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Efficient and Ecological Indicators of CI Engine Fuelled with Different Diesel and LPG Mixtures

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Abstract

The effect of additionally fed LPG (Liquefied Petroleum Gas) on the energy and environmental performance of a compression ignition (CI) engine that operates of diesel fuel and the combustion process in it has been examined. In course of the experimental tests, it was found that additionally fed LPG reduced the volume of diesel fuel injected into the cylinder and caused a delay of its injection moment, thus affecting the combustible mixture combustion process and reducing the engine efficiency. Increasing the LPG concentration causes growing of the concentrations of partial combustion products, such as CO (carbon monoxide) and HC (hydrocarbons), in the exhaust gas and the smokiness of the latter; however, emissions of NO_x (nitrogen oxides) and CO₂ (carbon dioxide) decrease. If 20–60% LPG is added to diesel fuel and the fuel injection advance angle Θ is up to 16° CA (crank angle) bTDC (before the Top Dead Center), the engine efficiency is close to the efficiency of the one operating on diesel fuel, because combustion of the combustible mixture is improved and the concentration of the partial combustion products decreases; however, NO_x concentration in the combustion products grows. The analysis of the combustion process upon applying AVL BOOST software showed that increasing the LPG concentration in the mixture of fuels and the injection advance angle Θ causes shortening of combustion duration and the *ROHR* (*Rate of Heat Release*) becomes more intensive. The latter cause increases mechanical and thermal loads of the engine details.

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1. Introduction

Environmental protection and ecology are important factors today. The air quality is mostly affected by mobile sources of pollution. Transport, especially motor vehicles, is the largest source of environmental pollution. Abundant multidirectional activities for minimization of the environmental pollution caused by ICE (internal combustion engines) take place. The major pollutants caused by usual hydrocarbon fuels include unburnt and partially burnt hydrocarbons, carbon dioxide, nitrogen oxides and solid particles harmful for human health [1]. Combustion products from ICE, mostly CO₂ and CO, together with other greenhouse gases, such as methane and nitrogen oxides, impact the changes of the global temperature [2]. In presence of the above-mentioned problems, the today motor industry confronts an increasingly growing challenge related to an increased power, improved fuel consumption and minimization of pollutant emissions [3–6]. Engines are equipped with combustion products recirculation (exhaust gas recirculation, or EGR) equipment, various catalysers, solid particles filters and so on. The said measures enable reducing the pollutants emission after the combustion process; however, in order to satisfy the provisions of the increasingly tightening norms of combustion products emissions, we should strive for minimization of the pollution during the combustion process by use of alternative fuels or their mixtures [7, 8].

In the Communiqué of the European Commission on the European Alternative Fuels Strategy, the main alternative fuels that could replace petroleum or reduce emissions caused by vehicles are assessed. The provided list of alternative fuels includes electric energy, hydrogen, biofuels, and LPG [9]. With regards to the gaseous fuels, LPG is most frequently used in cars both in Lithuania and Europe. Its popularity is predetermined by comparatively low price of gas that is about a half of diesel fuel or petrol prices, sufficiently developed network of gas equipment and service centers and the wide network of filling stations [10, 11]. The targets of the European Commission in the transport sector by the year 2020 include reduction of greenhouse gas emissions by up to 20%; however, if other developed countries assume similar obligations according to a wide-scale global agreement and developing countries contribute in accordance with their possibilities, the reduction of greenhouse gas emissions by up to 30% should be a real target. In addition, it is strived for an improvement of fuel consumption efficiency by 20% [12].

In a diesel engine, various gases (such as LPG, compressed natural gas, biogas, and hydromethane) may be used as an alternative to diesel fuel by a partial replacing of the diesel fuel by them. Usually, the said gaseous fuels are used as secondary fuels and the diesel fuel is considered a primary fuel required for starting the combustion process. Because of high cetane number, the injected diesel fuel ignites in a high temperature caused by the compression and then causes an ignition of the secondary fuel [2]. LPG distinguishes itself for a number of advantages over traditional fuels. The gas does not dilute the engine oil, so the period of use of the latter becomes longer. High cetane number of petroleum gas predetermines its good anti-knock properties. The calorific capacity of LPG is higher, so a higher power of the engine may be achieved at lower fuel consumption [13]. Petroleum gas is almost free of sulphur admixtures. Reduction of sulphur dioxide in an engine is an advantage of a great importance, because the said compound activates corrosion, so in this aspect, the service life of the engine will be longer on using the diesel fuel with gas [14]. C/H ratio in LPG is lower, as compared to diesel fuel or petrol, so LPG combustion causes lower CO₂ emission. CO₂ emission caused by LPG combustion is lower by 10%, as compared to petrol [15]. The principal properties of diesel fuel and LPG are provided in Table 1.

While examining the energy and environmental performance of a diesel engine, Saleh had found that the maximum engine efficiency was obtained, when the engine operated on a mixture of diesel fuel (60%) and LPG (40%) [14]. In the said case, tests with LPG of different compositions were carried out and it was found that propane (otherwise that butane) caused the reduced CO emissions but increased NO_x emissions. So, it was agreed and accepted that the best combustible mixture of diesel fuel (60 %) and LPG (40%) is obtained when LPG consists of propane (70%) and butane (30%). When the engine with 100% load operated on the said mixture, the NO_x and SO₂ emissions decreased by 27% and 69%, respectively, as compared to usual diesel fuel; however, CO concentration increased by 15.7%.

If LPG is additionally fed, it is possible (at the maximum load of the engine) to increase the engine efficiency by up to 6% and to reduce the smokiness by up to 71%, as compared to diesel fuel [16].

In course of the research, the following problem had been obtained: when LPG is used, poor combustion of fuels takes place in a diesel engine at a low load of it [2]. If an engine with 20% load operates on a mixture of LPG and

diesel fuel, HC concentration in the combustion products increases up to 16 times and CO concentration – up to 4 times. When the load is increased up to 100%, the HC concentration increases 4 times and CO concentration – by 27% [16].

On variation of the fuel injection advance angle Θ , when 30% – 70% of LPG is additionally fed, it was found that injection of diesel fuel should be advanced by $\sim 10^\circ$ CA to ensure the maximum efficiency of the CI engine, because when the gas is ignited by diesel fuel according to the algorithm of the electronic control unit (ECU), its ignition appears to be delayed [18].

The goal of the present research is to establish the changes of the energy and environmental performance of a CI engine. Also to carry out an analysis of the heat release and pressure rise in the cylinder during combustion on variation of the additionally fed LPG concentration in the mixture in a wide range when diesel injection timing is advanced and to determine what is the maximum LPG concentration in order to avoid engine thermal and mechanical overload.

2. Material and methods

Dual fuel was used during the experiment: diesel and LPG gas, which consisted of propane (C_3H_8) and butane (C_4H_{10}) in the ratio of 50%/50%. The main LPG gas and diesel fuel properties are given in the Table 1. The tests were carried out at the moment (Θ) of starting the diesel fuel injection set by ECU (Fig. 2) and when $\Theta = 16^\circ$ CA bTDC. In both cases, LPG concentration in the combustible mixture was varied from 0% to 75%.

1.9 TDI (1Z type) CI engine with electronic controlled BOSCH VP37 distribution type fuel pump and turbocharger was used for tests. EGR system was disconnected during tests. The test engine parameters shown in Table 2.

Table 1. Fuel properties [2, 20, 21].

Properties	LPG		Diesel
	Propane	Butane	
Chemical formula	C_3H_8	C_4H_{10}	$C_{10}H_{22}-C_{15}H_{32}$
Composition by weight %	82C–18H	83C–17H	86C–13H
Lower heating value MJ/kg	46.3	45.6	42.5
Density of liquid at 15° C kg/l	0.51	0.58	0.83
Auto ignition temperature $^\circ$ C	470	365	250
Boiling temperature $^\circ$ C	–42	–0.5	150–360
Stoichiometric A/F ratio kg/kg	15.6	15.4	14.5
Octane number	105	94	30
Cetane number	–5–0		45–58
Flame speed (cm/s)	38–40		2.0–8.0

Table 2. Parameters of tested engine 1.9 TDI 1Z.

Parameter	Value
Displacement (cm^3)	1896
Number of cylinders	4
Compression ratio	19.5
Power (kW)	66 (4000 min^{-1})
Torque (N m)	180 (2000–2500 min^{-1})
Bore (mm)	79.5
Stroke (mm)	95.5
The intake valve opens	13° CA before TDC
The intake valve closes	25° CA after BDC
The exhaust valve opens	28° CA before BDC
The exhaust valve closes	19° CA after TDC

Gas was supplied out of 50 l (2.5 MPa) gas cylinder, through the low pressure reducer (36 mbar, the gas throughput of 4 kg/h) to the intake manifold before the turbocharger (Fig. 1). Tests were carried out with the engine at $n = 2000 \text{ min}^{-1}$ rated speed and efficient torque $M_e = 60 \text{ Nm}$ (load $\sim 30\%$). This corresponds roughly to the operation mode of the car Audi B4 engine evenly on a road, at the fifth gear and speed of 90 km/h.

The engine efficient torque M_e and crankshaft speed n is determined with a stand KI-5543. The torque measurement error is $\pm 1.23 \text{ Nm}$. Diesel fuel consumption B_D was measured by measure SK-5000 with electronic scale and stopwatch, LPG gas by gas meter KG-0095-G06-94-10. The accuracy of the fuel consumption measurement was 0.5%. Pollutants in the exhaust gas were measured using AVL DICOM 4000 gas (CO , CO_2 , HC , NO_x) analyzer and smoke – AVL DiSmoke.

Cylinder pressure p is fixed by piezo sensor GG2-1569 integrated in the preheating plug and recorded using AVL DiTEST DPM 800 equipment. Pressure measurement accuracy – 1%. The pressure in the intake manifold of engine

is measured with pressure gauge Delta OHM HD 2304.0. Sensor of this device TP704-2BAI is mounted against the intake manifold. Measurement error of ± 0.0002 MPa. The temperature of exhaust manifold T_m is determined by infrared thermometer Emsitest IR 8839, with a precision of $\pm 1.5^\circ \text{C}$.

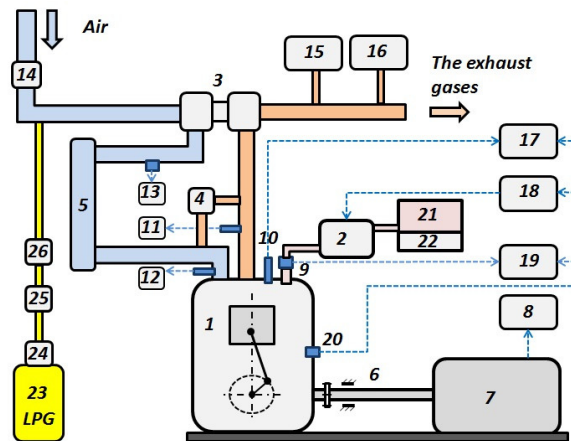


Fig. 1. The scheme of engine testing equipment:

1 – 1.9 TDI engine; 2 – high pressure fuel pump; 3 – turbocharger; 4 – EGR valve; 5 – air cooler; 6 – connecting shaft; 7 – engine load plate; 8 – engine torque and rotational speed recording equipment; 9 – fuel injection timing sensor; 10 – cylinder pressure sensor; 11 – exhaust gas temperature meter; 12 – intake gas temperature meter; 13 – air pressure meter; 14 – air mass meter; 15 – exhaust gas analyser; 16 – smoke analyser; 17 – cylinder pressure recording equipment; 18 – fuel injection moment control equipment; 19 – fuel injection moment recording equipment; 20 – crankshaft position sensor; 21 – fuel tank; 22 – fuel consumption calculation equipment; 23 – LPG cylinder; 24 – gas pressure reducer; 25 – gas flow control valve; 26 – gas meter. Source: authors.

The fuel injection timing is controlled by width of impulse, which depending on the injection timing is modulated by impulse generator. The actual start of the fuel injection timing is registered using a piezo sensor mounted on the tube of fuel supply to the injector. The actual start of injection timing was recorded with an AVL DiSystem 845 device.

The combustion synthesis process of the test engine carried out by simulation software AVL BOOST. The program used a two-zone combustion model [22]. According to the indicators of the tested fuel, an open thermodynamic system has been analyzed which exchange the mass and energy with other engine systems. It makes possible to form a working cycle model, sub-model processes such as: gas-change processes; pressure processes; combustion and expansion processes.

The intensity of heat release during the operating cycle is determined by using [23] heat release function:

$$\frac{dx}{d\varphi} = 6.908 \frac{m_v + 1}{\varphi_c} \left(\frac{\varphi}{\varphi_c} \right)^{m_v} \exp \left[-6.908 \left(\frac{\varphi}{\varphi_c} \right)^{m_v + 1} \right];$$

$$dx = \frac{dQ}{Q}, \quad (1)$$

where: Q – extracted amount of heat during the operating cycle; φ – crank angle; m_v – Vibe's combustion parameter; φ_c – duration of combustion expressed in crankshaft rotation angle.

The beginning of burning in the cylinder (start of combustion) φ_0 , duration φ_c and Vibe's combustion parameter m_v was determined by setting the engine operating parameters (pressure in the cylinder during of an operating cycle, the consumption of the fuel and air) in the AVL BOOST subprogram BURN.

3. The experimental results and discussion

On varying the concentration of the additionally fed LPG from 0% to 75% upon maintaining constant efficient torque $M_e = 60$ Nm and the engine's rotational speed $n = 2000$ min⁻¹, the moment of starting the diesel fuel injection (Θ) decreases from 5° to 2° CA bTDC (Fig. 2), because cyclic amount of diesel and injection duration is reducing. The LPG consumption B_{LPG} grows from 0 kg/h to 2.436 kg/h and diesel fuel consumption (B_D) falls from 3.243 kg/h to 1.006 kg/h (~69%) (Fig. 3). B_D falls, because when LPG concentration is increased and $M_e = 60$ Nm and $n = 2000$ min⁻¹ are constant, the accelerator is pressed less. According to the signal from the accelerator position sensor, the engine ECU controls the fuel pump BOSCH VP37 – reduces the cyclic fuel consumption and corrects (delays) Θ [15]. Simultaneously, the time of diesel fuel injection is shortened as well. The total fuel consumption B_{D+LPG} , when LPG concentration in the combustible mixture is increased to 75% and in absence of Θ correction, grows from 3.243 kg/h to 3.442 kg/h (~6.1%) (Fig. 3). If Θ is increased to 16° CA bTDC, the total fuel consumption reduces to 3.128 kg/h (~3.5%), because the gaseous fuel is ignited at the moment close to the optimum one.

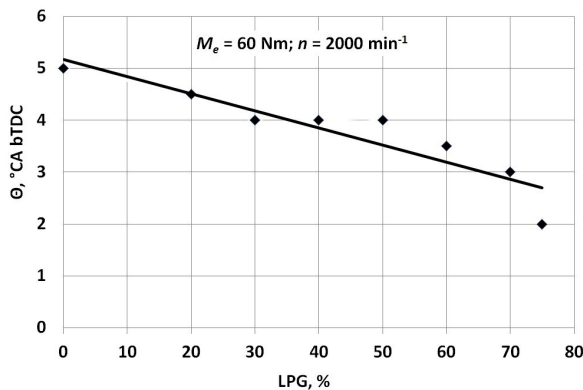


Fig. 2. The dependence of the diesel fuel injection advance angle (Θ) on LPG concentration in the combustible mixture. Source: authors.

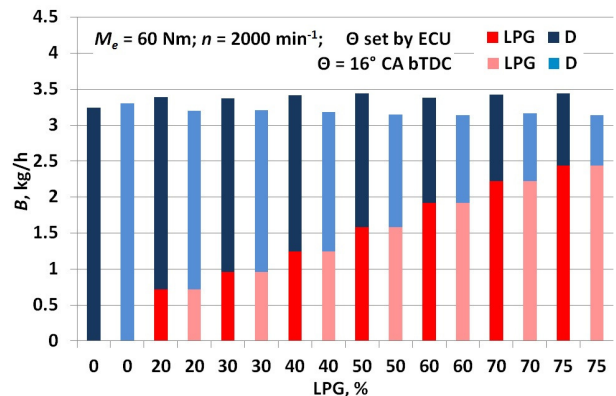


Fig. 3. The dependence of the fuel consumption (B) on the composition of the combustible mixture and the Θ . Source: authors.

On a comparison of the engine efficiency (η_e), it may be seen that when LPG replaces up to 75% of diesel fuel and at the standard diesel fuel injection moment, η_e falls down from 0.328 to 0.292 (~11%) (Fig. 4). η_e worsens because a delayed gas ignition and growing calorific value of the fuel mixture [24], because the calorific value of LPG (45.95 MJ/kg) is higher, as compared to diesel fuel (42.5 MJ/kg) (Table 1). When the diesel fuel injection advance angle is increased to 16° CA bTDC and LPG concentration is increased from 0% to 60%, η_e of the engine changes slightly and when LPG concentration achieves 75% – η_e falls down to 0.320 (~2.4%), although the calorific value of the combustible mixture grows.

On increasing the LPG content up to 75%, carbon dioxide concentration in the combustion products (Fig. 5) decreases because of lower C/H ratio in LPG (Table 1) and worsening combustion process [18]. If Θ is set by ECU, CO_2 falls from 4.8% to 4.1% (~14.5%), and if the injection advance angle is increased to 16° CA bTDC, reduction of CO_2 concentration (because of better combustion and higher temperature in cylinders) is less – to 4.3% (~10.4%).

Worsening of the combustion process caused by LPG additive is also confirmed by the growing concentration of partial combustion products (CO and HC) in the exhaust gas. If Θ is set by ECU and LPG concentration is increased from 0% to 60%, CO concentration in the exhaust gas grows from 0.01% to 0.36% (Fig. 6). When LPG concentration is increased from 60% to 75%, CO concentration in the exhaust gas falls down to 0.33%. If Θ is increased to 16° CA bTDC, CO concentration grows from 0.01% (0% LPG) to 0.16% (60% LPG), i.e. half as much than in the case, when Θ is set by ECU, because on earlier ignition of the mixture, the combustion process takes place in a smaller volume, so the combustion temperature increases and oxidation of carbon compounds is improved. When LPG concentration is increased from 60% to 75%, CO concentration in the exhaust gas falls down to 0.13%, because C/H ratio in the fuel falls.

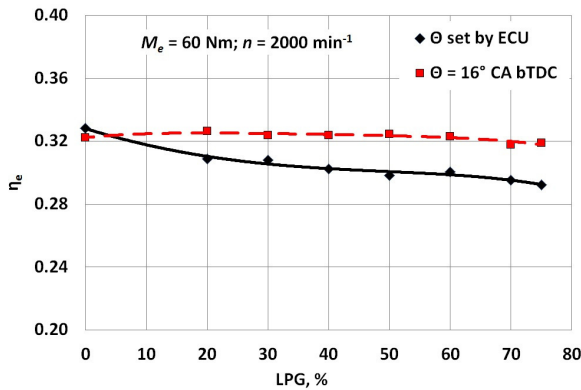


Fig. 4. The dependence of the engine efficiency η_e on the composition of the combustible mixture and Θ . Source: authors.

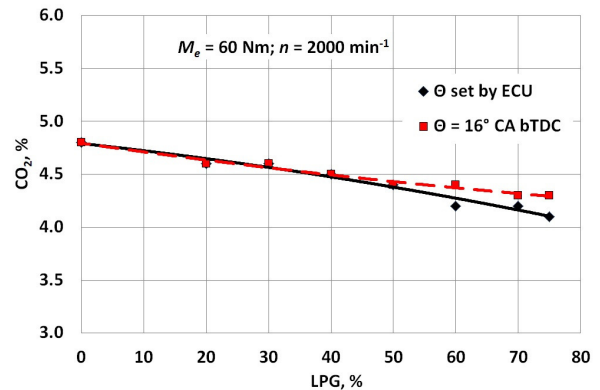


Fig. 5. The dependence of CO_2 concentration in the exhaust gas on the composition of the combustible mixture and Θ . Source: authors.

When LPG concentration is increased from 0% to 75%, HC concentration grows from 13 ppm to 715 ppm (Θ set by ECU) and from 10 ppm to 399 ppm ($\Theta = 16^\circ$ CA bTDC) (Fig. 7). In addition, the combustion process worsens, because LPG occupies a part of volume in the injected air causing a reduction of oxygen concentration in the combustible mixture. Furthermore, the trapped gaseous fuel in the piston land and other crevices may escape to the atmosphere either as product of partial combustion [25].

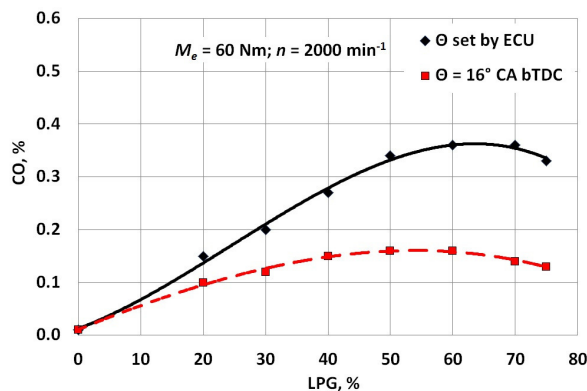


Fig. 6. Source: authors. The dependence of carbon monoxide (CO) concentration in the exhaust gas on the composition of the combustible mixture and Θ . Source: authors.

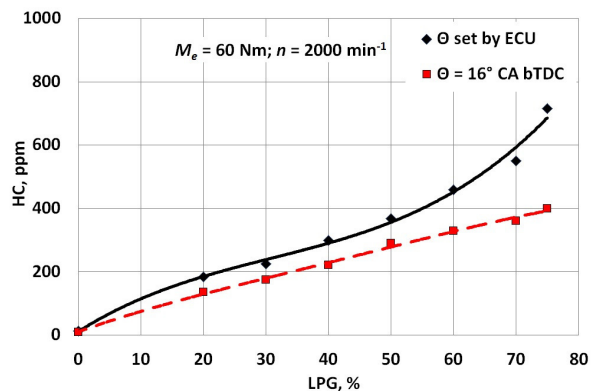


Fig. 7. Source: authors. The dependence of concentration of nitrogen oxides (NO_x) in the exhaust gas on the composition of the combustible mixture and Θ . Source: authors.

When from 0% to 50% of diesel fuel is replaced with LPG, the smokiness is growing because of decreasing content of oxygen in the combustible mixture, worsening combustion and reduced combustion temperature (Fig. 8). In addition, when LPG is used, abundant vapour is formed and it affects the smokiness (the optical density). If Θ is set by ECU, the smokiness grows from 7% to 8.5% (~21%). However, on further increasing the LPG concentration, the smokiness begins reducing because of reduction of carbon content in the combustible mixture. When Θ is increased to 16° CA bTDC, the smokiness (because of better combustion and higher combustion temperature that cause reduction of the concentration of partial combustion products) falls down from 8.5% to 5.4% (-36%).

When LPG concentration is increased from 0% to 75% and Θ is set by engine ECU, the concentration of NO_x in the combustion products falls from 363 ppm to 80 ppm (Fig. 9), because the fuel injection delayed by ECU (Fig. 2) causes shortening the diesel fuel ignition delay phase and reduction of temperature in the kinetic combustion phase [26]. On increasing LPG content, the volume of oxygen required for NO_x formation reduces. If the injection advance angle is increased to $\Theta = 16^\circ$ CA bTDC, NO_x concentration in the whole range of variation of LPG concentration increases

3.6–8.4 times. However, when diesel fuel is replaced with 75% LPG and the injection advance angle Θ is increased to 16° CA bTDC, NO_x concentration increases by $\sim 80\%$ (from 363 ppm to 670 ppm) only.

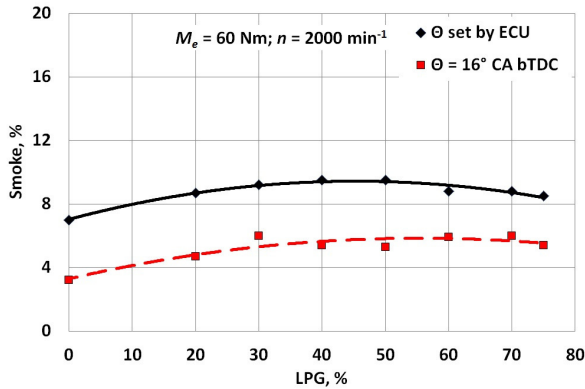


Fig. 8. The dependence of the smokiness of the exhaust gas on the composition of the combustible mixture and Θ . Source: authors.

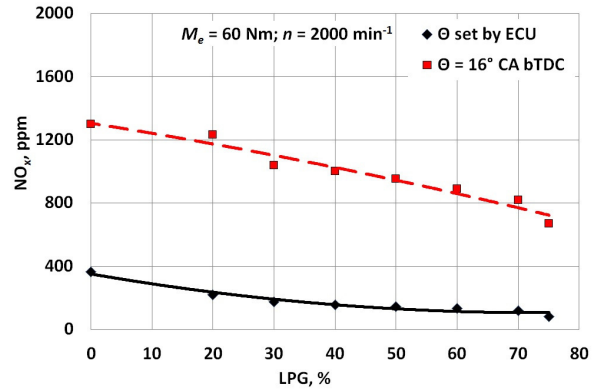


Fig. 9. Source: authors. The dependence of concentration of nitrogen oxides (NO_x) in the exhaust gas on the composition of the combustible mixture and Θ . Source: authors.

4. Engine work cycle simulation using AVL BOOST software

An analysis of the combustion process was carried out for a case when diesel fuel injection advance angle Θ was increased to 16° CA bTDC and LPG concentration varied from 0% to 75%. An analysis of Figs 10–13 shows that increasing the LPG concentration does not proportionally affect the combustion process.

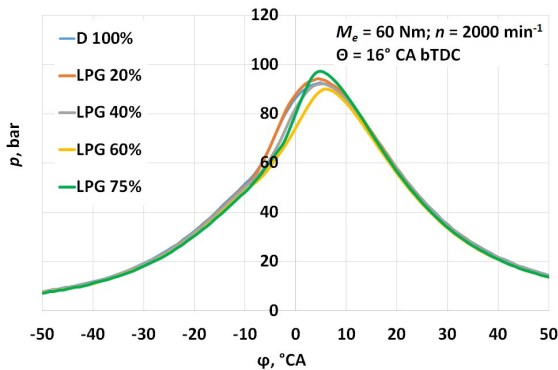


Fig. 10. Dependence of the pressure in the cylinder (p) on LPG concentration. Source: authors.

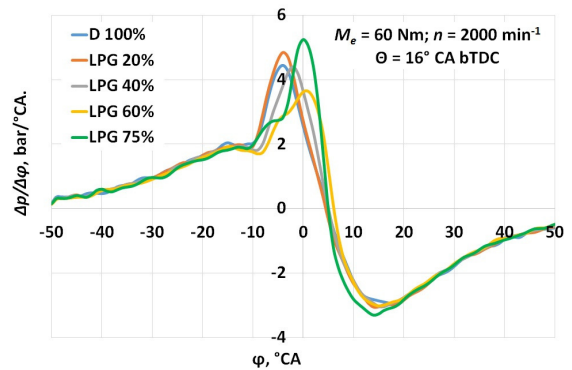


Fig. 11. Dependence of the pressure increase ($\Delta p / \Delta \phi$) in the cylinder on LPG concentration. Source: authors.

During research of similar 1.9 TDI 1Z engine, in the engine mode under research ($n = 2000 \text{ min}^{-1}$ and $M_e = 60 \text{ Nm}$), if the combustible mixture includes 40–60% LPG, the maximum engine efficiency is achieved when the injection advance angle was changed from $\Theta \approx 4^\circ$ CA bTDC (which was set by ECU) to $\Theta = 16^\circ$ CA bTDC [18]. For this reason, a fixed $\Theta = 16^\circ$ CA bTDC was chosen. At 20% LPG concentration, the maximum pressure in the cylinder is achieved earlier, as compared to the engine operation on 100% diesel fuel (D); however, at LPG 40% and LPG 60%, the maximum pressure is achieved later. Furthermore, at LPG 75%, is achieved earlier again and is higher, as compared to the engine operation on 100% diesel fuel (Fig. 10). The lowest maximum pressure increase ($\Delta p / \Delta \phi = 3.64 \text{ bar}/^\circ\text{CA}$) is achieved at LPG 60% and it is lower, as compared to the engine operation on 100% diesel fuel ($4.45 \text{ bar}/^\circ\text{CA}$). However, at LPG 75%, the maximum pressure increase achieves $5.25 \text{ bar}/^\circ\text{CA}$. Plausible

that using 75% LPG, the combustion intensity increases because of knock phenomena. Higher pressure increase causes a larger mechanical load of the engine parts [26].

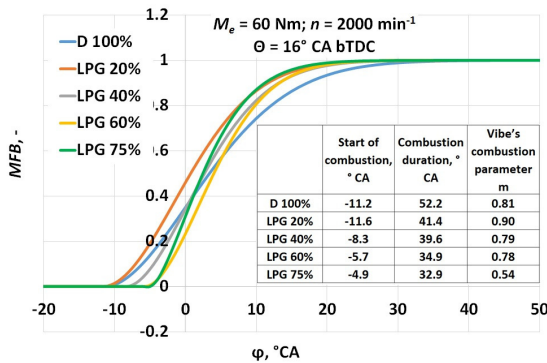


Fig. 12. Dependence of the Mass Fraction Burned (MFB) on LPG concentration. Source: authors.

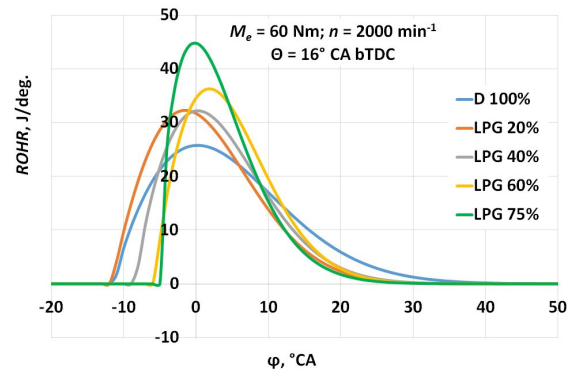


Fig. 13. Dependence of the Rate of Heat Release (ROHR) on LPG concentration. Source: authors.

While examining the Mass Fraction Burned (MFB) and the Rate of Heat Release (ROHR) diagrams (Fig. 12 and Fig. 13), we can see that when LPG concentration is being increased from 0% to 75%, the ignition delay period is extended (the start of combustion delays) and the combustion duration becomes shorter. It is caused by decreasing quantity of the injected diesel fuel (Fig. 3). When LPG concentration is being increased, the maximum Rate of Heat Release grows as well, because an increasingly growing LPG quantity (ignited by diesel fuel) is burning together with the injected diesel fuel and, in course of reduction of the cyclic consumption of diesel fuel, the combustion duration becomes shorter. When LPG concentration is increased and the flammability limit (2.15–9.6% volume in air) [2] is achieved, it is sufficient to ignite LPG only and then gas combustion will take place.

5. Conclusions

In the analysis of the experimental tests of a compression ignition engine operating at middle revolutions of the crankshaft and 30% load, where the moment of diesel fuel injection was altered and a wide range of mixtures of diesel fuel with LPG was used, it was found that:

1. When LPG concentration is increased, the volume of injected diesel fuel is reduced thus causing a reduction of the fuel injection advance angle Θ (set by ECU), so the combustible mixture is ignited later and the combustion temperature is reduced (the combustion occurs in a larger volume).
2. Addition of LPG to diesel fuel causes a reduction of CO_2 (that contributes to the greenhouse effect) concentration in the combustion products, because C/H ratio in the combustible mixture becomes lower. Because LPG additive worsens air filling of cylinders, reduces the combustion rate and temperature, the combustion extends to the final combustion phase and a higher share of partial combustion products (CO and HC) is emitted to the environment; however, NO_x emission decreases.
3. Upon striving to reduce fuel consumption and avoid worsening the engine efficiency η_e , when a compression ignition engine operates on mixtures of diesel fuel and gaseous fuels, it is necessary to increase the diesel fuel injection advance angle Θ by varying ECU algorithm and ensuring the optimum rate of combustion of the combustible mixture in the minimum volume.
4. If LPG mass concentration in the combustible mixture is increased to 60% and the fuel injection starting moment Θ is advanced, the achieved engine efficiency η_e is close to that for diesel fuel. However, upon taking into account a lower price of LPG fuel, use of higher LPG concentrations ensures higher economic efficiency.
5. If the diesel fuel injection starting moment Θ is advanced and LPG concentration is increased, the maximum Rate of Heat Release grows, because the increasingly growing LPG quantity ignited by diesel fuel is burning together with the injected diesel fuel more rapidly and, upon a reduction of cyclic consumption of diesel fuel,

the duration of combustion becomes shorter. When LPG mass concentration in the combustible mixture concentration in the combustible mixture is over 60%, the pressure increase as well as the mechanical and thermal loads of the engine details grows considerably.

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References

- [1] D. C. Rakopoulos, D. T. Hountalas, E. C. Kakaras, E. G. Giakoumis, R. G. Papagiannakis, Investigation of the performance and emissions of bus engine operating on butanol/diesel fuel blends, *Fuel* 89 (2010) 2781–2790.
- [2] D. B. Lata, Investigation of dual fuel diesel engine with hydrogen and LPG fuel, Birla Institute of Technology, Lambert Academic Publishing, Berlin. 2012.
- [3] A. T. Ergenc, D. O. Koca, PLC controlled single cylinder diesel–LPG engine, *Fuel* 130 (2014) 273–278.
- [4] D. B. Lata, A. Misra, S. Medhekar, Investigations on the combustion parameters of a dual fuel diesel engine with hydrogen and LPG as secondary fuels, *International Journal of Hydrogen Energy* 36 (2011) 13808–19.
- [5] W. M. Budzianowski, R. Miller, Towards improvements in thermal efficiency and reduced harmful emissions of combustion processes by using recirculation of heat and mass: a review, *Recent Patents on Mechanical Engineering* 2 (2009) 228–39.
- [6] E. Johnson, LPG: a secure, cleaner transport fuel? A policy recommendation for Europe, *Energy Policy* 31 (2003) 1573–1577.
- [7] H. S. Tira, J. M. Herreros, A. Tsolakis, L. W. Wyszynski, Influence of the addition of LPG-reformate and H₂ on an engine dually fuelled with LPG–diesel, –RME and –GTL Fuels, *Fuel* 118 (2014) 73–82.
- [8] K. Laurinaitis, S. Slavinskas, Homogeninių degalų ir oro mišinių savaiminio užsiliepsnojimo HCCI variklyje eksperimentinis tyrimas. [Experimental study of homogeneous air fuel mixture autoignition in HCCI engine], *Research papers of Aleksandras Stulginskis University* 44(1–3) (2012) 254–264.
- [9] Directive 2014/94/eu of the European parliament and of the council on the deployment of alternative fuels infrastructure. Available from Internet: <http://eur-lex.europa.eu/legal-content/EN/TXT/HTML/?uri=CELEX:32014L0094&from=LT>, 2014 (accessed 16.12.23).
- [10] L. Raslavičius, A. Keršys, S. Mockus, N. Keršienė, M. Starevičius, Liquefied petroleumgas (LPG) as a medium-term option in the transition to sustainable fuels and transport, *Renewable and Sustainable Energy Reviews* 32 (2014) 513–525.
- [11] V. Katinas, J. Savickas, Dujinių degalų vartojimo transporto plėtos analizė [The analysis of development of gaseous fuel use for transport], *Research papers of Aleksandras Stulginskis University* 44(1–3) (2012) 144–153.
- [12] Europa 2020. Sustainable growth – for a resource efficient, greener and more competitive economy. Available from Internet: http://ec.europa.eu/europe2020/europe-2020-in-a-nutshell/priorities/sustainable-growth/index_en.htm, 2012 (accessed 16.12.23).
- [13] M. T. Chaichan, Exhaust analysis and performance of a single cylinder diesel engine run on dual fuels mode, *Journal of Engineering* 17(4) (2011) 873–885.
- [14] H. E. Saleh, Effect of variation in LPG composition on emissions and performance in a dual fuel diesel engine, *Fuel* 87 (2008) 3031–3039.
- [15] K. Reif, Diesel engine management: systems and components, Springer Vieweg, Wiesbaden. 2014.
- [16] P. Vijayabalan, G. Nagarajan, Performance, Emission and Combustion of LPG Diesel dual fuel engine using glow plug, *Jordan Journal of Mechanical and Industrial Engineering* 3(2) (2009) 105–110.
- [17] E. Elnajjar, M. O. Hamdan, M. Y. E. Selim, Experimental investigation of dual engine performance using variable LPG composition fuel, *Renew Energy* 56 (2013) 110–116.
- [18] A. Rimkus, M. Berioza, M. Melaika, R. Juknelevičius, Z. Bogdanovičius, Improvement of the compression-ignition engine indicators using dual fuel (diesel and liquefied petroleum gas), *Procedia engineering* 134 (2016) 30–39.
- [19] A. Cernat, C. Pana, N. Negurescu, C. Nutu, On combustion of diesel fuel drops at LPG fuelling by diesel gas method, *Procedia Technology* 22 (2016) 705–712.
- [20] R. Bosch, GmbH, Automotive Handbook, 8 th Edition, Bentley Publishers. 2011.
- [21] P. Jučas, Chemotologija, Lietuvos žemės ūkio universiteto Leidybos centras, Kaunas. 2006.
- [22] G. Stiesch, Modeling Engine spray and combustion processes, Springer Verlag Berlin Heidelberg, Berlin. 2010.
- [23] I. I. Vibe, Brennverlauf und Kreisprozeß von Verbrennungsmotoren, Verlag Technik, Berlin. 1970.
- [24] E. Sendzikiene, A. Rimkus, M. Melaika, V. Makareviciene, S. Pukalskas, Impact of biomethane gas on energy and emission characteristics of a spark ignition engine fuelled with a stoichiometric mixture at various ignition advance angles, *Fuel* 162 (2015) 194–201.
- [25] D. B. Lata, A. Misra, S. Medhekar, Effect of hydrogen and LPG addition on the efficiency and emissions of a dual fuel diesel engine, *International Journal of Hydrogen Energy* 37 (2012) 6084–6096.
- [26] J. B. Heywood, Internal Combustion Engine Fundamentals, McGraw Hill series in mechanical engineering. 1988.