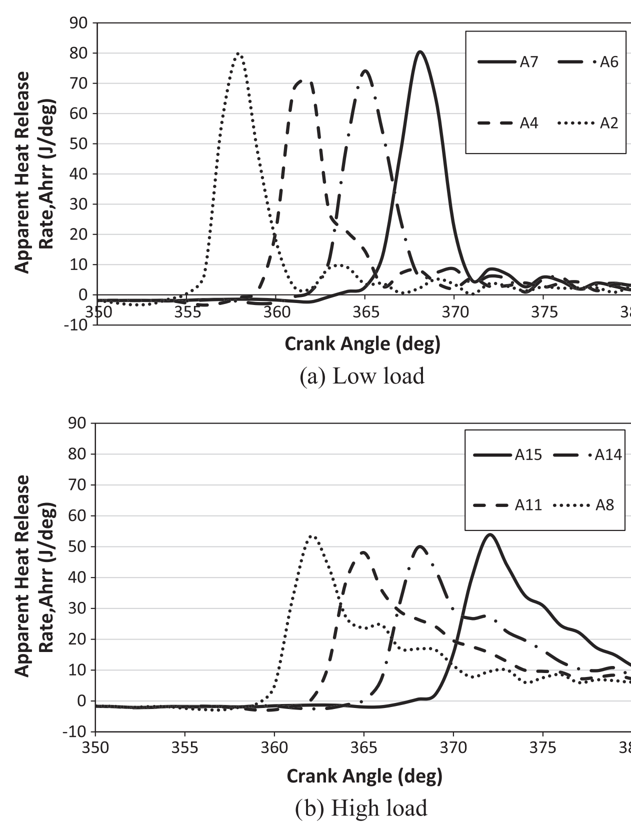

Fig. 1. (a and b) Apparent heat release rate against instantaneous crank angle (θ) at different engine operating conditions at low load and high load for diesel fuel.