Scientific Journals
Maritime University of Szczecin

Zeszyty Naukowe
Akademia Morska w Szczecinie

2013, 36(108) z. 2 pp. 102–109
ISSN 1733-8670

2013, 36(108) z. 2 s. 102–109

Combustion and performance parameters of a Diesel engine operating on ethanol-Diesel fuel blends

Gvidonas Labeckas[1], Stasys Slavinskas[1], Tomasz Lus[2], Oleh Klyus[3]
Marius Mažeika[1]

[1] Aleksandras Stulginskis University
LT-53361 Kaunas-Academy, Students street, 11, Lithuania
e-mail: {gvidonas.labeckas;stasys.slavinskas}@.lzuu.lt; marius.mazeika@yahoo.com
[2] Polish Naval Academy, Mechanical-Electrical Faculty
81-103 Gdynia, ul. Śmidowicza 69, e-mail: t.lus@amw.gdynia.pl
[3] Maritime University of Szczecin, Faculty of Marine Engineering
70-500 Szczecin, ul. Wały Chrobrego 1–2, e-mail: olegklus@o2.pl

Key words: Diesel engine, ethanol, Diesel fuel, autoignition delay, combustion, NOx, CO, HC emissions, smoke opacity of the exhaust

Abstract
The article presents bench test results of a DI (60 kW) Diesel engine D-243 operating on class 2 Diesel fuel (DF) as baseline fuel and its 5vol%, 10vol% and 15vol% blends with anhydrous ethanol. The purpose of the research was to investigate the effect of the ethanol addition to Diesel fuel on the autoignition delay, combustion, engine performance efficiency and emissions of the exhaust. The results of engine operation on ethanol-Diesel blends are compared with baseline parameters of normal Diesel running at full (100%) load and rated 2200 rpm speed.

Introduction

The number of Diesel engines used in a heavy-duty haulage trucks, automobiles, city buses and passenger light-duty cars has greatly been increased during the last two decades. For this reason increases Diesel fuel demand that can be attributed to growing number of Diesel engines as more efficient, consuming less fuel per unit of power developed and suggesting less damage to the environment. To produce necessary amount of Diesel fuel technically is feasible but every next barrel of crude oil is getting farther, deeper and harder with a higher extraction, production, and delivery price. Limited crude oil reserves, high market prices of mineral fuels, ambient air pollution and global warming are matter of urgent concern that encourage researchers to intensify investigations on new environment friendly alternative and renewable energy sources suitable for Diesel engine powering. Using of renewable fuels in agricultural, transport, maritime and military sectors can be recognised as only rational way leading to production of less the CO_2 emitted into atmosphere in a global cycle and mitigation of climate changes.

Alcohol-based fuels have been important energy sources since the 1800s. As early as 1894, France and Germany were using ethanol in internal combustion engines [1]. The first investigations on the use of ethanol in Diesel engines were carried out in South Africa in the 1970s and continued in Germany and the USA during the 1980s [2]. As potential mineral fuel extender bioethanol is indigenous and locally available, environment friendly and renewable, sustainable and reliable, safe to store and easy to handle, non-polluting and sulphur-free material, and is one of the cleaner-burning alternatives to mineral fuels. To solve technical problems, several methods can be adapted to employ a certain amount of ethanol for Diesel engine powering, which are known as alcohol fumigation [3], application of a dual injection systems [4], preparation of the alcohol-Diesel fuel micro-emulsions [5] and using of the alcohol-Diesel fuel blends [3, 6, 7 8].

Investigations conducted on a single cylinder DI, variable compression ratio Diesel engine [3] showed that biofuel blends prepared by mixing of anhydrous (200 proof) ethanol and Diesel fuel would be acceptable for the Diesel engine fuelling when applied in proper proportions. At a higher than 15vol% blending ratio the lubrication problems of plunger-barrel and needle-valve-body units may arise especially during long-term operation on ethanol-Diesel blends. More advantages and disadvantages of the ethanol as potential Diesel fuel additive have been elucidated in reports [9, 10]. The molecular weight of ethanol is lower 3.91 times, density is 4.9% lower at temperature of 20°C and its kinematic viscosity is also 1.47 times lower at temperature of 40°C compared to the normal Diesel fuel. Noted changes in density and viscosity along with extremely deep CFPP at the temperature below –38°C may elevate the fuel flow in the fuelling system and elevate the engine starting in severe winter conditions.

The purpose of the research was to perform analyses of the effects of anhydrous (200 proof) ethanol addition to arctic (class 2) Diesel fuel on biofuel properties and the autoignition delay, combustion, engine performance efficiency and emissions of the exhaust. The engine performance on 5vol%, 10vol% and 15vol% ethanol-Diesel blends, including changes in nitrogen oxides NO_x, carbon monoxide CO and dioxide CO_2, total unburned hydrocarbons HC, residual oxygen O_2 content and smoke opacity of the exhaust was compared with corresponding parameters obtained when operating on the normal Diesel fuel over a wide range of loads at rated 2200 rpm speed.

Objects, apparatus and methodology of the research

Tests have been conducted at Aleksandras Stulgiskis University on four stroke, four cylinder, DI (60 kW) Diesel engine D-243 with a splash volume $V_l = 4.75$ dm^3, bore 110 mm, stroke 125 mm and compression ratio $\varepsilon = 16:1$. The fuel was delivered by an in line fuel injection pump thorough five holes injector's nozzles with the fuel delivery starting at 25° BTDC.

Diesel fuel was produced at the manufactory "Orlen Lietuva" and its quality parameters satisfied requirements EN 590:2009+A1. Anhydrous ethanol (200 proof) was brought from the producer Ltd. "Biofuture" and its parameters corresponded to standard EN 15376:2009. The purity of ethanol was determined with the laboratory device Anton Paar density / concentration meter DMA 5000 with the accuracy of ±0.000005 g/cm^3 at the temperature of 20±0.001°C.

The ethanol-Diesel blends were prepared by pouring 5vol% (B5), 10vol% (B10) and 15vol% (B15) of anhydrous (99.81 purity) ethanol to Diesel fuel container and mixing by hand-splash to keep them in homogeneous conditions. The ethanol-Diesel blends B5, B10, B15 distinguished themselves as having the fuel oxygen mass fraction 2.1%, 3.9%, 5.6%, stoichiometric air-to-fuel equivalence ratio 14.18, 13.91, 13.64 kg/kg and net heating value 42.15, 41.35 and 40.52 MJ/kg.

Load characteristics were taken at rated 2200 rpm speed of the engine operating alternately on arctic Diesel fuel (class 2) and ethanol-Diesel blends B5, B10 and B15. The engine torque was increased by 9 load setting points from the lowest level of 26 Nm up to maximum value of 280 Nm, which corresponded approximately to rated load of bmep = 0.68 MPa.

The engine torque was measured with a three phase asynchronous 110 kW stand dynamometer KS-56-4 with a definition rate of ±1 Nm and the rotation speed was determined with the AVL crank angle encoder 365C that guaranteed an accuracy of less than ±0.2% of measured value.

The air mass consumed by the engine was measured with an AVL air mass meter (0–400 kg/h) installed downstream the air cleaning filter before the air tank to reduce pressure pulsations that guaranteed an accuracy of less than ±1% of measured value. The fuel mass consumption was measured by weighting 100 g of fuel on the AVL dynamic fuel balance 733S flex-fuel with an accuracy of ±0.10%. The temperature of Diesel fuel and tested blends was at the level of 25°C. The engine operated at cooling liquid and oil temperatures of 80–85°C, which were measured with thermo-electrical Diesel package MKD-50M.

The AVL IndiModul 622 was introduced as a high speed multi-channel indicating system for the acquisition and processing of fast crank-angle-based cylinder gas pressure signals. Single-cycle and summarized over 100 engine cycles cylinder gas pressures versus the crank angle were recorded with an accuracy of 0.1° crank angle degree (CAD). The positions of the TDC and crank angles were recorded by using the AVL crank angle encoder 365C mounted at the front-end of the crankshaft with an accuracy of ±0.1° CADs.

A piezoelectric uncooled transducer GU24D with the measurements range of 0–280 bar mounted into the head of the first cylinder and connected to the microIFEM piezoelectric amplifier-signal conditioning were used to measure gas pressure for every load-speed setting point with an accuracy of ±0.1 bar within temperature range of 25°C to 200°C.

The autoignition delay was determined as the period in degrees (φ_i) and units of time (τ_i) between the start of injection (SOI) and the start of combustion (SOC) with an accuracy ±0.1° CADs. As the start of injection was taken the point at which the injector's needle-valve lift compiles about 5% of its total 0.28 mm travel. As the start of combustion was taken the point at which the curve of the heat release rate crosses the zero line and changes its value from the minus side to plus one.

The emissions of nitric oxide NO (ppm), nitrogen dioxide NO_2 (ppm), carbon monoxide CO ppm), dioxide CO_2 (vol%), total unburned hydrocarbons HC (ppm) and residual oxygen O_2 (vol%) in the exhaust were measured with a Testo 350 XL flue gas analyzer. The total nitrogen oxide NO_x emissions were calculated as a sum of both NO and NO_2 components with an accuracy of ±5 ppm. The smoke density D (%) of the exhaust was measured with a "Bosch" RTT 110 opacity-meter, the readings of which are provided as Hartridge units (% opacity) in a scale range 0–100% with an accuracy of ±0.1%.

Results and discussions

The laboratory test results proved that the miscibility of anhydrous (200 proof) ethanol up to 15vol% added to the Diesel fuel is excellent so that it matches well with the test results of other researchers [2, 8]. Properties of tested ethanol-Diesel blends are listed in table 1.

Adding of the ethanol to Diesel fuel decreases density and viscosity of biofuel blends because ethanol differs as having low density (790.0 kg/m^3) at temperature of 20°C and critically reduced viscosity (1.40 mm^2/s) at temperature of 40°C compared to Diesel fuel. Furthermore, the ethanol addition to Diesel fuel decreases carbon-to-hydrogen ratio (C/H), stoichiometric air-to-fuel ratio, net heating value, cetane number and lubricating properties of the fuel blend almost proportional to the biofuel bound oxygen mass (wt%) content. Changes in chemical and physical properties of the fuel blends may have effect on the injection and atomisation characteristics, autoignition delay, combustion, engine performance efficiency and related emissions, especially temperature related NO_x production.

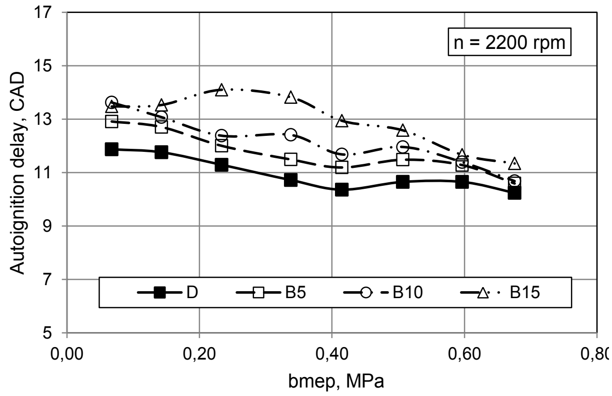

Fig. 1. The autoignition delay as a function of brake mean effective pressure for the normal Diesel fuel and various ethanol-Diesel fuel blends

Lower density, viscosity and higher bulk modulus of compressibility of ethanol-Diesel fuel blends affected speed of high-pressure waves propagating within the injection line. Consequently, the nozzle-needle-valve opening and the start of injection took place later compared to normal Diesel operation. The higher the percentage of ethanol was added to Diesel fuel, the more significant delay in the start of injection was registered in CADs. As figure 1 shows, the autoignition delay period φ_i in CADs for biofuel blends' B5, B10 and B15 also was 7.2%, 15.8% and 28.9% longer than that (10.7°) of normal Diesel running at moderate load of bmep = 0.34 MPa and rated 2200 rpm speed. Differences in the autoignition delays compared to the normal Diesel fuel (10.2°) decreased to certain extend with engine load and compiled 3.4%, 4.3% and 10.7% for re-

Table 1. Properties of tested ethanol-Diesel blends

Property parameters	Test method	DF	B5	B10	B15
Density at 20°C, kg/m^3	EN ISO 12185:1999	827.0	824.8	822.6	820.2
Kinematic viscosity at 40°C, mm^2/s	EN ISO 3104+AC:2000	2.068	1.907	1.840	1.802
Lubricity, corrected wsd, 1.4 μm at 60°C	EN ISO 12156-1:2007	379	417	400	387
Cetane number	EN ISO 5165:1999	51.5	49.9	46.7	44.4
Oxygen mass content, max wt%	–	0.04	2.1	3.9	5.6
Carbon/hydrogen ratio (C/H)	–	6.90	6.77	6.63	6.49
Net heating value, MJ/kg	–	42.95	42.15	41.35	40.52
Stoichiometric air/fuel ratio, kg/kg	–	14.45	14.18	13.91	13.64

spective ethanol-Diesel blends at full (100%) load of bmep = 0.68 MPa. The longer autoignition delay can reasonably be attributed to low cetane number of the ethanol, 1.5 times higher autoignition temperature compared to Diesel fuel (~250°C) and nearly threefold higher latent heat for vaporisation, which vary in-between 840 kJ/kg [11] and 880 kJ/kg [12] and causes significant cooling effect of the fuel sprays.

As a result of late start of injection and long period of autoignition delay the heat release in the engine cylinder started BTDC at approximately –2.7° for Diesel fuel and –2.2°, –2.0° and –1.2° for fuel blends B5, B10 and B15. Changes in the start of combustion and differing chemical and physical properties of ethanol-Diesel fuel blends affected the heat release intensity in the engine cylinder. The maximum heat release rate $(HRR)_{max}$ has been increased from 116.4 kJ/m^3·deg (DF) to 139.0 (14.4%), 148.8 (27.8%) and 155.4 kJ/m^3·deg (33.5%) by using respective ethanol-Diesel fuel blends at full (100%) load and rated 2200 rpm speed (Fig. 2).

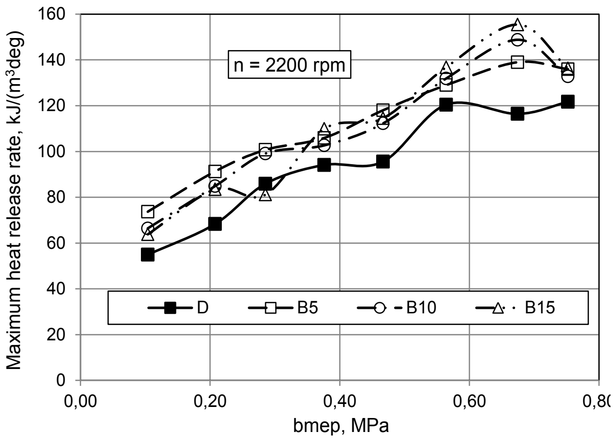

Fig. 2. The maximum heat release rate (HRR) as a function of brake mean effective pressure for the normal Diesel fuel and various ethanol-Diesel fuel blends

The maximum heat release rate representing angles AHRR in CADs were pushed further ATDC from 4.4° (DF) to 5.8° (B15) and specific heat release angles AI 5, AI 10, AI 50 and AI 90 also were dislocated from 0.9°, 2.3°, 9.7°, 39.3° (DF) to 2.8°, 4.2°, 10.3°, 42.6° (B15), respectively, for considered loading conditions. This means that the combustion process of ethanol-Diesel fuel blends occurred later in the expansion stroke with higher heat losses in the cooling system. Because the angle Ap_{max}, was moved ATDC from 8.1° (DF) to 8.5° (B5), 8.6° (B10) and 9.4° (B15) towards a bigger cylinder volume, the maximum cylinder pressure p_{max} was only increased by nearly 1%, i.e. from 66.4 MPa (DF) to 67.1 MPa (B15). However, the cylinder pressure gradients increased from 5.85 bar/deg (DF) to 6.25 bar/deg, i.e. the fully loaded Diesel engine on ethanol-Diesel fuel blend B15 operated 6.8% rougher compared to the normal Diesel fuel case.

As figure 3 shows, the brake specific fuel consumption (bsfc) gradually decreased with engine load and its minimum values compiled 230.7 g/kW·h for the normal Diesel fuel and 239.4, 240.1 and 254.8 g/kWh for ethanol-Diesel fuel blends B5, B10 and B15. The 3.8%, 4.1%, and 10.4% higher brake specific fuel consumption can be mainly attributed to the lover both cetane number and net heating value of ethanol-Diesel fuel blends. However, difference in the heating value of the tested blends was probably not the only reason that led to the higher fuel consumption in grams per unit of energy developed because significant changes occurred in the combustion process.

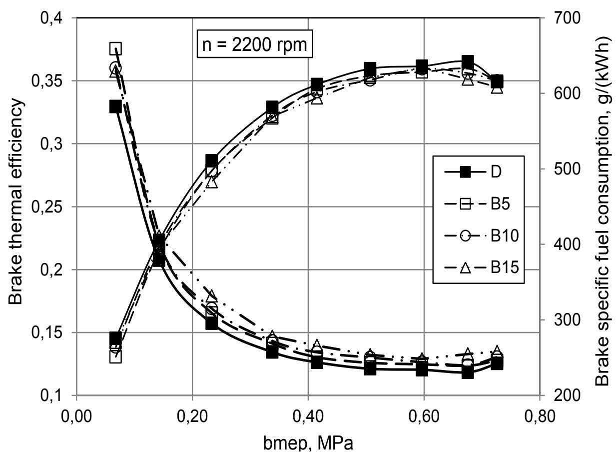

Fig. 3. The brake specific fuel consumption and the brake thermal efficiency as a function of brake mean effective pressure for the normal Diesel fuel and various ethanol-Diesel blends

The brake specific fuel consumption and net heating value of ethanol-Diesel blends were taken into account to calculate performance efficiency of the engine at tested loading conditions. As figure 3 shows, the brake thermal efficiency increased with engine load and reached the maximum value of 0.365 for the normal Diesel fuel and 0.360, 0.356 and 0.351 for ethanol-Diesel fuel blends B5, B10 and B15. This means that the maximum brake thermal efficiency was 1.4%, 2.5% and 3.8% lower for the tested oxygenated blends. Similar results have been found by other researchers [2, 3], who also obtained the brake thermal efficiency lover or nearly the same as conventional Diesel when operating at full (100%) load on ethanol-Diesel fuel blends up to 15% by volume.

Production of the NO_x emission depends on the combustion chamber and injection system design, engine load and speed, the fuel injection timing advance and autoignition delay caused by changes in physical properties, such as bulk modulus, viscosity and density of fuel blends, the fuel composition and the air and fuel mixture quality [6, 13]. An important role in the NO_x production also plays oxygen mass (weight) fraction stored in the fuel blend, its composition and chemical structure, including presence of double bonds in the molecular, as well as the engine performance efficiency related maximum cylinder gas temperature [10, 14]. Test results with a Case model 188D four cylinder, DI Diesel engine confirm that up to 60% of replacement of Diesel fuel by ethanol can be achieved however engine misfiring appears because of extreme autoignition delay and severe knocking occurs under some testing conditions [4].

It can be seen in figure 4 that the NO_x emissions gradually increased with the engine load and maximum cylinder gas temperature, however NO_x produced from the combustion of ethanol-Diesel fuel blends sustained at lover level within the entire load range tested at rated 2200 rpm speed. To be precisely, the NO_x emissions decreased from 1576.9 ppm (DF) to 1457.2 ppm (7.6%), 1447.1 ppm (8.2%) and 1329.5 ppm (15.7%) for ethanol-Diesel fuel blends B5, B10 and B15 used at full (100%) engine load of bmep = 0.68 MPa and rated 2200 rpm speed. Actually, the NO_x emission decrease was the bigger the higher the percentage of ethanol added to Diesel fuel. The emissions of NO_x were decreased despite unfavourable influence of low cetane number of oxygenated blends, long autoignition delay (Fig. 1), thus more fuel premixed for rapid combustion in the kinetic phase and high maximum heat release rate (Fig. 2).

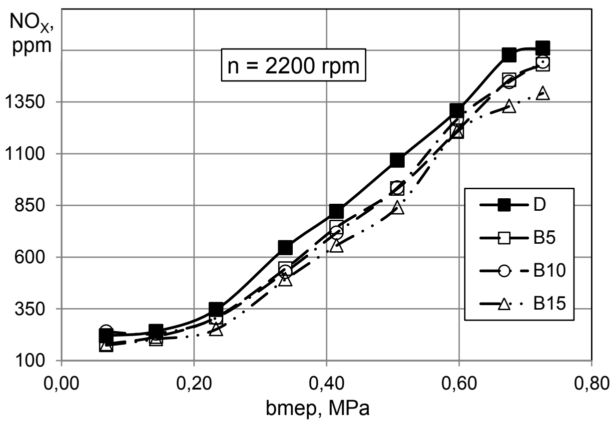

Fig. 4. Total nitrogen oxide NO_x emissions as a function of brake mean effective pressure for the normal Diesel fuel and various ethanol-Diesel blends

The first reason of such emissions' behavior was that the NO_x production is sensitive to the cooling effect caused by the ethanol-Diesel fuel sprays [13]. The second reason of decreased NO_x emissions can be attributed to stoichiometric air-to-fuel equivalence ratio, which is 37.3% lower for ethanol compared to that 14.45 of Diesel fuel. As a result, less atmospheric air-burn oxygen participated in combustion of overall identical combustible mixtures. The role of the ethanol conserved oxygen seems as not so much significant in NO and NO_x production than the cylinder air-born oxygen. The third reason can be associated with combustible mixture of ethanol-Diesel blends, which is less heterogeneous compared to the normal Diesel fuel because the fuel spray is shorter and its cone angle is wider that improves mixing with the cylinder air and contributes to the reduction of NO_x [14]. The test results of a turbocharged and intercooled 7.3 l Diesel engine T 444E HT showed that maximum cylinder gas pressures and temperatures decreased slightly with increasing percentage of the ethanol, therefore lower NO_x emissions were also observed, ethanol-Diesel fuel blend E10 decreased NO_x emissions by close to 3% [4].

The amount of carbon monoxide CO emissions depends on engine load, speed, the in-cylinder air-swirl turbulence intensity, the quantity of fuel delivered per each engine cycle, the air-fuel equivalence ratio and the fuel bound oxygen mass (wt%) content. The highest CO emission levels of 762 ppm, 764 ppm and 965 ppm were produced from the combustion of ethanol-Diesel fuel blends B5, B10 and B15 at light load bmep = 0.14 MPa and rated 2200 rpm speed (Fig. 5). The CO emissions for respective blends were 34.2%, 34.5% and 69.9% higher than normal Diesel produces (568 ppm) at considered loading conditions. As the engine load increased to bmep = 0.68 MPa, the CO

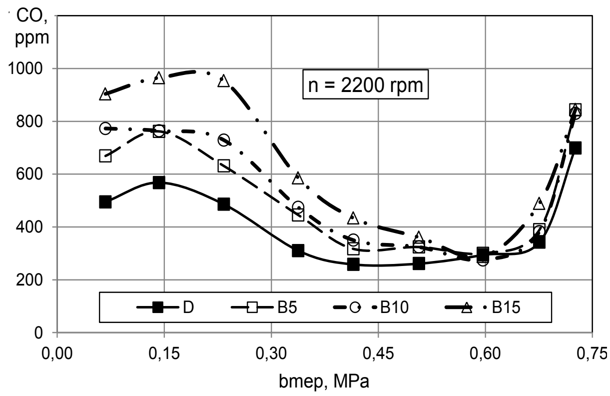

Fig. 5. Carbon monoxide CO emissions as a function of brake mean effective pressure for the normal Diesel fuel and various ethanol-Diesel blends

emission gradually decreased reaching nearly the lowest level of 342 ppm for Diesel fuel and 389, 382 and 489 ppm for blends B5, B10 and B15, i.e. the CO emission was 13.7%, 11.7% and 43.0% higher than that of normal Diesel at rated 2200 rpm speed. Despite slightly lower C/H ratio of ethanol-Diesel fuel blends (Table 1), significant CO increase at rated speed may occur due to worse operating properties of the ethanol. The higher CO emissions emanating in the entire load range match well with lower NO_x emissions (Fig. 4) produced by the combustion of ethanol-Diesel fuel blends.

Emissions of total unburned hydrocarbons HC generated by the combustion of ethanol-Diesel fuel blends B5 and B10 were 17.4% and 23.3% higher at low (bmep = 0.14 MPa) load, and 16.9% and 21.1% higher at full (bmep = 0.68 MPa) load operation compared to those, 860 ppm and 710 ppm, produced by normal Diesel at rated 2200 rpm speed. Whereas, the most oxygenated ethanol-Diesel fuel blend B15 (5.6wt% oxygen) suggested the HC emissions from 64.0% (310 ppm) to 74.6% (180 ppm) lower than normal Diesel produces at respective loading conditions. Both extremely low cetane number of the ethanol (8) and high latent heat of vaporisation varying from 840 to 880 kJ/kg are mainly responsible for long autoignition delay and slow combustion over late phases of the expansion stroke.

The HC oxidation later in the expansion stroke, after combustion stopped, resulted in lower the HC emission and residual oxygen O_2 in the exhaust accompanied by a higher gas temperature, but late combustion did not improve performance efficiency and effective power developed by the engine. This fact can be taken as an indicator that the Diesel engine would only be able to operate properly on comparably small percentages of the ethanol premixed to Diesel fuel, up to 15% by volume, as it would not be at full performance efficiency.

The experimental test results of a single cylinder Cummins 4 type engine indicate that with increased ethanol percentage in the blended Diesel fuel reduction in NO_x varied from zero to 4–5%. Both decreases and increases in CO emissions occurred, while total hydrocarbons (THC) increased substantially, but both were still well below the regulated emissions limit [7]. Park et al. [11] examined the influence of ethanol (99.9%) and Diesel fuel blends on combustion and exhaust emissions of a four-stroke, four cylinder, Bosch common-rail, Diesel engine run at 1500 rpm and various injection timings. By using high-speed camera Photron, Fastcam-APX RS authors estimated that 10vol% and 20vol% ethanol-Diesel fuel blends have a shorter spray tip penetration, a larger spray cone angle and smaller droplets compared to pure Diesel fuel (ULSD). An increase in the ethanol blending ratio led to a decrease in the NO_x and increase in the CO and HC emissions at the same loading conditions and injection timings.

Smoke opacity of the exhaust started from low level of 1.7–2.6 % at light engine load (bmep = 0.07 MPa) and reached 34.3% for the normal Diesel fuel and 44.2%, 32.7% and 45.9% for ethanol-Diesel fuel blends B5, B10 and B15 when operating at full (100%) load of bmep = 0.68 MPa (Fig. 6). This means that oxygenated fuel blends B5 and B15 produced the smoke opacity 28.9% and 36.7% higher than the normal Diesel fuel. According measurement methodology, the higher smoke opacity, which continued over the entire load range, does not always mean more unburned carbon particles were produced by the engine because vaporised aerosols and unburned fuel particles also presented in the manifold that affected transparency of the exhaust.

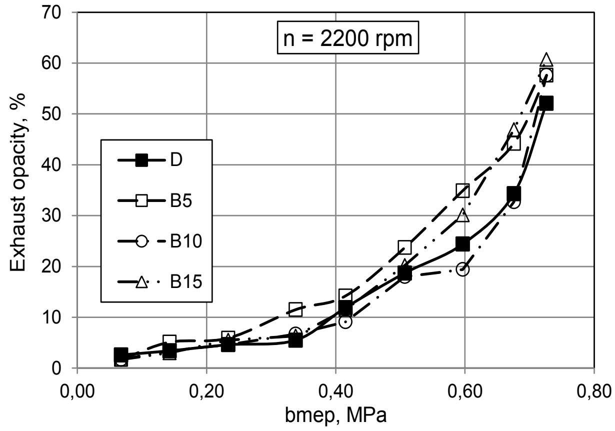

Fig. 6. Smoke opacity of the exhaust as a function of brake mean effective pressure for the normal Diesel fuel and various ethanol-Diesel blends

Higher smoke density produced by the combustion of ethanol-Diesel fuel blends B5 and B15 matches well with a bigger specific fuel mass consumption (Fig. 3), higher CO and HC emissions and reasonably lower the NO_x emission (Fig. 4). However, ethanol-Diesel fuel blend B10 suggested the smoke density lower in the load range of 0.07 to 0.68 PMa and slightly 4.7% better transparency than that (34.3%) of normal Diesel running at rated mode. Experiments conducted in a steel combustion chamber with 5vol%, 10 vol% and 20vol% ethanol-Diesel blends showed that blending Diesel fuel with additives having considerably higher H/C ratios improves the combustion process, reducing pollutants and soot mass concentration in the exhaust [5]. However, when using ethanol-Diesel blends the fuel bound oxygen may come into effect with a little help and, rather, to late to improve engine performance efficiency (Fig. 3), reduce the CO, the HC emissions and smoke opacity of the exhaust [2]. One can predict that autoignites and burns the Diesel fuel first and, afterwards, continues oxidation of the ethanol fraction within increasing cylinder volume in the expansion stroke. Such approach suggests that the ethanol comes into effect latter, however, with an essential help to accelerate oxidation processes in the diffusion phase. The ethanol bound oxygen, which is always on the spot ready to burn the fuel residues completely, may improve the NO_x emissions, however, it does not always lead to better performance efficiency, lower the CO, HC emissions and transparency of the exhaust, especially at unfavourable loading conditions.

Conclusions

Comprehensive experimental studies were conducted with a direct injection, four-cylinder, naturally aspirated 60 kW Diesel engine to evaluate the influence of using 5vol% (B5), 10vol% (B10) and 15vol% (B15) blends of anhydrous (99.8%) ethanol with the normal Diesel fuel when operating over wide range of loads at rated 2200 rpm speed.

It was determined that the autoignition delay φ_i increased due to anhydrous (200 proof) ethanol addition 5vol% (B5), 10vol% (B10) and 15vol% (B15) to arctic (class 2) Diesel fuel for all loads and speeds tested. To be precisely, the autoignition delay was 3.4%, 4.3% and 10.7% longer for respective ethanol-Diesel fuel blends than that (10.2^o) of the normal Diesel fuel when running at full (100%) load of bmep = 0.68 MPa and rated 2200 rpm speed.

The maximum heat release rate $(HRR)_{max}$ was increased from 116.4 kJ/m^3·deg (DF) to 139.0 (14.4%), 148.8 (27.8%) and 155.4 kJ/m^3·deg (33.5%) when running on ethanol-Diesel fuel blends B5, B10 and B15 at full (100%) load and rated 2200 rpm speed. The angles AHRR were pushed further ATDC from 4.4° (DF) to 5.8° (B15) and specific heat release angles AI 5, AI 10, AI 50 and AI 90 also were dislocated from 0.9°, 2.3°, 9.7°, 39.3° (DF) to 2.8°, 4.2°, 10.3°, 42.6° for oxygenated blend B15, respectively.

The higher 239.4, 240.1 and 254.8 g/kWh brake specific fuel consumption of ethanol-Diesel fuel blends B5, B10 and B15 compared to that 230.7 g/kWh of normal Diesel can be attributed to lower both, the cetane number and net heating value of the ethanol. The maximum brake thermal efficiency 0.365 of normal Diesel decreased to 0.360 (1.4%), 0.356 (2.5%) and 0.351 (3.8%) when using ethanol-Diesel fuel blends B5, B10 and B15 at rated 2200 rpm mode.

The NO_x emission decreased from 1576.9 ppm (DF) to 1457.2 ppm (7.6%), 1447.1 ppm (8.2%) and 1329.5 ppm (15.7%), respectively, when running on ethanol-Diesel fuel blends B5, B10 and B15 at full (100%) load of bmep = 0.68 MPa at rated 2200 rpm speed. The NO_x emission decrease was the bigger the higher the percentage of ethanol added in the fuel blend that can be attributed to cooling effect of the ethanol and reduced performance efficiency of the engine.

The highest 762 ppm (34.2%), 764 ppm (34.5%) and 965 ppm (69.9%) CO emission compared to normal Diesel (568 ppm) was produced from the combustion of ethanol-Diesel fuel blends B5, B10 and B15 at light load of bmep = 0.14 MPa. The CO emission reduced with increased engine load to bmep = 0.68 MPa and was 13.7%, 11.7% and 43.0% higher than normal Diesel produces (342 ppm) at rated 2200 rpm speed.

The HC emission generated by the combustion of ethanol-Diesel fuel blends B5 and B10 was 17.4% and 23.3% higher at low load of bmep = 0.14 MPa, and 16.9% and 21.1% higher at full load of bmep = 0.68 MPa compared to those, 860 ppm and 710 ppm, produced by normal Diesel running at rated 2200 rpm speed. The most oxygenated fuel blend B15 (5.6wt% oxygen) suggested the HC emission from 64.0% (310 ppm) to 74.6% (180 ppm) lower than normal Diesel produces at respective loading conditions.

The smoke opacity of the exhaust was 34.3% for the normal Diesel fuel and 44.2%, 32.7% and 45.9% for ethanol-Diesel fuel blends B5, B10 and B15 tested at full (100%) engine load of bmep = 0.68 MPa and rated 2200 rpm speed. Slightly higher smoke opacity suspended in the atmosphere does not always mean more unburned carbon particles were produced by the engine because vaporised aerosols and unburned fuel particles also contributed to aggravate transparency of the exhaust.

References

1. MINTEER S.: Alcoholic Fuels. Published by CRC Press at Taylor & Francis Group, LLC, United States of America, 6000 Broken Sound Parkway NW, Suite 300, 2006.
2. RAKOPOULOS C.D., ANTONOPOULOS K.A., RAKOPOULOS D.C.: Experimental heat release analysis and emissions of a HSDI Diesel engine fuelled with ethanol-Diesel blends. Energy, Vol. 32, 2007, 1791–1808.
3. ABU-QUDAIS M., HADDAD O., QUDAISAT M.: The effect of alcohol fumigation on Diesel engine performance and emissions. Energy Conversion and Management, Vol. 41, 2000, 389–399.

4. SHROPSHIRE G.J., GOERING C.E.: Ethanol injection into a Diesel-engine. Transactions of the ASAE, Vol. 25(3) 1982, 570–575.
5. ASFAR K.R., HAMED H.: Combustion of fuel blends. Energy Conversion and Management, Vol. 39, Issue 10, 1998, 1081–1093.
6. HANSEN A.C., GRATTON M.R., YUAN W.: Diesel engine performance and NO_x emissions from oxygenated biofuels and blends with Diesel fuel. Transactions of the ASAE, Vol. 49 (3), 2006, 589–595.
7. HANSEN A.C., ZHANG Q, LYNE P.-W.L.: Ethanol-Diesel fuel blends – a review. Bioresource Technology, Vol. 96, 2005, 277–285.
8. LI W., REN Y., WANG X-B., MIAO H., JIANG D-M., HUANG Z-H.: Combustion characteristics of compression ignition engine fuelled with Diesel-ethanol blends. Proc. ImechE, Vol. 222, 2008, Part D: J. Automobile Engineering, 265–274.
9. LABECKAS G., SLAVINSKAS S.: Comparative performance of direct injection Diesel engine operating on ethanol, petrol and rapeseed oil blends. Energy Conversion and Management, Vol. 50, 2009, Issue 3, 792–801.
10. LABECKAS G., SLAVINSKAS S.: Study of exhaust emissions of direct injection Diesel engine operating on ethanol, petrol and rapeseed oil blends. Energy Conversion and Management, Vol. 50, 2009, Issue 3, 802–812.
11. PARK S.H., YOUN I.M., LEE C.S.: Influence of ethanol blends on the combustion performance and exhaust emission characteristics of a four-cylinder Diesel engine at various engine loads and injection timings. Fuel, Vol. 90, 2011, 748–755.
12. LI W., REN Y., WANG X-B., MIAO H., JIANG D-M., HUANG Z-H.: Combustion characteristics of compression ignition engine fuelled with Diesel-ethanol blends. Proc. ImechE, Vol. 222, 2008, Part D: J. Automobile Engineering, 265–274.
13. LOTKO W., LUKANIN V.N., KHATCHIYAN A.S.: Using of Alternative Fuels in Internal combustion Engines. MADI, Moscow 2000 (in Russian).
14. HEYWOOD J.B.: Internal Combustion Engine Fundamentals. Co – Singapore for manufacture and export (International edition) 1988.