COMBUSTION CHARACTERISTICS OF COMPRESSION IGNITION ENGINE FUELLED WITH RME AND ETHANOL

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Abstract

At the previous conference KONES'2004 emission characteristics of C.I. engine fuelled with either rape oil methyl ester (RME) or diesel fuel (DF) as a main fuel and ethanol as additional fuel were presented. Comparison of emission and brake fuel conversion efficiency (BFCF) of the engine fuelled with both main fuels showed that fuelling with RME and ethanol is more advantageous on account of both emission and efficiency than with DF and ethanol. In this paper combustion characteristics of the engine fuelled with RME and ethanol are presented and analysed. They enabled the insight into combustion processes of RME and ethanol resulting in better understanding influence of ethanol fraction in total fuel on emission and efficiency.

1. Introduction

Major research aspects of engine development are application of renewable alternative fuels and continuous concern of environmental protection. European directive on promotion of renewable fuels proposes the share of biofuels in the market in the year 2005 at the level of 2 percent and in 2010 – 5,75 percent [1]. Great prospect is concerned with application of oxygenates as components to the fuels, including alcohols. European Commission proposes application of 15 percent of ethanol to diesel fuel [2]. Scania tested 5% ethanol fraction in the blend with diesel fuel in demonstration bus [3]. Study on application of ethanol-diesel blend on performance and emission was carried out by Yu et al. [4] and Lü [5]. Also methanol-diesel blends were tested by Huang et al. [6, 7]. In this work two renewable fuels were used to operate D.I. C.I. engine not as blend, but separately. While RME was injected directly into engine cylinder by the standard fuel injection system, ethanol was injected into inlet port during suction stroke and entered into the cylinder as a vapour. This manner of ethanol application differs significantly with the former one, in which ethanol was applied as a blend with a base fuel and evaporated in the cylinder with delay.

The main objective of this project was to prove the hypothesis, that burning of ethanol-air mixture accelerates combustion changing its character from diffusional into kinetic one, resulting in shorter total combustion period. This process should result in improving combustion: to increase efficiency and decrease emissions.

Formerly ethanol injection was applied to the same engine operating on diesel fuel with a good result – decrease of CO_2 and smoke level [8]. Comparison of emission of the engine operating either on diesel fuel or on RME with and without additional ethanol injection was carried out in [9, 10]. In this paper investigation on combustion processes in C.I. engine fuelled with RME and ethanol has been carried out. During experiments described in the paper presented at the conference KONES'2004 [11], measurements of cylinder pressure, fuel pressure before the injector and injector needle lift vs. C.A. were also carried out and used to determine characteristic parameters of combustion.

Main objective of the work was to investigate the influence of ethanol injection on combustion processes and combustion characteristics.

2. Engine test stand

Experiments were carried out at a test stand shown in Fig. 1. Main part of it is single-cylinder 1HC102 direct injection, naturally aspirated diesel engine. Engine data are shown in Table 1.

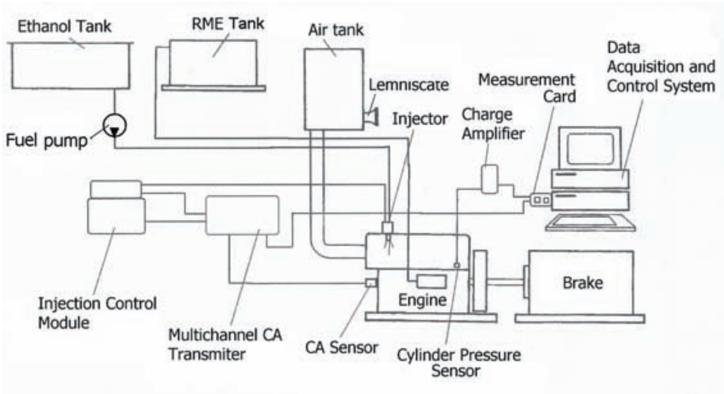

Fig. 1. Engine test stand

Engine torque was measured by means of eddy-current dynamometer Vibrometer 3WB15. RME fuel consumption was measured with the use of automatic dose meter PG-80. Ethanol dose per cycle was measured with the use of volumetric method. Airflow was measured with the use of flowmeter installed on the air tank which reduced pressure pulsation.

Table 1. Engine data

Type of the engine	1HC102 (Polish production)
No of cylinders	1
Swept volume	980 cm^3
Compression ratio	17
Bore/Stroke	102/120 mm
Max power	11 kW at 2200 rpm
Max torque	55 Nm at 1500 rpm
Injection pump*)	plunger type
Injector nozzle	pintile type
Orifice diameter	0.95 mm
Injection pressure	13.2-14.2 MPa
Injection pressure of ethanol	0.30 MPa

*Standard fuel injection pomp was replaced by another one, giving higher fuel delivery

Pressure in the cylinder was measured with AVL transducer 8QP505 inserted in the cylinder head. Injector needle lift was measured with inductive sensor CL80 of Polish production. Fuel pressure before the injector was measured with AVL sensor QL21D. The high speed measurements were synchronised with crank angle measured with Introl sensor. All measured quantities were transmitted to the high speed measurement system developed in the Department of Internal Combustion Engine and Automobiles [12]. For exhaust gas analysis, especially CO, CO_2, HC and air excess ratio λ, an AVL 465 DiGas analyser was used. NO_x emission was measured by Beckman analyser Model 951. Also HC was measured with Beckman analyser Model 402.

Properties of both fuels used are given in Table 2.

Measured values of pressure in the cylinder, fuel pressure before injector and needle lift in function of crank angle were input data to the computational programmes, with which diagrams of the rate of heat release, fraction of fuel burnt, ignition delay, time of combustion and total time of combustion were computed.

Table 2. Physic-chemical properties of RME and ethanol

Property	RME	Ethanol
Chemical formula	–	C_2H_5OH
Molecular weight	~ 300	46
Density @ 20ºC, kg/m^3	878	789
Calorific value, MJ/kg	38.5	26.8
Calorific value of stoichiometric mixture, MJ/m^3		3.85
Heat of evaporation, kJ/kg	250	840
Temperature of self-ignition, K	~400	665
Stoichiometric air/fuel ratio, kg air/kg fuel	13.6	9.06
Lower flammability λ_l	–	2.06
Higher flammability λ_h	–	0.30
Kinematic viscisity, mm^2/s @ 40ºC	4.58	1.4
Surface tension N/mm^2	$31.5 \cdot 10^{-3}$	$23.61 \cdot 10^{-3}$
Cetane number	60	8
Flame temperature, K	–	2235
Molecular composition (by mass)		
C	0.775	0.522
H	0.121	0.130
O	0.104	0.348

Ethanol contained water 8% (by vol.)

3. Course of investigation

Investigation was carried out for engine operating conditions shown in Table 3. Measurement point were chosen in such a way, that the comparison of engine parameters and emission could be obtained for the same load but for different proportion of ethanol to both fuels (RME and ethanol).

Table 3. Conditions of investigation

Engine speed n, rpm	1200	1800	2200
Load T, Nm	20 40	20 40	20 40
Angle of beginning of injection of RME, deg BTDC	25*, 30, 35*		
Angle of beginning of injection of ethanol to inlet port, deg ATDC (during inlet stroke)	60		
Ethanol energy to total fuel energy Ω_E	From 0% to about 50%		

*) Only for emission and bfce measurements

4. Results and Discussion

4.1. Influence of ethanol fraction on emission and efficiency

Influence of ethanol fraction on emissions and efficiency depends on the engine load. For low load increase of ethanol fraction has a slight influence on CO_2 and smoke emission, but increases CO and HC and decreases NO_x emission. Brake fuel conversion efficiency (BFCE) shows slow decrease with increase of ethanol fraction. All these phenomena are the result of cooling effect of ethanol evaporation. Influence of ethanol fraction on charge temperature is shown in Fig. 2.

However, for high load the influence of ethanol fraction is different. In case of high speed BFCE increases for all injection timings from very low value for neat RME, Fig. 3, resulting in:

- increase of NO_x, Fig. 4,
- decrease of CO_2, Fig. 5,
- decrease of smoke level, Fig. 6,

– decrease of CO, Fig. 7,
– very low HC emission, showing a slight tendency to decrease, Fig. 8.

In the range of ethanol fraction in which the above data were measured, air excess ratio was contained within 1,2÷1,4, Fig. 9.

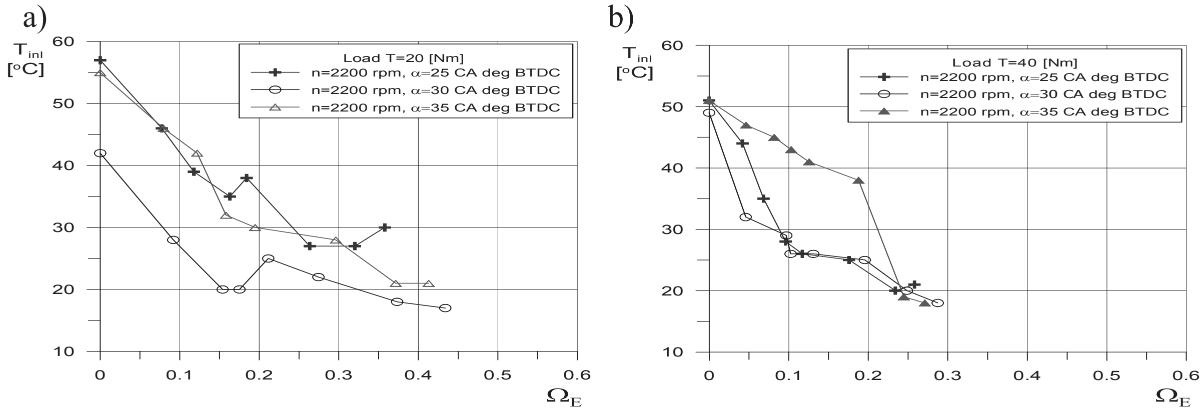

Fig. 2. Temperature of inlet charge in function of the ethanol energy fraction in total fuel at low load (a) and at high load (b)

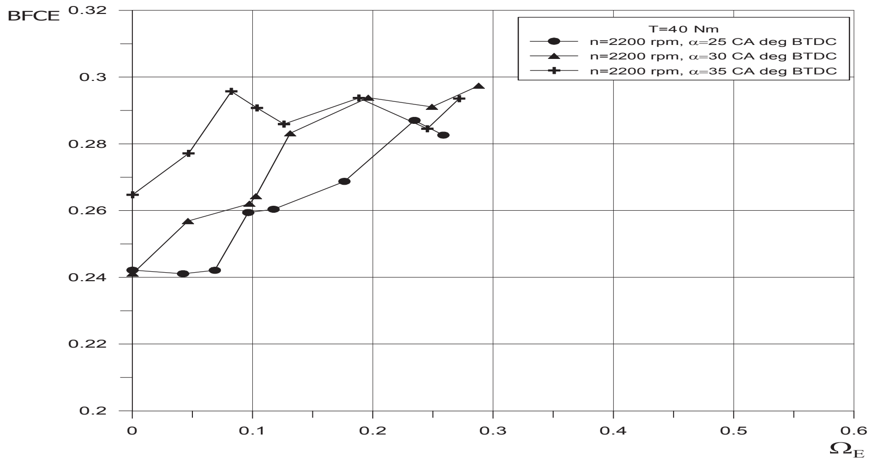

Fig. 3. Brake fuel conversion efficiency in function of ethanol energy fraction in total fuel for high load

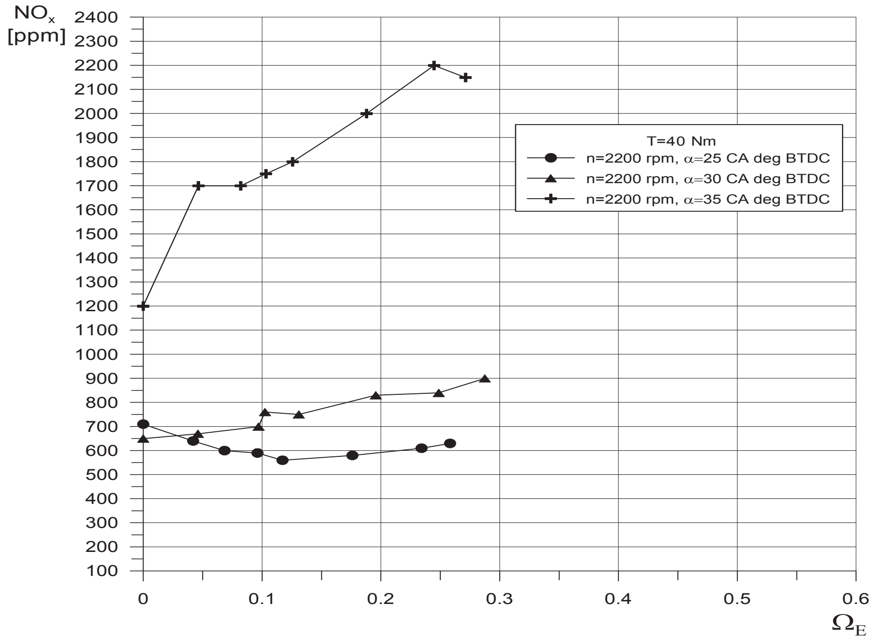

Fig. 4. Nitric oxide emission in function of ethanol energy fraction in total fuel for high load

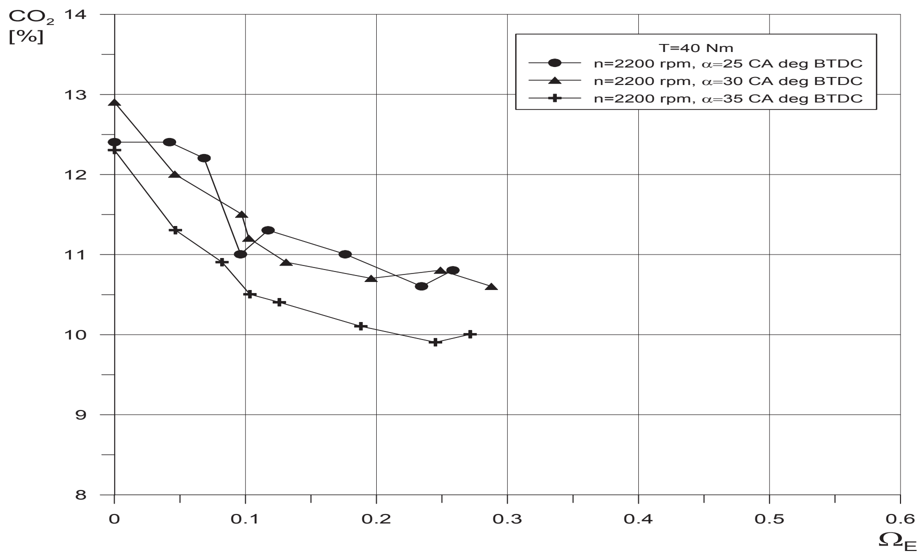

Fig. 5. Carbon dioxide emission in function of ethanol energy fraction in total fuel for high load

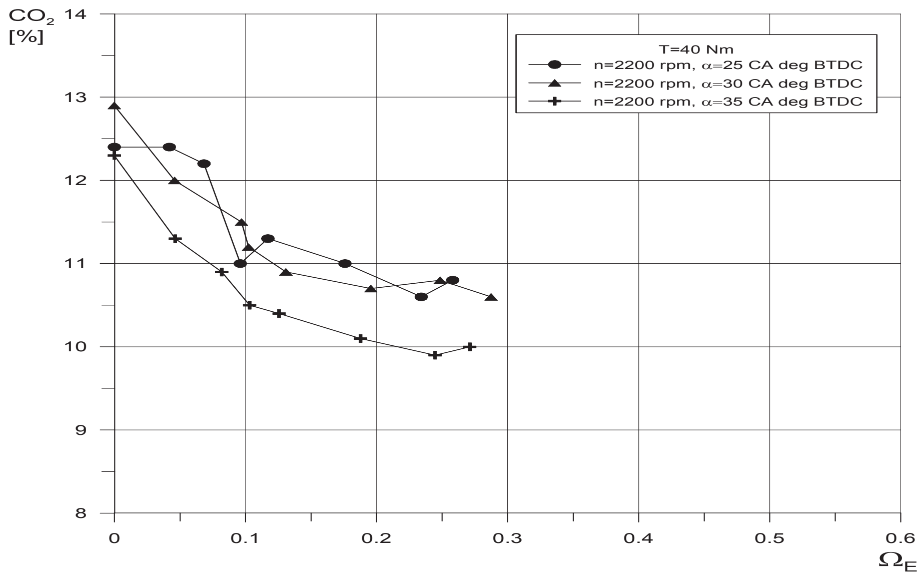

Fig. 6. Smoke emission in function of ethanol energy fraction in total fuel for high load

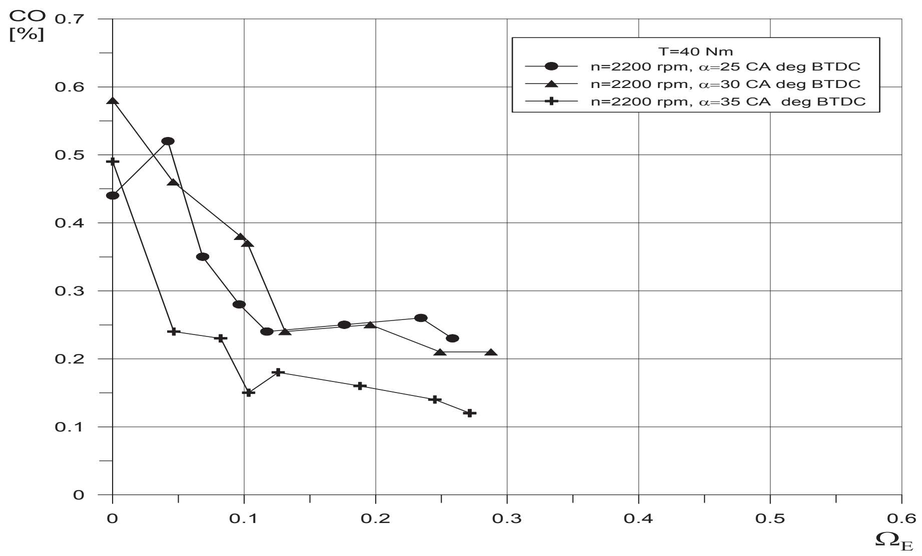

Fig. 7. Carbon monoxide emission in function of ethanol energy fraction in total fuel for high load

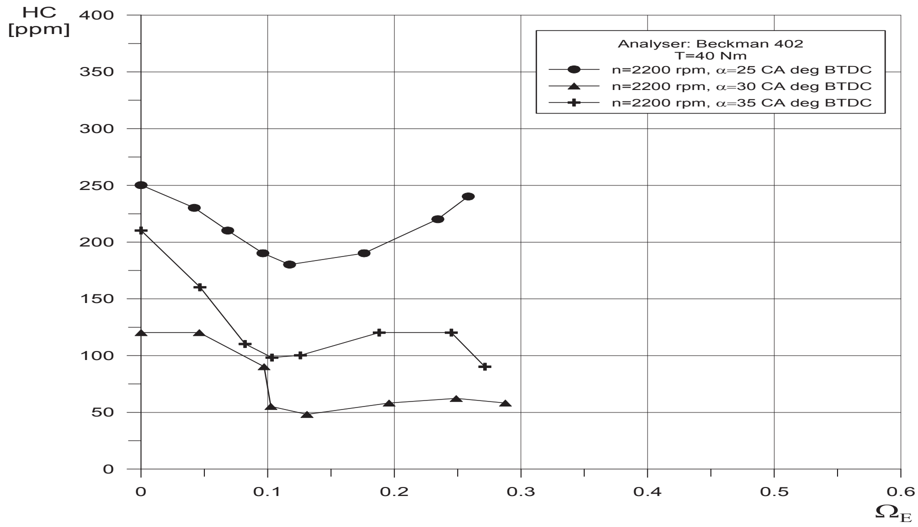

Fig. 8. Hydrocarbon emission in function of ethanol energy fraction in total fuel for high load

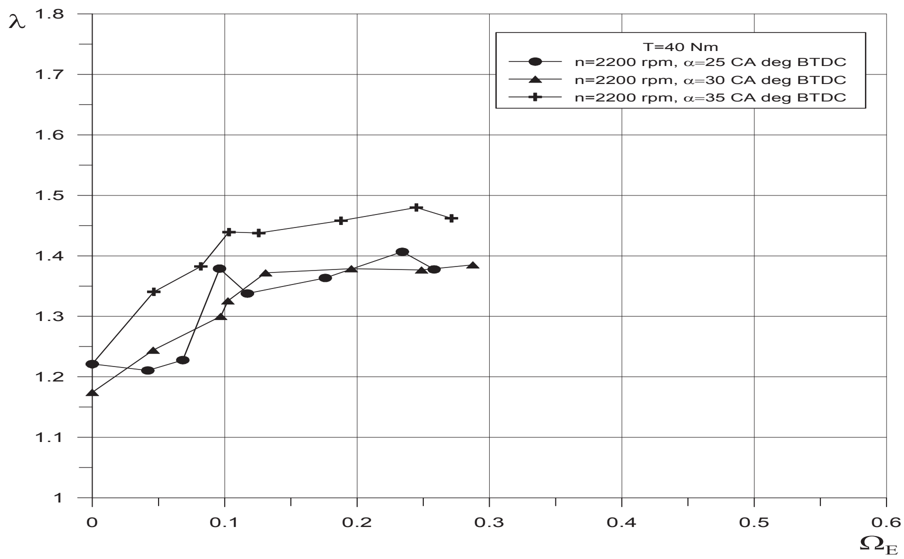

Fig. 9. Coefficient of air excess in function of ethanol energy fraction in total fuel for high load

Explanation of these results is as follows. BFCE and NO_x emission increases due the high temperature (the higher the pressure, Fig. 11, the higher the temperature in the cylinder). CO_2 decreases as a result of BFCE increase with ethanol fraction (less fuel is burnt). Moreover, products of ethanol combustion consist less CO_2 and more H_2O. CO and HC emission decreases as a result of better (more efficient) combustion.

4.2. Influence of ethanol fraction on combustion characteristics

As far as pressure diagrams are concerned, for low load (when cooling effect of ethanol evaporation is significant) the higher the ethanol fraction, the lower the pressure, Fig. 10. For high load – vice versa: the higher ethanol fraction, the higher the pressure, Fig. 11. Computation of heat release rate and fraction of fuel burnt vs. ethanol fraction were carried out for the same operating conditions as the pressure diagrams. Together with pressure-time history they constitute fundamental information on combustion. From these and other diagrams (for 1200 and 1800 rpm) the following conclusion may be drawn.

For low load:

- the higher the ethanol fraction, the lower the maximum heat release rate and its maximum is more delayed, Fig. 12,

– the character of burning is kinetic (burning of fuel vapours)

For high load:

– the higher the ethanol fraction, the higher the maximum of heat release rate, Fig. 13,
– diffusional character of burning changes into kinetic one with increase of ethanol fraction,
– with increase of ethanol fraction, the fraction of fuel burnt increases slowly at the beginning of combustion and very quickly at the end for all cases of engine operation, Fig. 14 and 15.

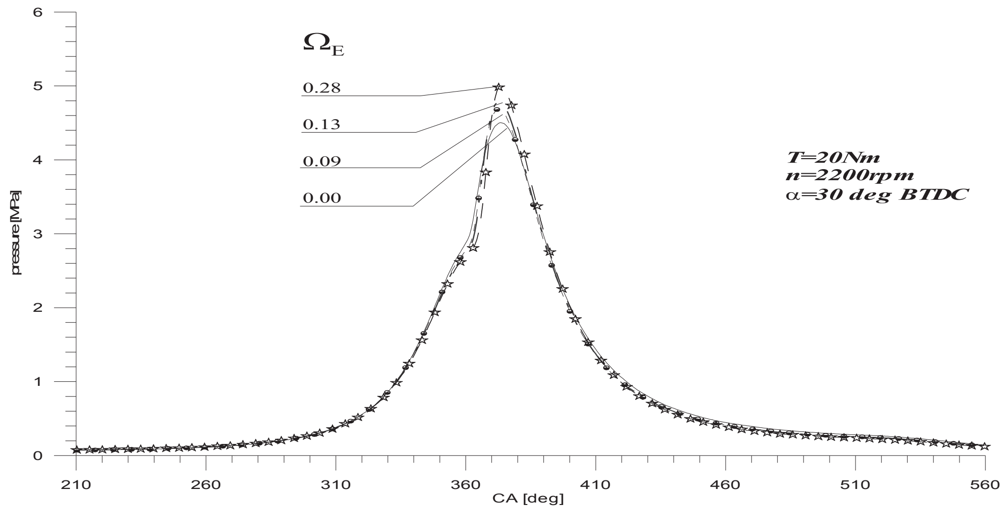

Fig. 10. Pressure vs. CA at low load, speed 2200 rpm and injection timing of RME = 30 CA deg BTDC for different ethanol fractions Ω_E.

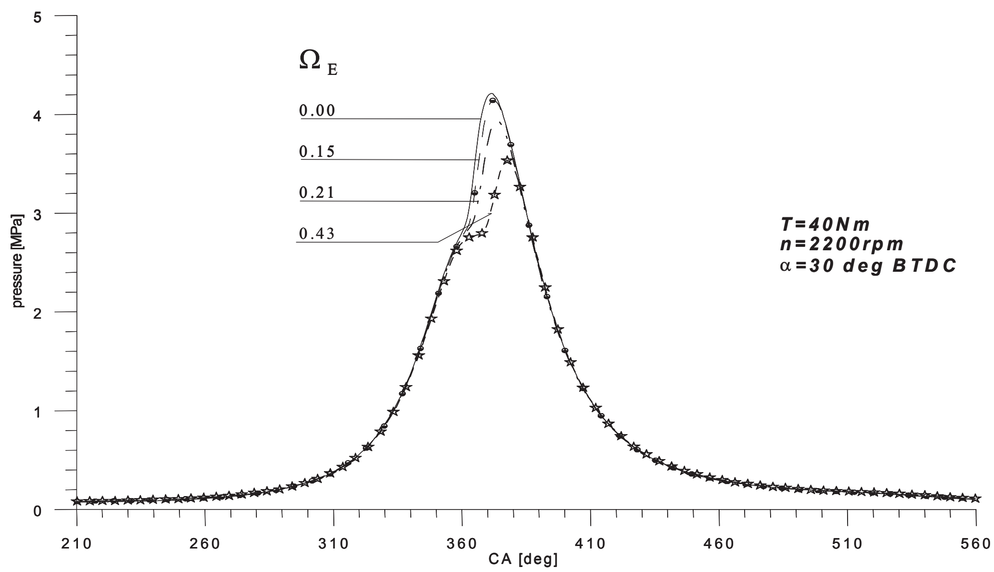

Fig. 11. Pressure vs. CA at high load, speed 2200 rpm and injection timing of RME = 30 CA deg BTDC for different ethanol fractions Ω_E.

Summing up, under high load, when cooling effect of ethanol evaporation is low in comparison with the effect of high temperature of the cycle corresponding to big amount of heat evolved, the increasing fraction of ethanol results in higher pressure (and temperature), better efficiency, lower emissions, except NO_x, especially CO_2 and smoke and the change of character of combustion from diffusional (for neat RME) into kinetic one.

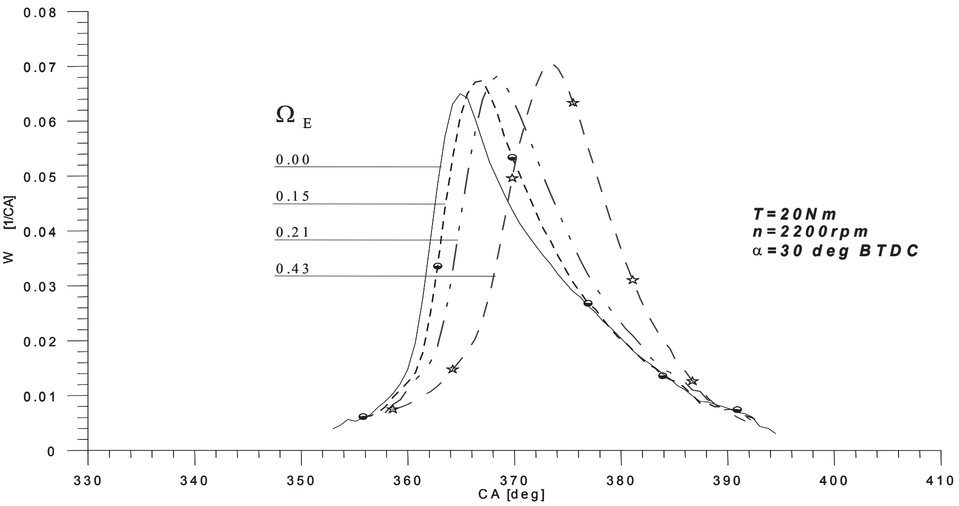

Fig. 12. Relative heat release rate vs. crank angle at low load and speed 2200 rpm for several ethanol energy fraction in total fuel Ω_E

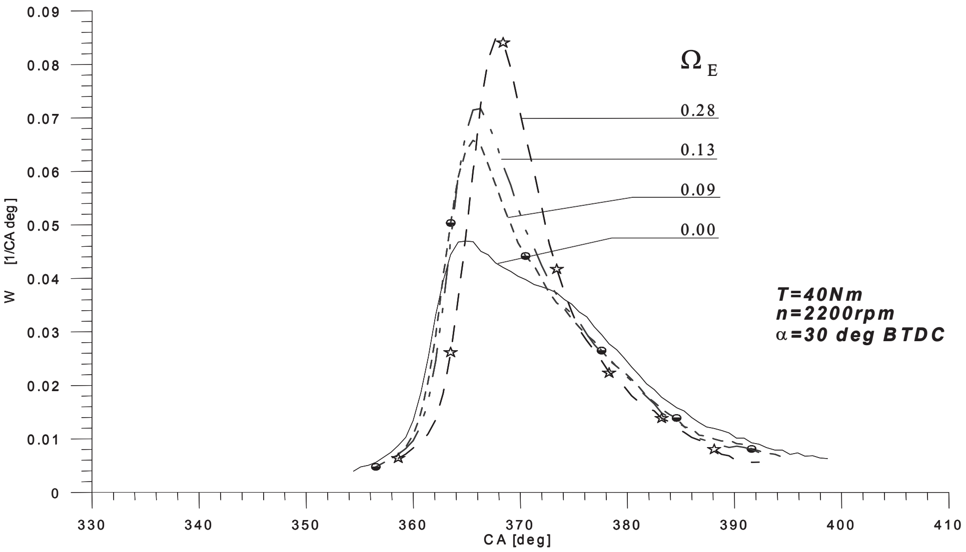

Fig. 13. Relative heat release rate vs. crank angle at high load and speed 2200 rpm for several ethanol energy fraction in total fuel Ω_E

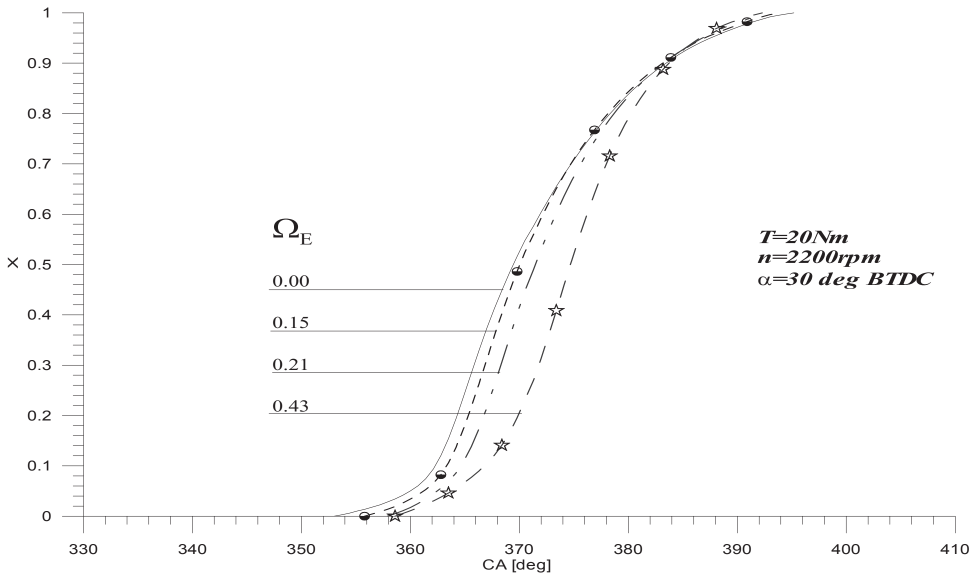

Fig. 14. Fraction of fuel burnt crank angle at low load and speed 2200 rpm for several ethanol energy fraction in total fuel Ω_E

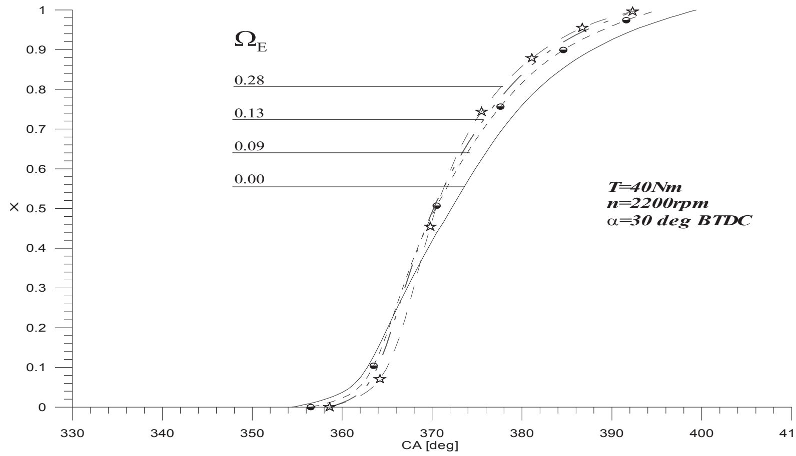

Fig. 15. Fraction of fuel burnt crank angle at high load and speed 2200 rpm for several ethanol energy fraction in total fuel Ω_E

4.3. Ignition delay and combustion time

Measured data of the beginning of needle lift, the point of the start of combustion (heat release rate becomes to increase from zero) and the point of the end of combustion (the fraction of fuel burnt x = 0,98) were used to compute ignition delay and time of combustion. Total time of combustion is the sum of ignition delay and combustion time, and is very important for diagnostics of combustion processes. All these characteristic times (in deg of CA) for injection timing 30 CA deg, three speeds and two loads vs. ethanol fraction are shown in Table 4.

Table 4. Ignition delay, combustion time and total time of combustion vs. speed, load and ethanol fraction Ω_E

n [rpm]	M [Nm]	Ω_E	τ_z [CA deg]	τ_s [CA deg]	$\tau_z+\tau_s$ [CA deg]
1200	20	0.0	12.4	41.0	53.4
	20	0.149	10.7	40.0	50.7
	20	0.262	9.7	38.0	47.7
	20	0.535	11.7	34.0	45.7
	40	0.0	10.0	42.0	52.0
	40	0.086	9.3	40.0	49.3
	40	0.161	11.0	41.0	52.0
	40	0.348	11.0	37.0	48.0
1800	20	0.0	9.9	36.0	45.9
	20	0.140	10.2	34.0	44.2
	20	0.252	11.2	33.0	44.2
	20	0.478	11.2	33.0	44.2
	40	0.0	8.9	45.0	53.9
	40	0.088	9.9	42.0	51.9
	40	0.150	8.9	38.0	47.8
	40	0.310	9.9	36.0	45.9
2200	20	0.0	7.8	42.0	49.8
	20	0.154	10.8	38.0	48.8
	20	0.211	12.8	35.0	47.8
	20	0.434	12.4	34.0	46.4
	40	0.0	9.8	45.0	54.8
	40	0.097	11.8	38.0	49.8
	40	0.131	13.8	34.0	47.8
	40	0.288	12.4	34.0	46.4

Time of combustion and total time of combustion are presented in function of ethanol fraction Ω_E in Fig. 16 to18.

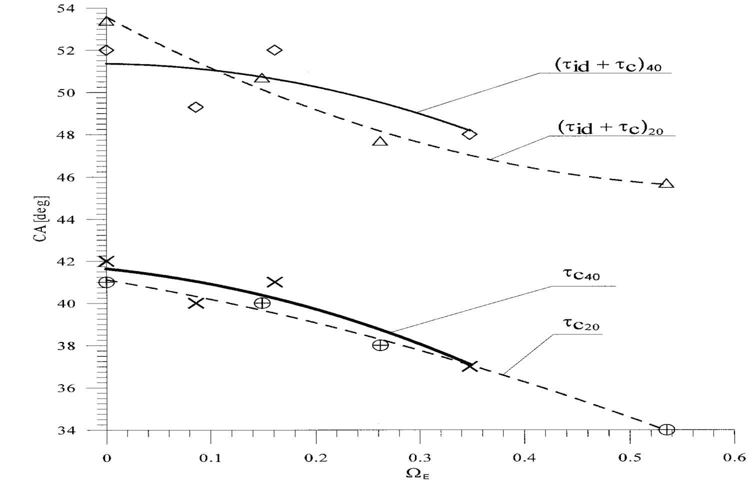

Fig. 16. Combustion time τ_c [CA deg] and total combustion time τ_{id} + τ_c [CA deg] in function of ethanol energy Ω_E at 1200 rpm and two loads

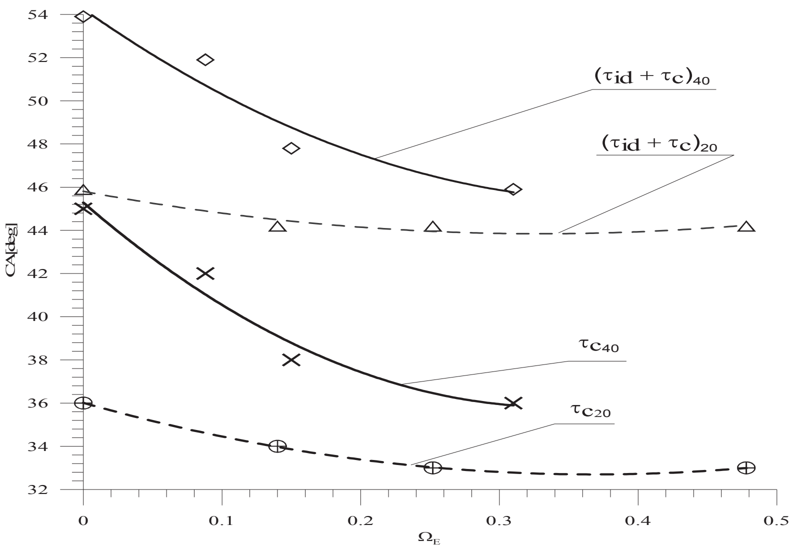

Fig. 17. Combustion time τ_c [CA deg] and total combustion time τ_{id} + τ_c [CA deg] in function of ethanol energy Ω_E at 1800 rpm and two loads

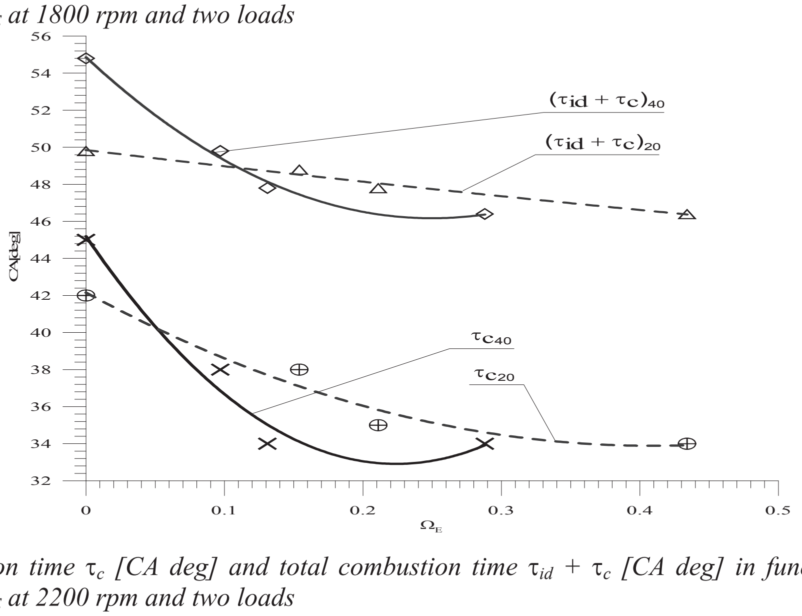

Fig. 18. Combustion time τ_c [CA deg] and total combustion time τ_{id} + τ_c [CA deg] in function of ethanol energy Ω_E at 2200 rpm and two loads

From these figures (and also from diagrams of heat release rate and fraction of fuel burnt) the following conclusions may be drawn.
For all speeds and loads:
- in spite of longer ignition delay, total combustion time is shorter with increasing ethanol fraction, Ω_E,
- combustion time decreases with increase of ethanol fraction, Ω_E.

5. Conclusions

As far as combustion processes in D.I. C.I. engine are concerned, ethanol injection into inlet port during suction stroke accelerates combustion changing diffusional character of burning into kinetic one. Although the ignition lag increases with ethanol fraction, combustion time decreases, resulting in much shorter total combustion time. Additional injection of ethanol into the inlet port results in lower emission of smoke and carbon dioxide and, at low load, also nitric oxide. Ethanol energy fraction in total fuel energy Ω_E may reach 50% at low load and 30% at high load and is limited by occurrence of diesel knock.

Acknowledgement
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List of notation

n – engine speed, rpm,
T – torque, Nm,
W – rate of heat release, 1/deg CA,
X – fraction of fuel burnt,
α – CA of the beginning of injection of RME, deg BTDC,
τ_c – combustion period, CA deg,
τ_{id} – ignition delay, CA deg,
$\tau_{id} + \tau_c$ – total combustion period, CA deg,
Ω_E – the ratio of ethanol energy to total fuel energy (ethanol energy fraction).

Abbreviations

BTDC – before top dead centre,
CA – crank angle,
CI – compression ignition,
DI – direct ignition,
RME – rape oil methyl ester.