HEAT RELEASE OF DIESEL ENGINE FUELLED WITH RME

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Abstract

Characteristics of the relative heat release quantity during combustion process have been appointed basing on the analysis of 100 engine work cycles received from experimental indicator diagrams. Indicator diagrams were obtained at work of the engine according to the external speed engine characteristics. The analysis of indicator diagrams of the engine oriented on to calculating characteristics of the heat release was realized at the regard of the change composition and the working charge quantity of kilo mole during combustion process and at neglect of losses of heat caused with the dissociation combustion products. The quantity heat exchanged between the working charge and walls combustion chamber was appointed basing on the empirical dependence for coefficient proposed by Woschni

Basic technical data of the AD3.152UR test engine, basic physical and chemical properties of fuels used in the researches, research stand scheme, scheme of the measurement system for fast changing quantities in piston compression ignition engine, changes in the injector needle lift, fuel pressure in the injection duct, open indicator diagram, exemplary averaged changes in the fuel pressure values in the injection duct as the function of crankshaft rotation angle, values of the maximum pressure differences in the nozzle and the engine cylinder, values of spray fuel jet penetration, graphic presentation of the method for the determination of ignition delay angle, values of self-ignition delay angle in an engine, analyses of the relative quantity and rate of the heat release during combustion are presented in the paper

Keywords: *internal combustion engines, diesel engines alternative fuel engines, rapeseed oil methyl esters, fuel injection, heat release*

1. Introduction

The average effects of biodiesel fuels on exhaust emissions are shown on Fig. 1. Generally, for example, impact of 20 vol. % biodiesel on emissions is level reducing of HC by 20 percent CO and PM by 10 percent and increasing of NO_x by 2 percent. In some cases reduction of the emission level NO_x is also possible [13]. Biodiesel fuel is also predicted to reduce fuel economy by 1-2 percent for a 20 volume percent biodiesel blend. Aggregate toxics are predicted to be reduced, but

the impacts differ from one toxic compound to another. We were not able to identify an unambiguous difference in exhaust CO_2 emissions between biodiesel and conventional diesel. However, it should be noted that the
CO_2 benefits commonly attributed to biodiesel are the result of the renewability of the biodiesel itself, not the comparative exhaust CO_2 emissions.

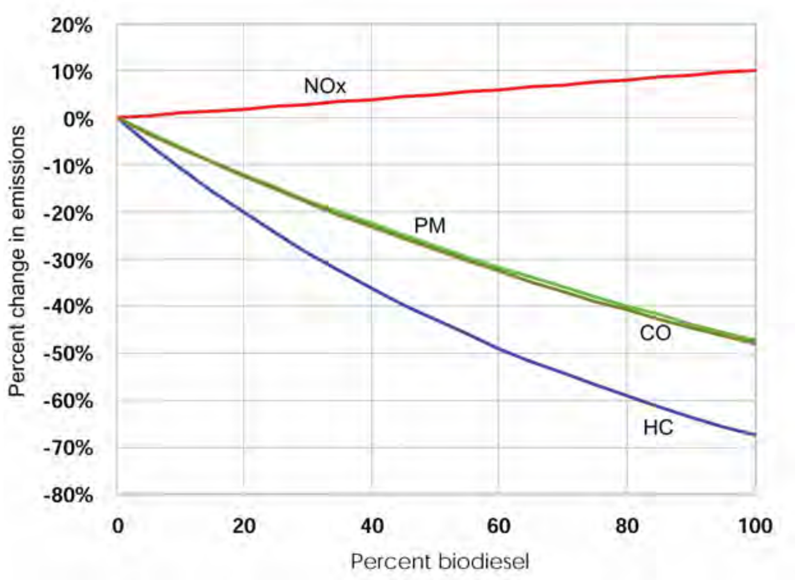


Fig. 1. Average emission impacts of biodiesel for heavy-duty highway engines [7]

The object under investigation was three-cylinder AD3.152UR CI engine. It had direct fuel injection, DPA 3328 F-510 injection pump and injectors with four-hole nozzles (3). The engine basic technical data is presented in Tab. 1.

Tab. 1. Basic technical data of the AD3.152UR test engine

Parameter	Unit	Value
Cylinder system	-	row
Number of cylinders	-	3
Kind of injection	-	direct
Sequence of cylinder operation	-	1 – 2 – 3
Compression ratio, ε	-	16.5
Cylinder diameter, γ	mm	91.44
Piston stroke, S	mm	127
Engine cubic capacity, V_{SS}	dm^3	2.502
Connecting rod length, l_K	mm	223.80÷223.85
Maximum engine power, Ne	kW	34.6
Maximum power speed of rotation, n_{max}	rpm	2250
Maximum torque, M_{emax}	Nm	168.7
Maximum torque speed of rotation, n_M	rpm	1350
Fuel injection advance angle, α_{WW}	°CAD	17
Idle run speed of rotation, n_l	rpm	750±50
Nozzle orifice diameter, d_r	mm	0.28
Nozzle channel length, l_r	mm	4

The aim of investigations was to experimentally determine the character of changes in fuel pressure, averaged over 100 cycles, in the injection duct and the working medium pressure in the engine cylinder and also the nozzle needle lift. Values of the above–mentioned parameters made it possible to determine the rate of the fuel flow out of the nozzle, the mean diameter of sprayed fuel droplets, the spreading angle of the injected fuel jet and its penetration. Those provided means to plot the diagrams of the relative amount of heat released during combustion. The measurements of the above-mentioned values were taken for the engine operating in accordance with external speed performance. The engine was fuelled by EKODIESEL PLUS 50B and BIODIESEL D-FAME, the basic physical and chemical properties of which are given in Tab. 2.

Tab. 2. Basic physical and chemical properties of fuels used in the investigations

PARAMETER	EKODIESEL PLUS 50 B	BIODIESEL D-FAME
Cetane Number, CN	51.5	51.3
Density at 20^{o}C (10^{-3} kg/m^3)	0.836	0.882
Kinematical Viscosity at 40^{o}C (10^{-6} m^2/s)	2.75	4.52
Calorific Value, MJ/kg	43.4	38.9
Surface Tension σ, 10^{-2}, N/m	3.47	3.55
Physical/ chemical properties for the diesel oil were given in accordance with the Polish and European standard PN-EN 590, for the FAME biofuel_- PN-EN 590:2005		

The investigations covered the measurements of pressures in the cylinder and the injection duct and also the injector needle lift as the function of the rotation angle of the engine crankshaft in the thermal steady state and operating under the conditions corresponding to the rated power and the maximum torque.

2. Research stand

The block diagram of the system for the measurements of quantities subject to rapid change is shown in Fig. 2.

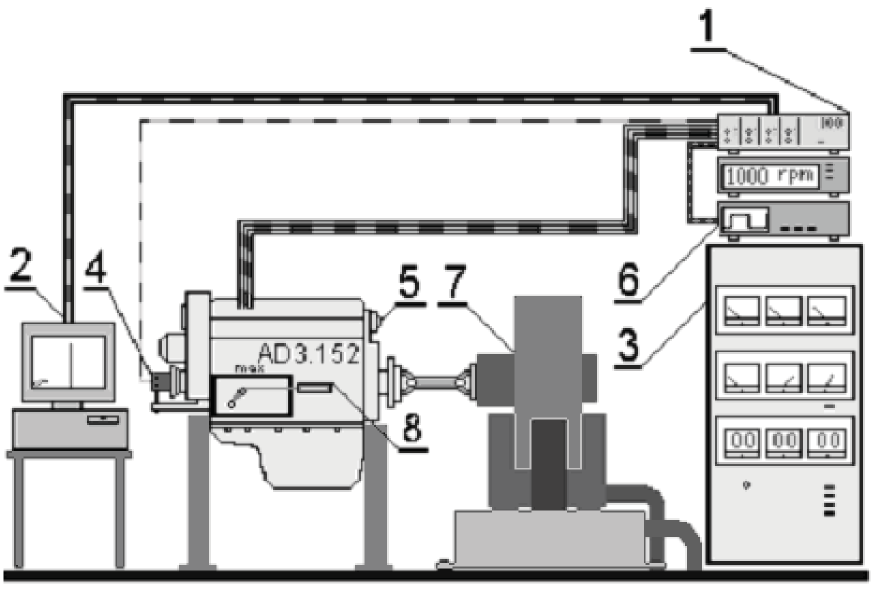


Fig.2. Research stand diagram [6]:1 - block of amplifiers of signals from piezoelectric sensors, 2 - PC with a measurement card, 3 - control and check module, 4 - transmitter of the crankshaft rotation angle, 5 - AD3.152 UR engine, 6 - time base generator, 7 - water brake, 8 - injection pump control slat

The engine under investigation was mounted at the engine test stand equipped with water brake and control / measurement devices. They made it possible to control the engine and break operation and also, to read and control the quantities measured. Amplifiers in the measurement paths enhanced signals generated by sensors. They were converted to voltage signals and then transferred to analogue-to-digital converter and recorded in the digital form.

3. Analysis of experimental results

Exemplary graphs of pressure changes in the cylinder and the injection duct and also the injector needle lift changes are shown in Fig. 3.

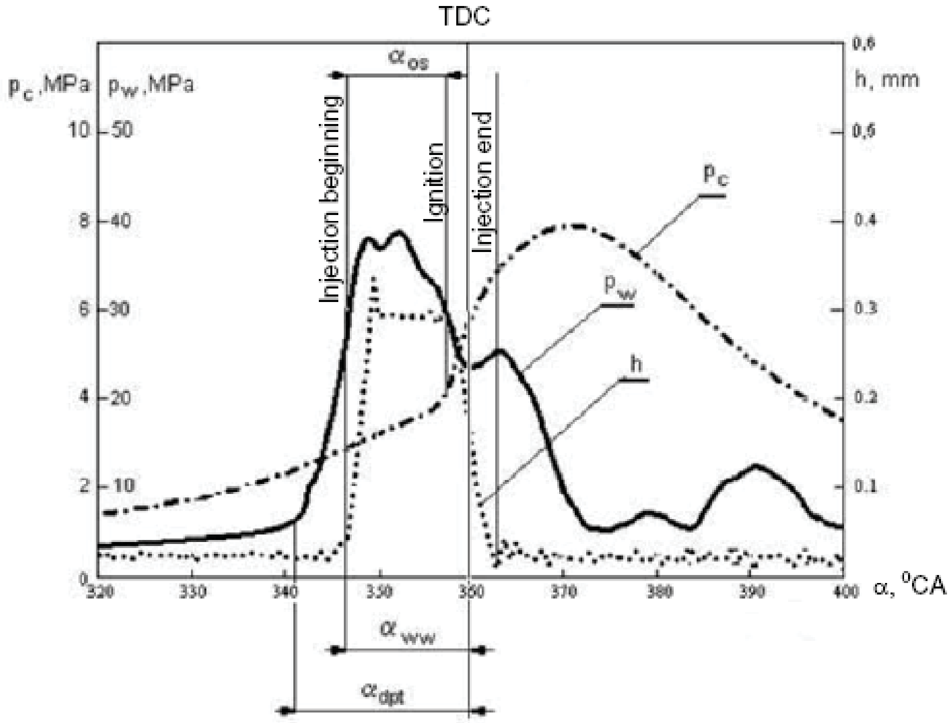


Fig.3. Changes in the injector needle lift (h), fuel pressure in the injection duct (p_w) and open indicator diagram (p_c) with marked angles: α_{dpt} - angle of dynamic injection beginning, α_{ww} - injection advance angle, α_{os} - ignition delay angle

4. Results of fuel injection researches

The way the volatile mixture is produced inside the engine significantly affects the combustion process and heat release in the cylinder. The character and parameters of fuel injection depend on the injection system type and its design parameters, the amount and quality of the injected fuel, the process phases and duration and also the parameters of the fuel jet sprayed in the cylinder.

Investigations were carried out at the engine test stand. They concerned the AD3.152 UR engine, operating under the conditions of rated power and the maximum torque. The engine was fuelled by two kinds of environmentally friendly fuels, i.e. low sulphur diesel oil EKODIESEL PLUS 50B and Fatty Acid Methyl Esters of Rapeseed Oil BIODIESEL D-FAME. The investigations were intended to determine changes, averaged over 100 cycles, in the pressure values in the cylinder, the injection duct and also the nozzle needle lift changes. Exemplary graphs of averaged fuel pressure values in the injection duct are presented in Fig. 4.

The following were computed on the basis of measurements results: the rate of fuel flow out of the nozzle, its critical value, at which droplet disintegration starts, the critical diameter value of the droplets formed at their secondary disintegration, the spread angle of the fuel jet cone, droplet Sauter mean diameter d_{32} and the jet range for $t \leq t_{break\text{-}up}$ and $t > t_{break\text{-}up}$. The calculation results (1, 4) are shown in Tab. 3, 4, 5, and 6.

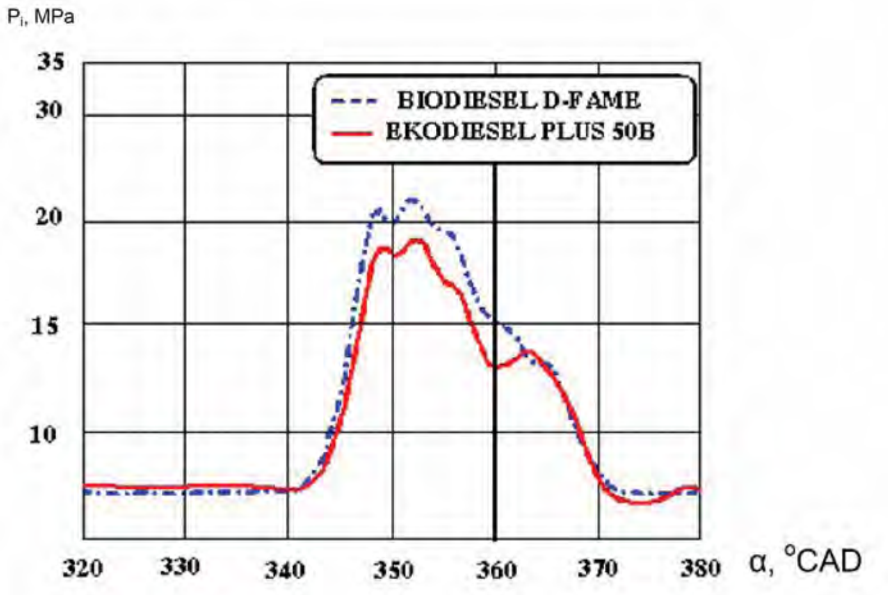


Fig. 4. Exemplary averaged changes in the fuel pressure values in the injection duct as the function of crankshaft rotation angle of AD3.152 UR engine operating at n = 1400 rpm and Me=165 Nm

Tab. 3. Values of the maximum pressure differences in the nozzle and the engine cylinder Δp and the speed initial w_p and critical w_e values of fuel droplets

Fuel	Pressure difference in the nozzle and the cylinder		The stream initial speed	Droplet critical speed
	ΔP	Pa	w_p, m/s	w_e, m/s
EKODIESEL PLUS 50-B Diesel Oil				
EKODIESEL PLUS 50-B	19	400	136.92	158.84
BIODIESEL D-FAME Biofuel				
BIODIESEL D-FAME	23	600	149.80	164.18

Tab. 4. Droplet Sauter Mean Diameter

Parameter	Diesel Oil EKODIESEL PLUS 50-B	Biofuel BIODIESEL D-FAME
Fuel charge, mm^3/ cycle	54	50
Droplet Sauter Mean diameter d_{32} μm	40.2	38.85

Tab. 5. Spread angles of spray fuel jet cone

Spray angles	Diesel Oil EKODIESEL PLUS 50-B	Biofuel BIODIESEL D-FAME
arctgΘ/2	0.1384	0.1366
tg Θ/2	7. 50^{o}	7. 40^{o}

Tab. 6. Values of spray fuel jet penetration

Parameter	Diesel Oil EKODIESEL PLUS 50-B	Biofuel BIODIESEL D-FAME
Disintegration time, ms	0.535	0.512
Jet penetration, mm for $t \leq t_{break-up}$	129	137
Jet penetration, mm for $t > t_{break-up}$	124	154

5. Heat release analyses

Owing to experimentally taken indicator diagrams of the working medium in the engine cylinder, it is possible to determine the self-ignition delay time and make analyses of heat release in the combustion process. The paper presents heat release analyses for AD3.152 UR engine operating under the conditions of rated power and the maximum torque, at the injection advance angle α_{ww} = 17 °CA, fuelled by the above-mentioned environmentally friendly fuels.

The self-ignition delay time was determined as the difference between the angular displacement of the crankshaft corresponding to the beginning of the combustion α_{ps} and the angle corresponding to the fuel injection beginning α_{pw} i.e. $\alpha_{os} = \alpha_{ps} - \alpha_{pw}$. The combustion beginning was stated as the intersection point of the curve representing change in temperature $T_p(\alpha)$, determined for the compression process, at the assumption that air is the working medium, with the curve $T_s(\alpha)$ also determined for pressure values for α $\alpha \in \left\langle \alpha_{pw}, \alpha_{TDC} \right\rangle$, yet at the assumption that in the cylinder there are the products of the total and complete combustion. In the experiments conducted by the authors so far, the methodology has proved very convenient and accurate enough. The method graphic representation is shown in Fig. 5. The results obtained for the tested engine are presented in Fig. 6. Relative amount of heat released during the combustion process as well as the heat release rate were analysed on the basis of an equation for the first law of thermodynamics and the state equation. Changes in the working medium composition and amount in the course of the combustion process and also heat transfer to the walls enclosing the combustion space were taken into account. Heat losses due to dissociation were disregarded [2]. The graphs showing the analyses of heat release during combustion in AD3.152 UR engine are presented in Fig. 7 and 8.

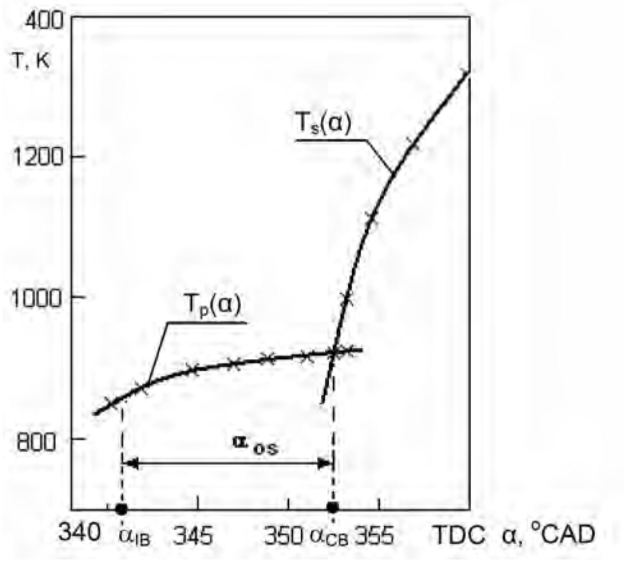


Fig.5. Graphic presentation of the method for the determination of ignition delay angle (2): α_{pw} – fuel injection beginning, α_{ps} – combustion beginning

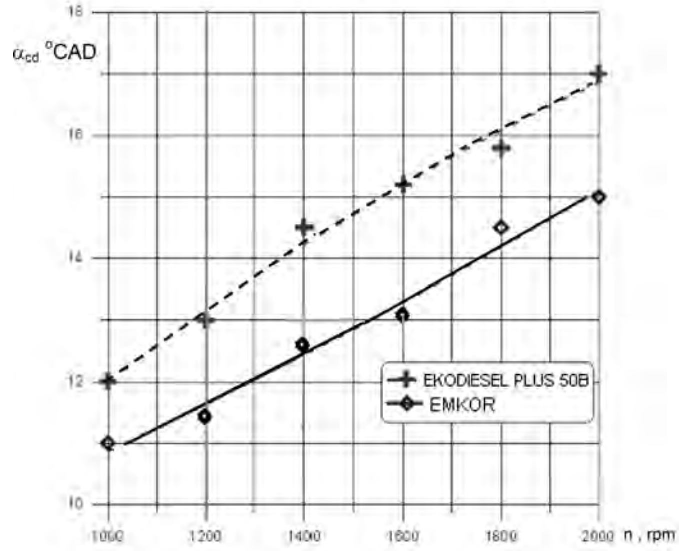
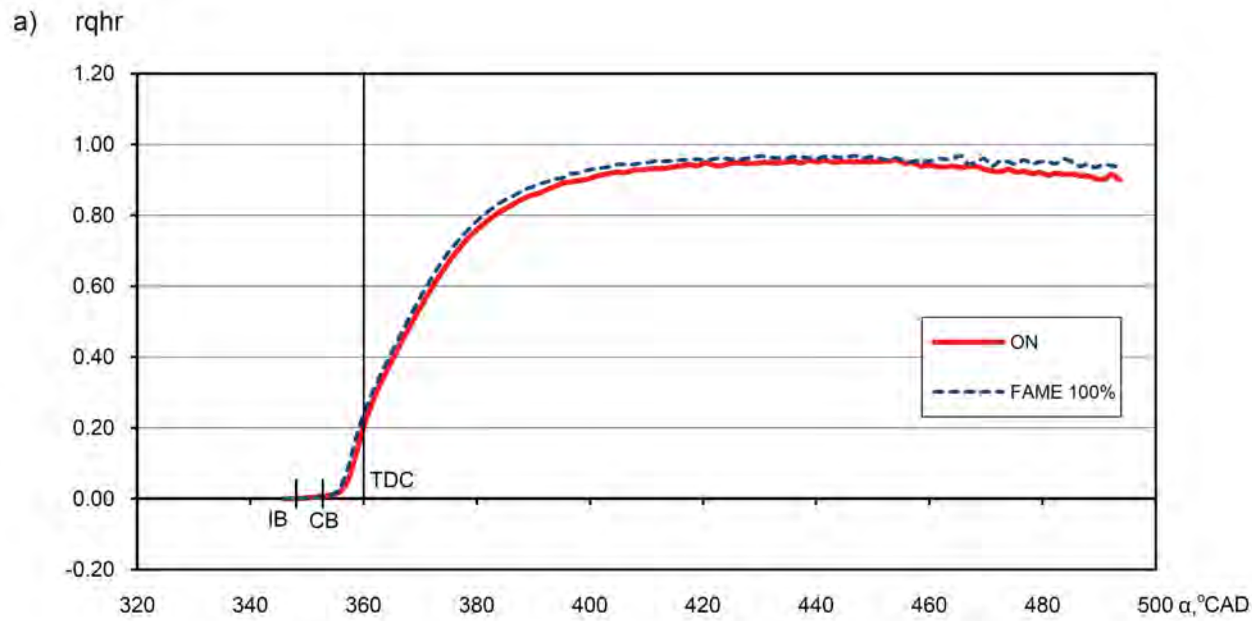


Fig. 6. Values of self-ignition delay angle in AD3.152 UR engine operating in accordance with the external speed performances, fuelled with two different fuels for the injection advance angle α_{ww} = 17 °CA

6. Conclusions

Experimental results and their analysis made so far indicate two-phase character of the combustion process. It is characterised by the occurrence of the two maximum heat release rates. The first one is found in the kinetic phase of the combustion of the mixture produced from a part of the fuel fed during self-ignition delay. The other maximum rate occurs when the combustion intensity is conditioned by the intensity of fuel vapours and oxidant mutual dissociation.

The rate of heat release during biofuel combustion reached higher values when compared with diesel oil combustion. The values of self-ignition delay angle were higher for the engine fuelled by diesel oil than biofuel. Computed droplet Sauter Mean Diameters d_{32} and critical droplet diameters d_{cr} of biofuels were smaller than those of droplets formed by diesel oils. Greater droplet critical diameters in diesel oil result probably from a little lower value of their surface tension. The determined initial values of velocities of the injected fuel jet, droplets disintegration time and fuel jet penetrations were comparable for both fuel types.



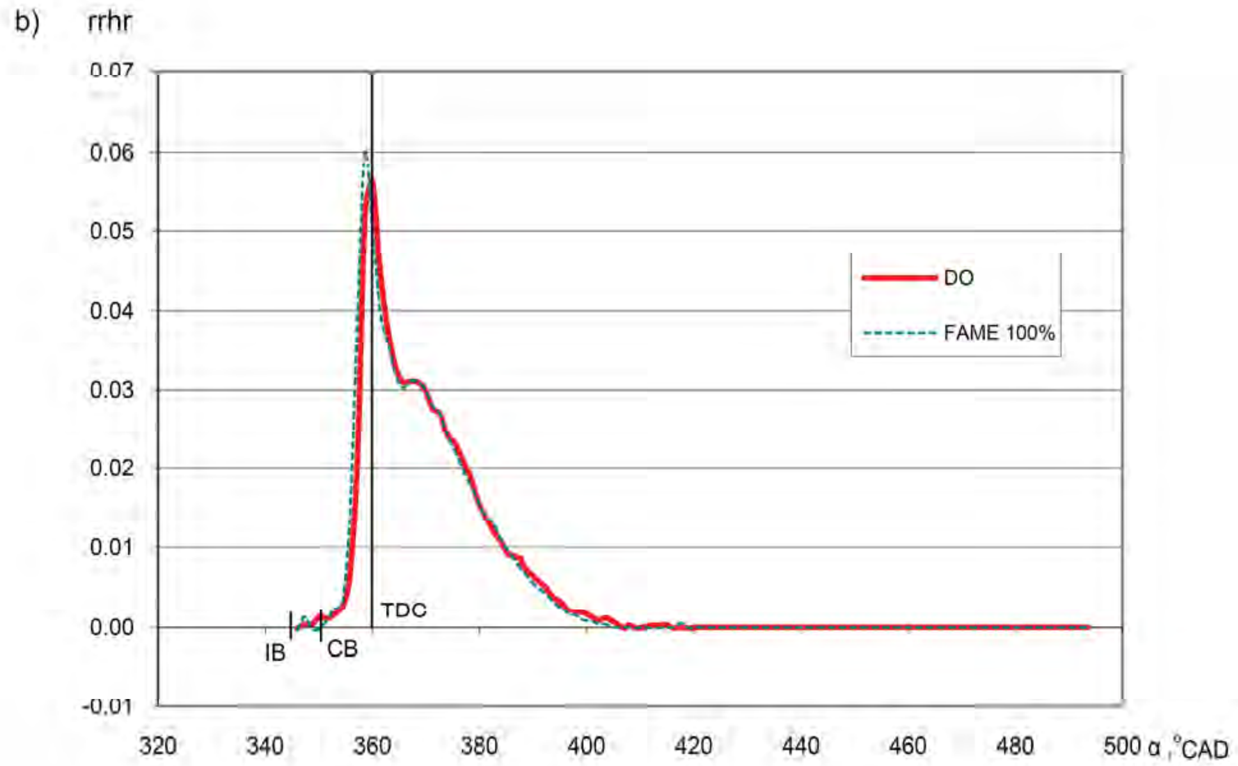
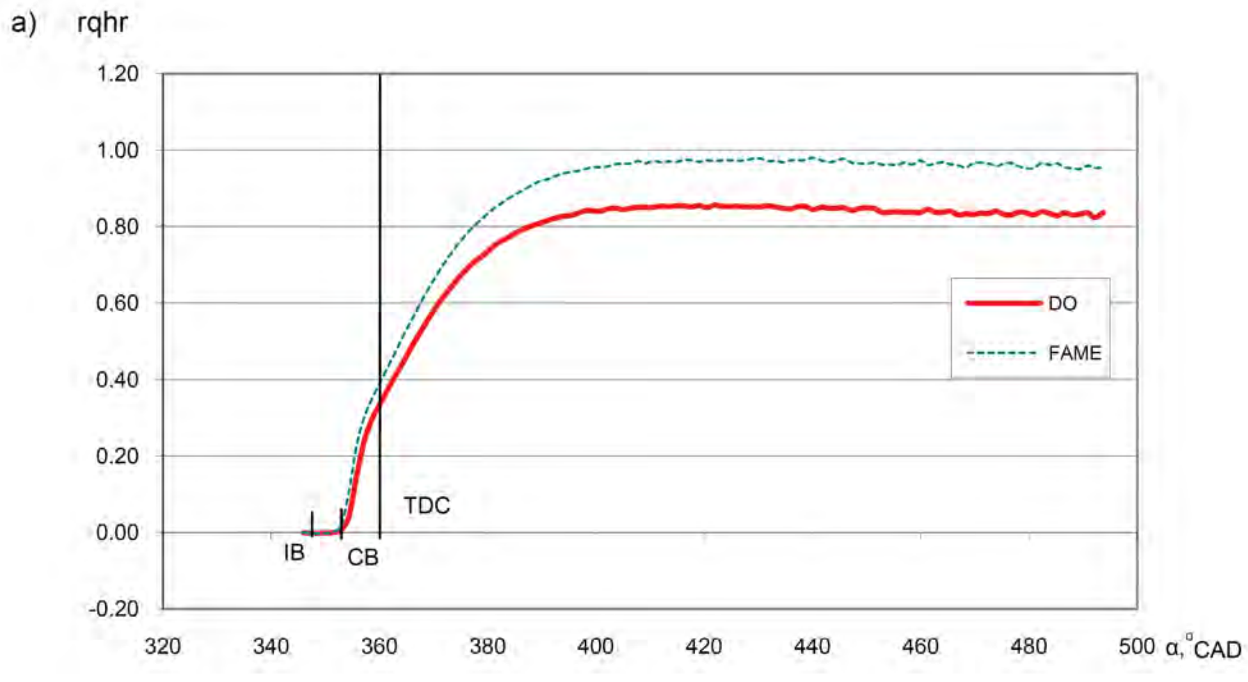


Fig. 7. Analyses of the relative quantity (rqhr) and rate (rrhr) of the heat release during combustion in AD3.152 UR engine operating at n = 2000 rpm and N_e = 32 kW, α_{aa} = 17 °CA

Although only a few issues that concern the fuelling of piston internal combustion engines by diesel oils and biofuels were presented, moreover, in a rather selective manner, they show the necessity of conducting further research. Thorough investigations should focus on complex physical and chemical processes of volatile mixture formation and combustion in CI engines. Measured pressure data that are ensemble averaged over a large number of single cycles are required for an accurate heat release analysis.



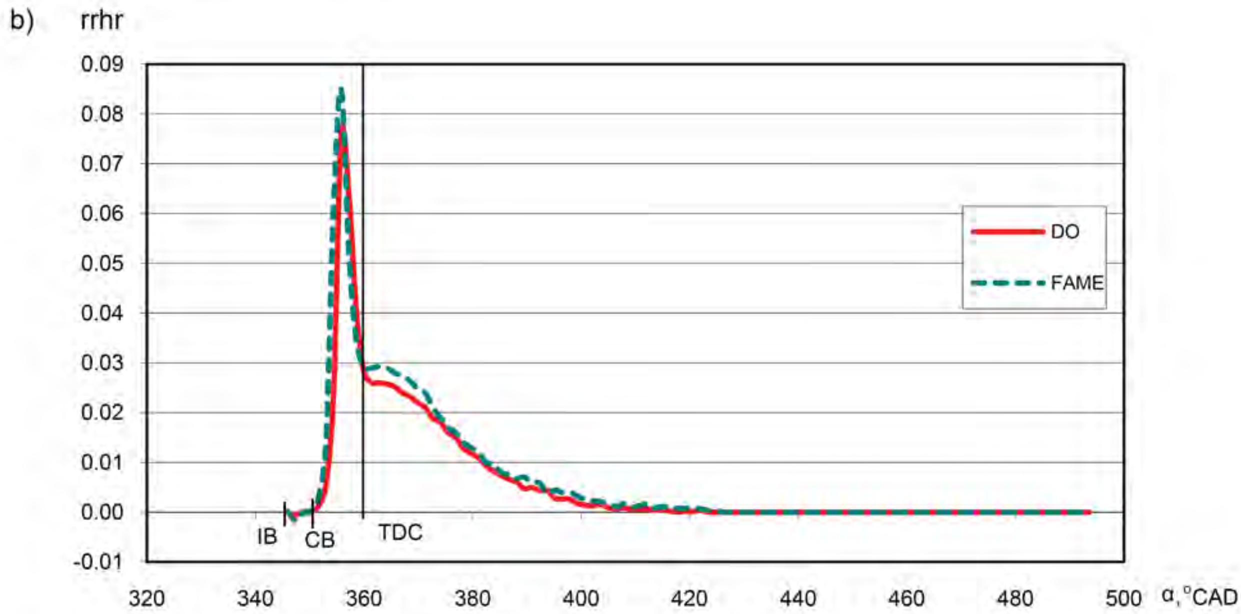


Fig. 8. Analyses of the relative quantity (rqhr) and rate (rrhr) of the heat release during combustion in AD3.152 UR engine operating at n = 1400 rpm and N_e = 23 kW, α_{ww} = 17 °CA

Good agreement in combustion chamber pressure profiles and performance results over the range of speeds and loads examined indicates that the combination of heat release rate and engine simulation models provide an effective means of predicting engine performance and loss mechanisms. The heat release analysis and engine simulation model provide fast and simple powerful and practical approaches to assist in understanding the charge combustion process. In addition, by simulating changes in engine design and operating conditions the engine simulation model can accurately predict the performance of a diesel engine.

The engine simulation model predicts a significant improvement in the performance of the engine performance with faster combustion and reduced leakage. The need for faster combustion, reduced leakage may be quite apparent from the results.

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